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Westinghouse Astronuclear Laboratory

PHASE 1

Quarterly Progress Report,

March 1 through May 31, 1967

AEC Contract AT (30-1) - 3800

Prepared for the

U.S. ATOMIC ENERGY COMMISSION

Prepared June 9, 1967

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PREFACE

This report is the first quarterly progress report on the SNAP-23A Project, Advanced Terrestrial Radioisotope Power Systems, under Contract AT(30-1)-3800 authorized by the United States Atomic Energy Commission. Phase I, which covers the period from December 2, 1966 through March 31, 1968, is being conducted by the Astronuclear Laboratory of the Westinghouse Electric Corporation under the direction of the Atomic Energy Commission.

This document reports progress for the period from December 2, 1966 to May 31, 1967; however, the major portion of the work was performed in the quarter from March 1, 1967 to May 31, 1967.

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SUMMARY

A total of four reports were prepared and submitted to the AEC in fulfillment of SNAP-23A contract milestones. These include WANL-PP-(SS)-001, "Phase 1 Project Plan"; WANL-SS-001, "WANL-3M Interface Document"; WANL-SS-002, "SNAP-23A Conceptual System Design"; and WANL-SS-003, "SNAP-23A Product Assurance Program Plan".

A 60 watt(e) system optimization study considering size, weight and cost was completed. It was found that system weight is minimum at a system diameter of approximately 22 inches. The cost, however, continues to decrease with increasing diameter over the range of interest. A system diameter of 24 inches has been selected for the first mock-up system. This involves a slight compromise in weight but yields the lowest cost within the design objective of a 24 inch envelop.

Stress analysis of the fuel capsule was completed. The capsule walls and end caps were sized to withstand the hydrostatic pressure associated with accidental sea burial to a depth of 20,000 feet. The fuel pellet diameter for the 60 watt (e) system has been selected as 4 inches, based on trade-offs in system optimization and extension in ORNL development. This is the largest capsule considered in an RTG system to date.

The design of the first mock-up system, 60 watt (e), was completed. This system will use an electrically heated capsule and a thermoelectric converter supplied by the 3M Company. Drawings have either been released or will be released shortly for procurement of system hardware. Based on delivery estimates, it appears that there will be a delay in the test date unless order placement and necessary approvals are expedited at an earlier time.

1. PROGRAM MANAGEMENT

Subtask 110-Project Management

The Phase 1 Project Plan (WANL-PP(SS)-001) was prepared and submitted to the AEC for approval. This plan outlined the detailed work that will be performed during Phase 1 and presented schedules, manpower, and cost data associated with each subtask. Of prime significance is the schedule for the design, fabrication, and testing of two prototype 60 watt (e) systems which essentially will dictate the schedular requirements for analysis and design of the various components.

AEC comments on the Phase 1 Project Plan were received, and Revision 1 of the document was issued reflecting the following changes that (1) quarterly progress reports be submitted, (2) the capsule design subtask be revised to delete effort relating to final closure technology and inspection since ORNL has been given that area of responsibility, and (3) Task 400 be revised to provide more detail in objectives and approach.

An informal design review was held with the AEC on April 27, 1967. Action items resulted in the areas of safety analysis and capsule design. Issuance of the Safety Criteria Report, HIT-248, was delayed in order to incorporate items identified during the design review. This report has been revised and will be issued early in June.

During the May interface meeting with the 3M Company, a potential problem in the delivery of the second converter unit within the Phase 1 time span was identified. This problem developed from the requirement that test results of the first system mock-up be fed back into the design of second converter series. Resolution of this problem will be pursued during the next report period.

The major milestones to date have been the submittal of the Phase I Project Plan, the Westinghouse-3M Company Interface Document, the Reliability and Quality Control Program Plan, and the completion of the initial design of the 60 watt (e) system. As mentioned above, submittal of the safety criteria has been delayed to the next report period.

Subtask 130 - Applications Engineering

The objectives of this subtask are to determine possible applications in which SNAP-23A may exhibit significant advantages over other power systems and to introduce into the system design features that may enhance its applicability.

During the initial literature search for applications, work was also started on a brochure that describes the prior history and state-of-the-art of terrestrial RTG systems, summarizes the design features of SNAP-23A, and indicates some possible application of the system. This brochure was revised and expanded as the result of initial reviews and will be issued subsequent to AEC approval.

Among the initial group of possible applications that were tabulated (see Table 1-1) the near-term application that appears most promising is in research or navigational buoys. The requirements will become more clearly defined during the conduct of the national buoy study which Travelers Research is undertaking. However, current information on buoy applications is being used to guide a preliminary investigation of optimum means for adapting the basic SNAP-23A design for use in buoys.

A program is being organized for systematic contact of potential or current RTG system users in order to update and broaden the list of applications shown in Table 1-1. Those contacted will include both governmental and industrial activities.

Subtask 140 - Liaison Engineering

The objective of this subtask is to assure the integration of efforts of the 3M Company, Hittman Associates, and others associated with SNAP-23A into the Westinghouse program.

The objective has been met by conducting regular interface meetings with 3M and Hittman. Once a month with 3M and twice a month with Hittman. Additional topical meetings were arranged as required. Contact was maintained with ORNL with respect to several of their activities that relate to the SNAP-23A program. Arrangements were also made to meet with NRDL.

TABLE 1-1 POTENTIAL APPLICATIONS OF SNAP-23A POWER SYSTEMS

APPLICATIONS	Approximate Power - Watts(e)	No. of Potential Power Systems
MILITARY		
1) Navigational Aids	25	260
2) Oceanographic and Weather Monitors		
East Coast	25-100	40-150
West Coast	25-100	60-130
Sea Going	60	17
3) Miscellaneous Military Programs	100	(classified)
GOVERNMENT AND CIVILIAN		
1) Inter-agency Weather	25	10
2) Navigational Aids		
Alaskan and Aleutian Shore Light	25	68
Buoys - Alaskan Waters	25	88
Other Beacons and Buoys - U. S. Coast Guard	50	27
Off-shore Oil Well Rigs - Civilian	50	100
3) Unattended Seismic Stations	60	50
4) Missile Range Splash Detectors and Tracking Ship Position Fixers (STAR)	25	262
5) Weather Stations (AMR-PMR)	25	30

The complex interface between WANL and 3M has required almost daily communication during this period. Since the converter and the overall system are separately funded, an official interface document was prepared jointly by WANL and 3M and was made a contractual document by the AEC. This document, WANL-SS-001, "WANL-3M Interface Document," SNAP-23A Phase I, specifies the general responsibilities of both parties with regard to the specification, design, and testing of the converters. Experience gained during the early design stages of converter series No. 1 is now being used to define a set of additions to this document which will clarify the specifications applicable to each experimental system and converter.

The interface with Hittman Associates is, in effect, an internal WANL interface since the Hittman shielding and safety analyses provide input that is translated into hardware at WANL. Close cooperation here is obviously required. The capsule design and manufacture is a Hittman responsibility, but requires coordination with WANL and ORNL to assure compatibility with the system mechanical and thermal requirements and with ORNL handling and fabrication requirements.

2. PRODUCT ASSURANCE

Subtask 210-Reliability

The objective of this subtask is to conduct a reliability program to assure a high probability of satisfactory performance of the SNAP-23A system during the specified life.

The Reliability portion of the SNAP 23-A "Product Assurance Plan," WANL-(SS)-003, was prepared and submitted for AEC approval.

Discussions were held with 3M Quality Assurance to assure proper interfacing of the WANL and 3M reliability programs.

Preliminary failure-mode analyses were conducted for electrical as well as mechanical equipment. The subsystems were generally found to be of a high potential reliability, due in part to the simplicity of their design. The following recommendations have resulted from the studies to date:

1) It was recommended that the RA-type reference amplifier be replaced by a conventional transistor and Zener diode combination in the regulator circuit. The RA unit would be expected to experience 3.4 failures per million operating hours compared to 0.9 for the recommended components.

2) It was recommended that a fillet weld be changed to a butt weld for the heat source jacket. The object of the change is to improve the quality of the weld and to facilitate its inspection. The recommendation was based on the criticality of a failure in this area.

Quantitative reliability estimates were obtained for the regulator circuit on a preliminary basis. Mean time between failures (MTBF) was calculated for each component, based on data tabulated in MIL-HDBK-217. A consolidation of these figures disclosed that the regulator circuit, as presently conceived, can meet a reliability level of 90 percent for a ten-year unattended operation in shipboard-type environment. An increase in reliability above this value is considered feasible during subsequent refinement of initial design.

An evaluation of the safety circuit associated with the power regulator was performed. The conclusion was reached that the safety circuit degraded overall system reliability rather than enhancing it. Therefore, a recommendation was made to remove the circuit. The following reasons were given:

- 1) Premature operation of the safety circuit would disable the system. The added system complexity increases this likelihood.
- 2) Reliable operation of the shorted condition may not be possible to achieve, therefore, single faults may disable the system indefinitely in one instance, or may be oscillatory in nature at some other time.
- 3) To make the safety circuit more independent, sensor redundancy is required, increasing the requirements for mechanical attachments and penetration which have a significant effect on reliability over the ten-year mission.
- 4) The safety circuit is not thermally independent of a prime source of degradation, namely the radiator. Most failures or degradations of the radiator would directly affect the power shunting capabilities of the safety circuit.
- 5) In cases of long term drift, the fraction of system up-time may be significantly affected. It is feasible that, with simple regulation capability and telemetry of system status, repair could be implemented with no system downtime.

Subtask 220-Quality Control

The objective of this subtask is to conduct a quality control program consistent with the objectives of the SNAP-23A program.

The Quality Control portion of the Phase I Product Assurance Program Plan (WANL-SS-003) was prepared and submitted to the A.E.C. for approval.

WANL reviewed and commented on the 3M "Quality Control Plan" MMM-3824-0001 in accordance with the requirements of "WANL/3M Interface Document" WANL-SS-001. In general, this plan complies with requirements. Referenced material was requested of 3M to assure complete understanding of the plan, and a meeting with 3M and WANL

Quality Control personnel has been scheduled to complete interfacing on the program plans.

A WANL Product Assurance review was made on the facilities at 3M to evaluate quality aspects in supplying thermoelectric converters for SNAP-23A. Manpower allocation, Quality Control, and Inspection facilities were found to be adequate for the proposed work.

WANL Quality Control reviewed the Hittman Associates responsibilities with respect to the fuel capsule contract with their Chief of Materials Fabrication. The need of specific information concerning design, welding and fabrication procedures, welding qualifications and related Quality Assurance and Inspection was discussed. As Hittman develops this information, it will be submitted to WANL for review, comment, and approval. Hittman is developing similar information in respect to ORNL loading, sealing, and inspection of the fuel capsule.

Quality Control, in conjunction with reliability has been working in close liaison with Design in preparation of SNAP-23A drawings. Most of the drawings released to date have been conceptual and study-layout drawings. Some twenty-five detail drawings are under preparation. Simultaneously with these drawing preparations, a Quality Control review is being made so comments may be incorporated as they develop. Specifications for fuel shielding and thermal insulation were reviewed and approved.

As design attributes are firmed up, action is being directed toward development of special fabrication, assembly, welding, and inspection procedures. It is planned that a dip-brazing-process will be used in fabricating the aluminum radiator. It will be necessary to qualify the brazing procedure and develop applicable inspection procedure. Brazing and inspection procedures have been reviewed with a potential vendor. Welding and inspection procedures for fabrication of the 718 Inconel jacket for the heat source assembly are being developed. Preliminary studies are under way for the development of detailed inspection instructions.

3. SYSTEM DESIGN

Subtask 310 - Systems Engineering

Objectives

The purpose of this task is to define the design parameters of the 60 watt(e) system and associated component requirements which will optimize the objectives of weight, size, lifetime, and power cost within the constraints imposed by safety, shipping and handling, maintenance, and reliability. In addition, this system must be applicable to a wide range of environmental conditions while exhibiting maximum commonality with the conceptual designs of the 25 and 100 watt(e) systems.

Summary of Results

During this reporting period, a digital computer code was developed for system analysis, and the optimum design of the 60 watt(e) system was established. The analysis resulted in the selection of an optimized 60 watt(e) system with a 24 inch outside diameter and a 4 inch diameter SrTiO_3 fuel cylinder. The overall height of the system is 26.8 inches; the radiator has 138 fins. Total system weight is estimated at 1115 pounds at a cost of \$49,490 or 9.42 \$/Kw(e) - hr. The off-design performance of this system was investigated over a range of ambient temperatures, and some overall system characteristics of the 25 watt(e) and 100 watt(e) systems were estimated.

Method

The analytical method used to optimize the 60 watt(e) system was a trade-off study of such factors as cost, weight and size, and commonality with the 25 watt(e) and 100 watt(e) systems. The results of this study permitted specification of such design parameters as system O.D., fuel pellet diameter, radiator fin number, insulation thicknesses, etc. The analysis was accomplished through development of a digital computer code incorporating detailed component parametrics of the shield, radiator, and T/E converter. A preliminary thermal analysis of the system was used in lieu of a more complete analysis which will be factored in during the next reporting period.

TABLE 3-1 SNAP 23-A T/E CONVERTER PARAMETRICS

Parameter	Configuration 1		2		3		4		5	
	BOL	EOL	BOL	EOL	BOL	EOL	BOL	EOL	BOL	EOL
Heat Input (w)	973.8	759.6	985.6	768.8	1013.5	790.6	1033.8	806.4	1068.4	833.4
Hot Junction Temp.	1075	890	1075	890	1075	890	1075	900	1075	900
Cold Junction Temp.	190	160	200	170	220	190	240	210	260	230
Number of Couples	263	263	266	266	272	272	276	276	283	283
Load Current	3.226	2.5	3.235	2.5	3.261	2.5	3.225	2.5	3.181	2.5
Load Voltage	24	24	24	24	24	24	24	24	24	24
Open Circuit Voltage	53.31	41.48	53.51	41.58	53.82	41.67	53.02	41.84	52.26	41.74

Description of SNAP-23A System Code

The detailed parametrics for the shield thickness (developed under subtask 370) and the radiator (developed under subtask 340) have been programmed into the code. The thermoelectric parametric data used in the code and supplied by 3M Company are summarized in Table 3-1. The analysis of these data is based on the average parameter description of thermoelectric devices to determine the characteristics of a single reference couple. Programming the characteristics into the code allowed the determination of varying numbers of couples operating temperatures, and power outputs as required by system constraints. The thermal performance of the system has been used for input to the code as extensions of the proposal analysis and will be updated when calculations are complete. Capsule wall thicknesses for 3 and 5 inch diameter fuel pellets were maintained constant at the values required to meet the sea water submergence criteria for the 4 inch diameter fuel pellet. The effects of this assumption on system trade-offs will be analyzed in the discussion of results.

The component costs were based on preliminary information and will be modified during the next report period.

Results

The first step in the optimization analysis of the 60 watt(e) system required the determination of the effects of variations in such parameters as fuel pellet diameter and system outside diameter using the detailed parametrics for the components. System weights and costs are shown in Figure 3-1 as functions of these parameters. The effect of the constant-capsule-wall-thickness assumption, noted in the previous section, is to accentuate the differences among the weight and cost curves for the 3, 4, and 5 inch diameter fuel pellets. Actually, if the wall thickness was allowed to increase with increasing pellet diameter, the 3 and 5 inch pellet diameter system weights and costs would be slightly closer to the 4 inch pellet diameter weights.

The minimum system weights occur at system O.D.'s of approximately 21 - 23 inches (depending on pellet diameter). As system O.D. increases from its value at minimum weight components such as the radiator, outside containers and insulation increase in weight,

resulting in an increase in overall system weight. As system O.D. decreases from its value at minimum weight, radial heat losses increase sharply (thinner insulation on sides) resulting in increased thermal input requirements to maintain 60 watt(e) power output. Thus, a taller fuel cylinder is required, and the result of an increased shield weight more than matches the decreased component weights at the smaller system O.D.'s.

The thermal power requirements are reflected in the system costs since the isotope represents a large contribution to overall cost. It is seen that the system costs decrease monotonically with increasing system O.D. The primary reason for this trend is the increase in radiator area at the larger system O.D.'s which effectively lowers the T/E cold junction temperature, thus increasing converter efficiency by permitting a larger ΔT at end-of-life conditions without exceeding the maximum hot junction temperature limit at beginning of life. The increased efficiency requires less isotope-thermal-input for the same EOL power output and thereby decreases the cost. Considering the weight and cost curves together, the conclusion is that a weight for cost trade-off will be advantageous; i.e., increasing weight from the minimum by increasing system O.D. will result in a cost savings.

It is noted on Figure 3-1 that the top insulation thickness is 0.5 inch for these designs as it was in the conceptual system design. Preliminary thermal analysis indicated that almost half of the total heat losses were out the top of the insulation just below the radiator assembly. Therefore, the top insulation thickness was increased to 1.0 inch in the subsequent optimization analyses in an attempt to decrease the heat losses from the top.

Figures 3-2, 3, and 4 show the system weights and costs for the 60 watt(e), 25 watt(e), and 100 watt(e) power output designs as functions of system O.D. and fuel pellet diameter. At this point, the 3 inch diameter pellet was eliminated from consideration since it obviously resulted in excessive weights and costs for the reference 60 watt(e) systems. Also, a multi-capsule design for the 100 watt(e) system (three - 3 inch SrTiO_3 fuel cylinders) resulted in a higher system weight and cost than the single 4 inch fuel cylinder. (The reason for the specification of the multi-capsule configuration for the 100 watt(e) system in the "Conceptual System Design Description" WANL-(SS)-002, was the limitation on fuel cylinder height which has been removed.) System characteristics for an alternate fuel form are presented in addition to the characteristics for the SrTiO_3 fuel form for

Figure 3-1. Preliminary SNAP-23A 60 Watt(e) System Weights and Costs

Figure 3-2. SNAP-23A 60 Watt(e) System Weights and Costs

Figure 3-3. SNAP 23A 25W(e) System Weights and Costs

Figure 3-4. SNAP 23A 100 W(e) System Weights and Costs

informational purposes. It is seen that considerable savings in both weight and cost are possible in the alternate fuel form systems, and investigations into the use of alternate fuel forms should be continued.

An engineering judgement was made in each of the three different power-output systems to determine the optimum system outside diameter. The system O. D. 's selected were 20 inches, 24 inches, and 28 inches for the 25 watt(e), 60 watt(e), and 100 watt(e) systems respectively. Based on the SrTiO_3 weights and costs, it was felt that the increased weight from the minimum was an equitable trade for the decreased cost which resulted as system O. D. increased to the selected optimum values.

Figure 3-5 shows the system weights at the optimum outside diameters as a function of electrical power output for the 4 and 5 inch fuel cylinder diameters. It is seen that the 4 inch diameter fuel pellet system weighs less than the 5 inch diameter fuel pellet system at 25 watts(e) output but has a higher weight at the 100 watts(e) output level. The reason for the dependence on fuel pellet diameter is presented in Figure 3-6. This figure shows the length-to-diameter ratio of the capsules versus fuel cylinder diameter for the various systems. The largest contributor to system weight is the U-8 w/o Mo shield (about 60 percent) which has minimum weight at a geometry consistent with an $L/D = 1.0$. It is seen that the L/D ratios for the 25 watt(e) systems are nearer to 1.0 for the 4 inch diameter pellet than for the 5 inch pellet, whereas for the 100 watt(e) systems, the L/D ratios are nearer to 1.0 for the 5 inch pellet systems. Again referring to the weight curves in Figure 3-5, it must be remembered that at 60 watts(e) power output, the actual weight of the 5 inch diameter pellet configuration should be slightly higher than shown (closer to the 4 inch diameter pellet). In fact, the weights of the 5 inch pellet diameter systems will be slightly higher at all EOL power outputs, thereby reducing the weight advantage at the 100 watt(e) output and increasing the disadvantage at the 25 watt(e) output level. On the basis of commonality for the three output power levels and on the basis that the manufacture of the 4 inch pellet capsule already requires an extension of present welding technology (5 inch pellet requires an even greater advance), the 4 inch diameter fuel pellet was selected as the common pellet diameter for the SNAP-23A systems. It should be noted that the previously mentioned advantages of the alternate fuel form can be fully realized with the 4 inch fuel pellet.

Figure 3-5. SNAP-23A System Weights Versus EOL Power Output

Figure 3-6. SNAP 23A L/D Ratio Versus Pellet Diameter

A summary of the operating and design characteristics for the optimum 60 watt(e), and conceptual 25 watt(e) and 100 watt(e) systems are presented in Table 3-2. Several additional comments are required to clarify the figures in Table 3-2. First, the ambient temperature was considered to be 70°F. Also, preliminary thermal analysis of the copper stud indicated that the 1100°F maximum allowable T/E hot junction temperature would not be exceeded if the average were 1085°F (up from the historic SNAP-23 value of 1075°F). The 70°F ambient and 1085°F average hot junction temperatures were used as boundary conditions in determining these designs. The radiator for the 60 watt(e) system has 138 fins at a thickness of 1/16 inch and weighs approximately 40 pounds. The number of couples in the converter at the nominal design conditions is 259, but a reduction of 25°F in BOL average hot junction temperature (to 1060°F) occurs when 266 couples are used, and this number of couples was judged to be best for the optimum 60 watt(e) system since it provides a design margin at the 1100°F maximum hot junction temperature limit. (The precise number of couples for the 60 watt(e) design will be mutually established by Westinghouse and 3M Company in the next report period.)

Selection of the 266 couple converter permits operation at an ambient temperature of 82°F before the maximum allowable hot junction temperature of 1085°F is reached. The performance of the optimized 60 watt(e) system design was evaluated over a range of ambient conditions at a constant load voltage of 24 volts. The thermal power input was maintained at approximately 1150 watts (due to fixed design geometry) as the ambient temperature was decreased, resulting in an increased electrical power output as the Carnot efficiency of the converter increased as shown in Figure 3-7. Increase in ambient temperature above 82°F requires a reduction in isotope loading to prevent exceeding the hot junction temperature limit at beginning of life. This action results in a lower electrical power output at end of life. For example, the power output at an ambient temperature of 160°F is only slightly more than 48 watts. This is not without compensation however, since the lower isotope loading results in a cost saving of \$2000 as shown in Figure 3-8. At ambient temperatures below 82°F, there is opportunity for selection of operating conditions of the unit. Either the additional EOL power output can be realized after decay from the initial loading of 1150 watts(th) (Figure 3-7) resulting in a lower power cost or 60 watts(e) power output can be obtained at EOL from a lower initial loading as

shown in Figure 3-8. For example, at an ambient temperature of -80°F , 60 watts(e) power output at EOL could be obtained at a cost of approximately \$2800 less than the cost of 60 watts(e) at nominal ambient conditions (70°F), thereby reducing the power cost from 9.42 \$/Kw(e)-hr. to 8.9 \$/Kw(e)-hr.

TABLE 3-2 SNAP-23A SYSTEMS CHARACTERISTICS

Operating Characteristics	Nominal Power Level					
	25W		60W		100W	
	BOL	EOL	BOL	EOL	BOL	EOL
Heat Source Thermal Power (w)	570	446	1150	893	1932	1508
Heat Losses (w)	129	105	148	120	164	134
Heat to T/E converter (w)	441	341	1002	773	1768	1374
Electrical Output (wP)	39.4	27.8	80	60	139	100
Heat Rejected by Radiator (w)	402	313	922	713	1629	1274
Overall Efficiency (%)	6.92	6.23	6.98	6.72	7.17	6.63
Thermal Efficiency (%)	77.5	76.3	87.5	86.6	91.4	91.2
Converter Efficiency (%)	8.93	8.17	7.98	7.76	7.84	7.27
Avg. T/E Hot Junction Temp ($^{\circ}\text{F}$)	1085	894	1085	896	1085	907
Avg. T/E Cold Junction Temp ($^{\circ}\text{F}$)	160	150	191	173	221	197
Load Voltage (v)	24	24	24	24	24	24
Load Current (amps)	1.48	1.04	3.33	2.5	5.78	4.17
Design Characteristics						
Overall Diameter (in.)		20		24		28
Overall Height (in.)		23		26.8		35.5
Design Life (yr)	5	10	5	10	5	10
System Weight (lb)	784	806	1056	1115	1500	1600
Fuel Cost (\$)	11,320	12,300	20,890	23,280	34,860	38,240
Capital Cost (\$)	21,560	21,810	25,650	26,210	29,060	35,000
Total Cost (\$)	32,880	34,110	46,540	49,490	63,920	73,240
Power Cost (\$/Kw(e)-hr)	30.0	15.6	17.6	9.42	14.6	8.36

25 W - 90% power conversion efficiency from 12v to 24v
 100 W - Two 50 W Converters in parallel

Figure 3-7. Off-Design Performance of Optimized 60W(e) Design

Figure 3-8. BOL Source Thermal Power and Cost Savings Versus Ambient Temperature

Subtask 315-Systems Engineering

The objective of this subtask is to assist Westinghouse in defining system criteria and interface specifications. During this reporting period the entire effort was directed toward development of the 60 watt(e) generator system. The major system efforts performed by Hittman Associates were concerned, for the most part, with the heat source design since it is one of the key components of the system.

A preliminary study was performed to establish heat source temperature distributions as a function of radial gap dimension and various accident and operational environments. Also, a preliminary study comparing the use of SrTiO_3 and an alternate compound was prepared as the SNAP-23A fuel form. A summary of the results of this study are presented below.

Thermal Analysis

1. Description of Method and Models

A thermal analysis was performed on the SNAP-23A thermoelectric generator utilizing the Thermal Analyzer Program (TAP-4). The objectives of this analysis were to define the capsule and shield temperature distribution as a function of radial gap dimensions and to evaluate the generator temperature distributions under various accident and operational environments.

The TAP-4 digital computer code which was used to perform the analyses is capable of solving complex, multi-dimensional, transient thermal problems. The code is general in construction and uses a network of conductors and capacitors to describe a heat-transfer nodal network whose transient properties are solved through simultaneous finite-difference equations. The steady state solutions performed in this analysis are a special case of the transient problems with the nodal capacities zero.

The thermal model used for this analysis simulates the entire SNAP-23A system with 71 nodal points and 133 nodal connectors.

The thermal model was set up with nodes concentrated in the capsule area, to assess the effect of radial gaps on the thermal profile. Figure 3-9a and b show the nodal network.

The following ground rules and/or assumptions were made in the set-up and use of the thermal model.

1) The capsule thermal inventory selected was 1168 watts. This corresponds to a 7.70 percent thermoelectric efficiency and an 86.0 percent thermal efficiency for the 60 w(e) system. No provision was made at this time for thermoelectric degradation inherent in long term operation of telluride thermopiles.

2) The dimensions and materials of the generator were based on the 60 w(e) design presented in the proposal with the capsule design modified to incorporate the dimensional changes associated with the stress analysis performed by Hittman Associates, Inc.

3) Radial gaps were assumed at all radial interfaces including the accumulator to thermopile surfaces. These gaps were assumed to be filled with helium at one atmosphere pressure. No axial gaps were assumed. Conduction and radiation in parallel were considered as mocking-up the helium gaps.

4) The ambient temperature was chosen at 80°F and the temperature drop between the outer thermopile surface and the thermoelectric cold junction was assumed to be 30°F.

5) The thermopile was simulated by (a) specifying a thermal admittance between the hot and cold junction nodes and (b) by supplying a heat input corresponding to the sum of Peltier heating (or cooling) and Joule heating (I^2R) at the hot and cold junction nodes. The thermoelectric parameters required were derived from hot junction heat balances using the 3M SNAP-23 performance data presented in Reference 1.

6) All known thermal losses were mocked-up. These are: insulation losses (top, bottom, and radial), thermopile seal losses, upper insulation seal losses, and support losses. It should be noted that the seal configuration and insulation thicknesses were based on the preliminary SNAP-23A design as specified in the proposal.

Figure 3-2a. Analytical Thermal Model Detail A.

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Table 3-3 summarizes the eight configurations and/or environments which were thermally analyzed.

Cases No. 1-3 represent the beginning-of-life normal operational case for radial gaps of 35, 10, and 60 mils, respectively. The four critical gaps which were varied are:

- 1) the fuel pellet to the inner Hastelloy-C liner,
- 2) the liner to the Hastelloy-C pressure capsule,
- 3) the capsule to the titanium barrier, and
- 4) the barrier to the inner shield surface.

Case No. 4 represents an accident condition which would lead to a generator open circuit situation. The heat balance at the hot junction of the thermopile is:

$$Q = NK(T_{HJ} - T_{CJ}) + N \alpha I T_{HJ} - \frac{1}{2} I^2 R_c$$

where:

- Q is the total heat withdrawn from the capsule
- N is the number of couples in the device
- K is the thermal conductance of a couple
- α is the Seebeck coefficient of the couple
- I is the current passing through the couple
- T_{HJ} is the hot junction temperature ($^{\circ}R$)
- T_{CJ} is the cold junction temperature
- R_c is the resistance of a couple

If current through the thermopile is accidentally lost ($I = 0$), the hot junction and heat source temperatures rise to reject the constant Q thermal load. This loss of current due to an open circuit accident eliminates Peltier cooling and subsequently causes overheating of the thermopile hot junction and heat source. An open circuit could be caused by a variety of accidents. Since the present 60 watt(e) thermopile is series connected, any one thermoelement that cracked

TABLE 3-3 THERMAL OPERATIONAL ENVIRONMENTS

Case No.	Beginning of Life or End of Life	Capsule Power (W)th	Normal or Open Circuit	Radial Gaps (mils)	Generator Configuration
1	BOL	1168	Normal	35	Normal Operation
2	BOL	1168	Normal	10	Normal Operation
3	BOL	1168	Normal	60	Normal Operation
4	BOL	1168	Open Circuit	35	Loss of Load
5	EOL	909	Normal	35	Normal Operation
6	BOL	1168	Open Circuit	35	Accidental Loss of Radiator Fins
7	BOL	1168	Open Circuit	35	Accidental Loss of Radiator Fins and Thermopile
8	BOL	1168	Open Circuit		Heat Source Operation in Air, No Fins, Thermopile or Insulation

or any junction shoe which disconnects from an element could lead to an open circuit condition. In addition, this condition could be caused by a faulty thermopile lead-through, a severed lead, or a failure or disconnection of the load.

Case number 5 represents the normal end of life temperature distribution. The minimum temperatures of the SNAP-23A U-8 Mo shield are of particular interest. This is true because the U-8 Mo alloy exhibits a deleterious time dependent phase change below 1065°F.

Case number 6 represents the accidental loss of the cold junction radiator. Loss of the radiator leads to a drastic reduction in cold side cooling which subsequently increases the thermopile, stud, and heat source temperatures. This condition could occur subsequent to 1) a fire accident, which would quickly melt the aluminum alloy fins or 2) a natural catastrophe such as an earthquake or landslide which could physically destroy the radiator fins.

It is highly probable that the thermopile would be inoperable after accidents of this type. Therefore, the open circuit case was assumed in conjunction with the loss of radiator.

Case number 7 simulates another accidental situation where it was postulated that both the radiator and the thermopile were accidentally removed. The probability of this accident occurring is even smaller than the removal of the fins alone. The relatively high melting point of the thermopile's stainless steel housing would preclude meltdown and removal subsequent to a transportation fire. Only a catastrophe such as an earthquake, landslide, or an exceedingly slow replacement of a degraded thermopile appear to be conceivable mechanisms leading to such a configuration. Although removal of the thermopile leads to a reduction in the available heat transfer area, compared to case 6 (fins only removed), elimination of the low conductivity thermopile is an overriding factor. As a result, the overall temperatures are lower for case 7.

Case number 8 was considered since the thermal profile is important in developing handling and shipping procedures for the heat source subsequent to fueling. This is particularly important since U-8 Mo reportedly exhibits temperature dependent oxidation. It should be noted that it was assumed, the heat source was sitting upright with no cooling at the bottom surface.

2. Results

Figures 3-10 to 17 present the heat source - stud assembly temperature profiles for the eight cases considered. It should be noted that the entire generator was simulated in cases 1 to 7. However, for the convenience of presentation, only the more important heat source-stud profiles are presented. Table 3-4 was prepared from the output data and compares several key temperatures for the eight cases.

Figures 3-10 to 12 present temperature distributions for cases 1 to 3 which show the effect of radial gaps on heat source temperatures. It should be noted that for the present SNAP-23A capsule design, radial gaps, which result from clearance and thermal expansion considerations, range from about 37 to 40 mils. The 35 mils per gap case, then, is

Figure 3-10. SNAP-23A Thermal Profile Heat Source Accumulator Assembly Case 1

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Figure 3-11. SNAP-23A Thermal Profile Heat Source Accumulator Assembly
Case 2

Figure 3-12. SNAP-23A Thermal Profile Heat Source Accumulator Assembly
Case 3

considered most representative of the present SNAP-23A design. As expected, all generator temperatures outside the outer capsule surface are approximately the same for these three cases. The effects of radial gaps are manifested only within the fuel capsule. As noted from Table 3-4, even the 60 mil gap case (case No. 3) does not lead to temperatures approaching the melting point of the fuel, Hastelloy-C or titanium. In addition, it appears that the maximum 1707°F fuel to Hastelloy-C interface is still below the 2000°F ORNL short term compatibility data. There is still some question, however, concerning the temperature at which titanium ceases to be an effective compatibility barrier between Hastelloy-C and U-8 Mo. However, this concern is minimized since the effect of the radial gap dimension on the maximum titanium temperature is quite small. The small titanium temperature variation is due to the fact that there is only one gap between the inner shield surface (whose temperature is effectively independent of gap dimension) and the titanium barrier.

Figure 3-18 illustrates the effect of the radial gaps on capsule temperatures. This figure can be used to estimate temperature differentials across the SNAP-23A capsule's helium gaps. For example, the case where the four radial gaps are 40 mils each, from Figure 3-18, the average ΔT per mil is about 1.8. Since the temperature drop across the clad metal is negligible, this 1.8°F/mil can be used to estimate temperature drops across various clads. As a rule of thumb, the ΔT from the outer pellet surface to the inner shield surface can be calculated as follows:

$$\Delta T = 40 \text{ mils/gap} \times 4 \text{ gaps} \times 1.8^\circ\text{F/mil} = 288^\circ\text{F}$$

Since the shield temperatures are effectively independent of gap size, the actual temperatures are effectively independent of gap size, the actual temperature of any of the clads can also be estimated by using Figure 3-18. Using the same example cited previously, the maximum outer pellet surface temperature (fuel to Hastelloy-C interface) can be estimated by adding the 288°F ΔT calculated previously to the maximum shield temperature of approximately 1336°F. This gives a maximum outer pellet temperature of 1624°F. Similar calculations can be used to estimate maximum temperatures of the Hastelloy liner, capsule, or titanium barrier. It should

TABLE 3-4 SNAP-23A THERMAL RESULTS (Temperature in °F)

Case No.	Max. Fuel Temp.	Fuel-Hast-C Interface Temp.	Max. H/C Temp.	Max. Ti Temp.	Max. U-8 Mo Temp.	Min. U-8 Mo Temp.	Max Cu Temp.	Max.SS Temp.	η, % Therm.eff.
	M.P. =3470	Compat.Data O.K.at 2000°F	M.P. =2310-2450	M.P. = 3034	M.P. =2100	Phase Change 1065°F	M.P. = 1980 Cu-U eutectic 1742	M.P. = 2700	
1	1737	1596	1516	1389	1336	1212	1318	1316	84.3
2	1577	1432	1406	1363	1340	1212	1317	1315	84.3
3	1842	1707	1589	1404	1333	1212	1319	1316	84.3
4	1918	1778	1698	1570	1517	1400	1499	1496	76.3
5	1480	1370	1308	1208	1155	1068	1163	1152	87.9
6	2354	2210	2160	2062	2030	1917	2007	2003	N.A.
7	2050	1906	1856	1759	1727	1607	1705	1701	N.A.
8	1238	804	723	592	539	487	503	502	N.A.

be noted that Figure 3-18 pertains only to temperature drops across the mid-plane of the capsule which represents the plane of maximum clad temperatures during normal BOL operation.

The open circuit case, No. 4 (Figure 3-13), leads to stud and heat source temperatures which are approximately 180°F higher than the analogous normal operational case (No. 1). Again, fuel, Hastelloy-C, and titanium temperatures are well below their melting points. The maximum Hastelloy-C fuel-interface temperature is 1778°F which is below the 2000°F ORNL preliminary fuel-Hastelloy-C compatibility data. The titanium barrier, however, operates at a maximum temperature of 1570°F which could compromise the effectiveness of this material as a compatibility barrier over long periods of operation. This potential problem is being investigated.

Figure 3-14 depicts the end-of-life (EOL) temperature distribution. As expected, all heat source and stud temperatures are considerably lower than the comparable beginning of life temperatures (case No. 1). The maximum EOL fuel temperature, for example, is 257°F lower than the BOL case. The most significant result is the minimum shield temperature of about 1068°F. This temperature is very close to the 1065°F transformation temperature of the metastable α to the $\alpha + \gamma'$ phase, for the U-8 Mo shield alloy. This means that only a small portion of the biological shield would transform to the $\alpha + \gamma'$ phase. The phase change is undesirable since there is a volume shrinkage associated with this transformation.

Case number 6, as illustrated in Figure 3-15, represents an accident situation where the radiator is destroyed. Note that this postulated accident leads to stud and heat source temperatures which are considerably higher than any of the other seven cases considered. In fact, from Table 3-4, it can be seen that this situation leads to a maximum copper temperature which exceeds the melting point of copper and is above the Cu-U eutectic point. In addition, the maximum shield temperature of 2030°F is close to the 2062°F U-8 Mo melting temperature. This appears to be the most serious accident situation since the integrity of the shield could be compromised. Further study, including transient analyses, will be performed to establish the heat-up time period.

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1 TEMPERATURES IN °F
NOTES

DATE	QTY	PART NUMBER	TEMP	DESCRIPTION	REV
REV D	REV D	REV D	REV D	REV D	REV D
SNAP 23A THERMAL PROFILE HEAT SOURCE ACCUMULATOR ASSY CASE 5 EOL CONDITION 35 MIL RAD AL GAPS					
NEXT ASSY		DRAWN		HITTMAN ASSOCIATES, INC.	
USED ON		CHECKED		BY	
APPROVED		APPROVED		DATE	
DRAWN		DATE		DISK 578	

Figure 3-14. SNAP-23A Thermal Profile Heat Source Accumulator Assembly
Case 5

Figure 3-15. SNAP-23A Thermal Profile Heat Source Accumulator Assembly Case 6

The generator steady state temperature profile after removal of both the radiator and the thermopile is depicted by Figure 3-16. Although the temperatures are considerably higher than the normal operational case, they are considerably lower than case 6 where only the radiator is removed. Since the maximum copper temperature is below the copper melting point and the Cu-U eutectic temperature, it does not appear that the integrity of the shield is jeopardized.

Figure 3-17 shows the temperature distribution of the heat source alone, in air. The maximum surface temperature occurs at the bottom of the heat source, node 4, and is 503°F. Although case 8 included the copper accumulator stud and stainless clad surrounding the heat source, the temperatures of the capsule-shield assembly alone in air would be quite close to this model. Since oxidation of the U-8 Mo shield subsequent to fueling could be a potential problem, it is important to note that the maximum shield temperature for this case is 592°F.

Preliminary Comparison of SrTiO_3 Versus an Alternate Fuel Form

The objective of this study was to compare the effectiveness of SrTiO_3 as the SNAP-23A fuel form. The comparison was based upon system performance, nuclear safety, and fabricative considerations. The following items were considered:

- 1) Systems considerations
- 2) Compatibility properties
- 3) Safety of the fuel relative to land release
- 4) Safety of the fuel relative to seawater release
- 5) Fabricative possibilities of the heat source

Due to the higher power density of the alternate form as compared to SrTiO_3 , a system weight savings of about 150 pounds is realized by use of the alternate form.

In comparing the safety problems for the two fuels, it was concluded that release of either form in a land or sea environment has a low probability. In the event of such a release, however, hazards could be unacceptable for either fuel.

Figure 3-16. SNAP-23A Thermal Profile Heat Source Accumulator Assembly
Case 7

Figure 3-17. SNAP-23A Thermal Profile Heat Source Accumulator Assembly
Case 8

Figure 3-18. Effect of Radial Gaps on SNAP-23A Midplane Capsule Temperatures

Subtask 320 - Mechanical Design

Radiator Subassembly

Design studies to investigate and establish the design features of the radiator subassembly were initiated in February and continued until the third week of April 1967.

On the basis of thermal analysis, the decision was made to use a finned radiator design. Little, if any, theoretical improvements would result by use of the pin type radiator as compared with the fin type. Potential disadvantages of the pin type radiator are the increased design complexity and the attendant manufacturing problems. It was also expected that the pin type radiator design would be somewhat inferior with regard to withstanding the required shock and vibration environment.

Thermal analysis inputs to the design imposed the requirement of a 50 psi minimum contact pressure at the interface to ensure adequate conductive heat transfer. The maximum allowable interface pressure was established to be 150 psi by the 3M Company. This restriction was imposed to prevent binding of the converter cold frame push rods during system operation, due to excessive hoop compression in the cold frame.

An investigation was made to determine the best method of manufacturing the aluminum radiator subassembly. Of the various ways considered, it was concluded that dip brazing is the preferred method. Early in the investigation, it was determined that the radiator assembly can best be manufactured as a sheet metal fabrication due to the required configuration. The fin spacing, thickness, and depth all but preclude machining the configuration from solid stock. Extruding the finned assembly was not investigated since this method would have application only to volume production. Brazing and welding, on the other hand, have potential both for one or two units or volume production.

With regard to the possibility of welding it was concluded that full-length, filleted weld joints of the fins to adjacent members would be required to insure minimum impedance to heat transfer, to provide maximum structural integrity, and to seal the joint to prevent crevices which could lead to ultimate corrosion damage. Due to the close spacing of the

fins at the inner hub, however, much difficulty would be expected in achieving full penetration weld joints without excessive thermal distortions because each weld would essentially overlap the next. Local welding along the joint length, analogous to tack welding, could be accomplished without severe distortion but would produce poor heat transfer characteristics at the joint, further magnified by possible corrosion due to the lack of sealing. Because access to the fins-to-hub joint is limited, it is quite difficult to conceive a practical joint design which would permit a fully sealed joint.

Dip brazing, on the other hand, appeared to be quite promising; initially, since this process has been widely and successfully used in the fabrication of such devices as refrigerators, heat exchangers, fuel tanks, aircraft parts and similar products. Characteristics of the dip-brazing process considered desirable with respect to the manufacture of the SNAP-23A radiator assembly are as follows:

- 1) Many joints can be made simultaneously with one overall application of heat, thus reducing thermal distortion tendencies.
- 2) Dip-brazed fillet joints perform well under repetitive stress and vibration conditions and provide efficient heat transfer characteristics through the joints.
- 3) Thin sections are readily brazed without distortion even when joined to relatively thick sections with no sacrifice in joint efficiency or resistance to corrosion.
- 4) Self-fixturing is possible and desirable to reduce fixturing costs and ensure proper contact of component parts.
- 5) Dip-brazing can be accomplished with a variety of aluminum alloys both of the non-heat treatable and heat treatable alloys.

Fluxes used in brazing aluminum alloys contain chlorides and fluorides which can cause corrosion if allowed to remain on the parts. However, successful cleaning procedures have been developed to clean parts after brazing. Entrapped flux in the braze joint itself is also a potential corrosion hazard; however, proper design with brazing in mind, in conjunction with careful control in the preparation of the parts and of the brazing process itself, will avoid flux entrapment problems.

From the standpoint of design simplicity and ease of manufacture, the preferred radiator geometry would be radial fins attached to a continuous unbroken inner hub and outer shroud. Installation of the radiator to the converter in this case would entail such methods as shrink fitting or tapering the converter interface.

Other design schemes were also evaluated, such as providing a split radiator assembly which would be clamped to the thermoelectric converter. Although various configurations and approaches were considered in the radiator attachment studies, all versions fall into one of the following three basic categories:

- 1) Solid inner hub with a tapered converter interface.
- 2) Solid inner hub with a cylindrical converter interface.
- 3) Split inner hub with a cylindrical converter interface.

In the solid inner hub with tapered interface configuration, shown in Figure 3-19 the radiator would be mounted to the thermoelectric unit before assembly to the heat source subassembly. Interface pressure on the taper surface is provided by means of axial preloading of the tie bolts. Reverse acting removal bolts serve as axial stops to ensure that additional radial forces are not built up during exposure to the shock and vibration environment and to provide the reverse force for radiator removal. In this design the tendency towards taper mismatch is overcome by designing the inner hub to yield at assembly, assuring conformity of adjacent surfaces. Thermal grease would be applied at the interface to reduce interface friction and fill the surface finish microvoids, thus improving thermal conductivity. Low friction is advantageous both at assembly to better relate axial bolt torque to interface pressure, and during disassembly to reduce binding tendencies.

To accommodate a tapered radiator hub a tapered and thickened thermoelectric converter outer shell would be required. This modification would be necessary to avoid changing the thermoelectric converter cold frame which has a cylindrical outer diameter. To prevent excessive radial pressure on the thermoelectric converter cold frame due to the shrinking-on of a thicker outer shell, it would become necessary to provide axial grooving through the taper to reduce the shell stiffness. The increased radial thickness of the thermoelectric converter outer shell in combination with the reduction of surface contact due to

Figure 3-19. Tapered Interface Radiator Configuration

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the grooves caused a significant reduction in thermal performance.

Some consideration was given to a solid hub, cylindrical interface version. Assembly of the radiator to the thermoelectric converter in this case could be performed by shrink fitting or brazing. However, such an arrangement would have the disadvantage that disassembly and reuse of the radiator would not be possible.

Two cylindrical interface split hub schemes which were studied in some detail are depicted in Figures 3-20 and 21. The version depicted in Figure 3-20 appeared initially favorable because it permitted assembly of the radiator to the converter without cutouts in some of the radiator fins for wrench clearance. Interface pressure is developed in this scheme by the drawing-together of the axial tie rod bolts. Analysis of this version, however, indicated a tendency to provide a non-uniform pressure profile in the axial direction peaking at the location of the taper rings.

The split hub version with cylindrical interface as depicted in Figure 3-21 was ultimately selected as being the most favorable arrangement. This approach appears to be superior from the thermal performance standpoint and presents no major difficulties from the design or manufacturing standpoints. Cylindrical surfaces are, in general, easier to machine and dimensional tolerances can be better controlled.

In the design selected, a relatively thin inner hub is used tending to maximize conformity of surfaces. Bolted flanges are used to provide hoop loading. The usual tapered transition from flange to band (hub) is deliberately deleted in this design to better ensure uniformity of contact pressure. Thermal grease will be provided at the interface to improve the thermal conductivity both by filling surface finish microvoids and by assuring a more uniform pressure profile in the axial direction in the vicinity of the flanges by providing adequate flange cross section to preclude bending of the flanges. Local high pressures between the radiator hub flanges and the thermoelectric unit are avoided by relieving the flange surfaces in contact with the thermoelectric unit. Interface pressures of 50 to 100 psi will be provided by properly torquing the flange bolts at assembly.

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Figure 3-20. Cylindrical Interface Radiator Configuration

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Figure 3-21. Radiator Design

Thermoelectric Converter-Heat Source Interface

A study of the thermoelectric converter-heat source interface was initiated in April to verify the design adequacy, and if necessary, to redesign adjacent members at this interface. It was determined that no major changes from the proposal concept would be required. The hot frame interface dimensions have now been fully established with the 3M Company. The exact dimensions of the heat source subassembly neck region which interfaces with the hot frame of the converter are dependent upon the thermal expansion characteristics of these members as well as the nature and thickness of the interface sleeve material used. The thermal expansion study is currently in progress.

Two requirements must be met with regard to the thermoelectric converter-heat source interface:

- 1) Heat from the heat source assembly must be efficiently transmitted across the interface.
- 2) The interface design must accommodate field replaceability of the thermoelectric unit.

The tapered joint design depicted in Figure 3-22 will be provided to facilitate ease of field assembly and assure adequate contact pressure by application of a single axial bolt. A Belleville washer will be located under the bolt head to take up relative movement due to long-term creep of the materials in contact. To provide assurance that the thermoelectric unit may be removed from the heat source and to obtain a reasonably uniform interface pressure with the expected taper tolerances, it was concluded that a compressible sleeve must be provided at the interface.

Mica will be used as the interface sleeve material. A mica thickness of 0.005 inch was selected as the best compromise between providing adequate material thickness to insure conformity of tapered surfaces while holding the conductive heat losses through the material to an acceptable level. Synthetic fluor-phlogophite, a mica product, will be specified for use in this application because it is capable of operating at temperatures in excess of 1400°F.

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Figure 3-22. Thermoelectric Converter Attachment to Heat Source Subassembly

Heat Source Subassembly

Work was initiated the early part of March to investigate and upgrade, as necessary, the heat source subassembly conceptual design. During this phase, a better definition of the various components of the heat source subassembly was established and the materials for the shield and heat accumulator determined. The scheduled release date of all heat source subassembly components is May 31, 1967; this includes the radiation shield, accumulator and heat source jacket.

1. Radiation Shield

The use of uranium alloys for the shield material is desirable from the standpoint of producing a minimum weight design. However, certain problems associated with these materials must be considered. For instance, uranium and its alloys have relatively poor oxidation resistance. Also, pure uranium undergoes phase transformations which could result in significant dimensional instability. It has been found that alloying uranium with molybdenum will greatly minimize the dimensional instability problems, but will cause a certain loss of ductility and some tendency toward stress corrosion problems. Uranium-8 w/o molybdenum has been selected as the shield material; it represents a compromise, reducing the dimensional instability problem to an acceptable level while still maintaining adequate ductility.

To overcome the potential problems of oxidation and stress corrosion, the basic approach taken in the design and assembly of the heat source subassembly was to avoid heating in the presence of oxygen. To achieve this end, component parts are so designed as to permit a maximum amount of cold assembly to be accomplished prior to the insertion of the radioisotope heat source. Thus, at the time of fuel capsule loading, the shield will be totally encapsulated by the heat accumulator and outer jacket with the exception of the access hole for fuel capsule insertion. Sealing is essentially accomplished upon assembly of the shield plug, titanium plate, end cap after insertion of the fuel capsule. The end cap design will permit a relatively high degree of mechanical sealing prior to seal welding. Full hermetic sealing will be accomplished upon completion of the end cap welding operation.

The U-8 w/o Mo would be cast and cored roughly to the required shape and finally machined to fairly close tolerances. Availability, manufacture, and cost have been investigated. Delivery of a finished shield can be expected 12 to 14 weeks after receipt of order at a cost per unit of approximately \$5500.

2. Accumulator

The heat accumulator which surrounds the radiator shield, must have high thermal conductivity and be capable of transferring heat from the fuel capsule to the thermoelectric converter. To meet this requirement, four grades of high purity commercial copper were investigated, oxygen-free high conductivity (OFHC) copper, tough pitch copper, silver bearing lake copper, and phosphorus deoxidized copper. Of these, OFHC copper was found to be the most desirable. All the others were lower in thermal conductivity and would require more volume to provide the same effectiveness as OFHC.

Although some degree of fusion is expected to take place between the shield and accumulator, it is not expected to affect the conductivity of the copper. Under operating conditions essentially, no structural strength or stability is expected from the accumulator as it will assume a plastic form. Fairly close tolerances must be maintained between matching components to minimize the flow of the copper into voids at operating temperature.

Studies regarding the method of manufacturing the accumulator were evaluated to determine availability and economics. Three methods were considered: 1) to spin the body from pre-machined plate, and machine to final dimensions; 2) to final machine from a forged shape; or 3) to machine direct from a forged billet. A decision to machine from a forged billet was made after availability of material and manufacturing costs were evaluated. The lower portion of the accumulator will be spun from plate. Completion of final machining of all accumulator parts can be expected 10 to 12 weeks after vendor receipt of order. Total unit cost for an accumulator should be approximately \$3800.

3. Heat Source Jacket

The heat source jacket completely encloses and hermetically seals the heat accumulator and shield to prevent oxidation and contamination of these components. It must also be designed to assure adequate structural integrity of the heat source assembly and permit low thermal impedances between internal components. Important considerations in the selection of the jacket material are adequate structural capability up to temperatures of 1400°F, and a coefficient of thermal expansion equal to or lower than that of the shield. The latter is of importance from the standpoint of obtaining good thermal contact between the copper heat accumulator and jacket at the neck portion of the heat source subassembly and between the accumulator and shield.

Materials investigated were Inconel 750, Inconel 718 and Hastelloy C. Stainless steel (300 series) was originally considered and found to be unsatisfactory from the standpoint of desired thermal expansion characteristics because it expands at a slightly greater rate than does the shield. Inconel 718 was selected as the heat source subassembly jacket material.

Manufacturing of the heat source jacket has been investigated and a combination of spinning, welding, and machining will be used in fabrication. The finished heat source jacket could be expected to be completed 10 to 12 weeks after vendor receipt of order at an approximate unit cost of \$3400.

The final machining of both accumulator and heat source jacket would be accomplished by one vendor with facilities and equipment capable of finishing both components to the close tolerances required.

4. Assembly Procedure

Figure 3-23 indicates the assembly of the components. The shield is first located inside the accumulator top, after which the accumulator bottom is fitted over the shield. The heat source jacket bottom is then positioned to hold the two portions of the accumulator in contact, and a weld is made to connect the bottom and top portions of the jacket.

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Figure 3-23. SNAP-23A 60 Watt (e) Heat Source Cold Assembly

For final assembly the major heat source components will be shipped to Oak Ridge where the fuel capsule, shield plug, plug bolts, (Figure 3-24) titanium plate, and jacket closure are inserted and the jacket closure welded. Purging will then be accomplished, followed by backfilling with helium gas through the purge tube provided for the purpose. The heat source subassembly will thus be designed to permit a maximum amount of cold assembly with no radiological exposure.

Heat Source Support System

Conceptual support system studies were initiated during the first week of March and continued to the end of April. During this period, consideration was given to both the method of attaching support members between inner and outer insulation cans and the method of direct support of the heat source subassembly eliminating the inner insulation can. Both tie rod and truss support schemes were studied with regard to use of the inner insulation can. Only the truss schemes were considered however, with regard to the direct support method. As a result of these studies, a decision was made to use the truss support method as opposed to the tie rod scheme. A final truss system design was started the last week of April. Detail design drawings are scheduled for release on May 31, 1967.

The direct support method entailed the attachment of truss support members directly to the heat source jacket. Although this scheme eliminated the insulation can and reduced the total weight of the unit, severe assembly problems existed. The heat source jacket and support system would be at elevated temperatures during installation within the outer container. This would also necessitate assembling and fitting the insulation around the thermally hot heat source subassembly. For these reasons the support method using an intermediate inner insulation can was selected. In this method, the mechanical support system will be located within the insulation subassembly, which can be fabricated prior to the heat source installation.

In the tie rod support scheme as depicted in Figure 3-25 two sets of pretensioned rods would be provided, one set acting in the upward direction, the other set in the downward direction. Side load capability would be provided by angling both sets of tie rods. However,

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Figure 3-24. SNAP-23A 60 Watt (e) Heat Source Final Assembly

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Figure 3-25. Tie Rod Support

during operation these tie rods are highly stressed and are exposed to high temperatures for a long period of time under load.

Potential advantages of the truss support system (Figure 3-26) as compared to the tie rod support scheme (Figure 3-25) are: relative ease of assembly (with insulation); elimination of long term creep considerations of the support structure material; and inherent adaptability to differential thermal expansion conditions between the inner and outer can of the insulation subassembly. Designing to a given spring constant for either the tie rod support system or the truss support system results in approximately the same total cross-sectional area and length of path of support member and, hence, quite similar heat losses.

A detailed structural analysis was made for 6-leg and 8-leg truss system configurations. The analysis performed for the 6-leg truss, having a cross-sectional area of 0.160 square inches per leg, indicates an axial natural frequency of 100 cps and lateral frequencies of 70 to 75 cps. Maximum G levels of the heat source under the design vibration conditions would be in the range of 7 to 10 G's, with resulting truss stresses ranging between 8,000 to 24,000 psi. Analysis of the 8-leg truss with the same cross-sectional area indicates an increase in natural frequencies in the lateral direction to 80 cps. Also, the 8-leg truss permits a reduction in the size of the reinforcing ring required on the insulation can.

Insulation Subassembly

Work was in progress throughout May which will result in the release of all required insulation detail drawings by May 31, 1967.

In the course of the original insulation investigation, it was found that no single insulator possesses all of the design characteristics desired; therefore, the final selection is ultimately a compromise. Of the materials considered, the Min-K series of insulations exhibits the most favorable combination of properties. These Johns-Manville products have been used in most radioisotope power systems designed and built to date. Of this series of insulations,

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Figure 3-26. Truss Support

Min-K 2000 has been initially selected as most promising, although other possibly superior grades of Min-K which are in the process of development, will be given consideration at a later date. In view of the range of service temperatures anticipated for the insulation in this design, Min-K 2000 exhibits satisfactory thermal conductivity, dimensional and chemical stability, and low degradation for long term operation. In addition, the thermal conductivity of this insulation may be reduced even further by back-filling with an inert gas such as Krypton.

Two methods of installing the heat source insulation in the heat source container, see Figure 3-27 have been considered. One method is to fabricate insulation components from sheet material and stack them in place; the other is to mold the Min-K insulation in place into the annular cavity between the inner and outer insulation cans and around the mechanical support system. The insulation for the first mockup will probably be machined from sheet material. The insulation will be fabricated from these sheets into component parts, in sections so they can be assembled into the insulation container cavity. By using this approach, the components could be formed by rough cutting the sheets, clamping, and machining to final dimensions. After the sheets are installed in the insulation container, powdered insulation material will be used to fill the voids. Then the insulation container will be sealed, purged, and back-filled with Krypton gas. Prior to the installation of the Min-K 2000 into the insulation can, an out-gassing operation will be performed to remove organic binders which would otherwise evolve gases at high temperature.

If the insulation were molded in place, the organic binder and hydrated asbestos fibers, normally used in Min-K 2000, would be omitted and glass or quartz fiber substituted in their place. The physical properties, i.e., density, thermal conductivity, liner shrinkage, and structural strength, would be comparable with the values for the block insulation. Although molding the insulation in place appears promising, further investigation is required to fully establish the feasibility of this approach.

Figure 3-27. SNAP-23A 60 Watt (e) Insulation and Insulation Container Assembly

Subtask 330 - Stress Analysis

Shock and Vibration Analysis

The assembled SNAP-23A unit must be capable of withstanding four different shock and vibration tests from MIL-STD-810A. These tests specified in the RFP are summarized below:

1) Method 514.1 Equipment Class 6 - The assembly is to be subjected to a sinusoidal resonance and cycling test along each axis. The excitation is to vary from 1.3 to 5 G's at frequencies between 5 and 50 cps.

2) Method 516.1 Procedure I. - The assembly is to be subjected to 3 shocks along each of 3 perpendicular axes. All shocks are to be of triangular shape rising to 5 G's in 10 milliseconds (MIL-STD-810A modified per RFP).

3) Method 516.1 Procedure III -d - The assembly is to be subjected to an impact test applied once to each end and side. The velocity at impact with a rigid surface is to be 7 feet per second.

4) Method 516.1 Procedures III - b and c - The assembly is to be subjected to an edgewise drop test once on each end and side. One edge of the base is to be supported on a sill while the opposite edge is permitted to drop 24 inches.

Specification MIL-STD-810A indicates that the assembly may be packaged for shipment during tests 1, 3, and 4. In test 2 the assembly is to be tested in the manner that it will be operated, including vibration isolators if the assembly is normally mounted on such isolators.

It was found that the heat source support structure could be sized to withstand tests 1 and 2 without excessive heat loss. However, none of the support systems studied could withstand tests 3 and 4 unless the assembly was supported on shock mounts.

The support system selected uses an 8-leg truss. The characteristics of the system are as follows:

Total Area	.96 in. ²
Length	12.1 in.
Spring Constants	992,000 No. 16/in. axially 468,000 No. 16/in. laterally
Natural Frequencies	107 cps axially 75 laterally

During test 1, resonance will not occur along any axis. About 9 G's laterally and 6.5 G's axially will occur at both heat source and converter at 50 cps. The stresses in the truss legs will be limited to 23,500 psi (lateral vibration) and 8,100 psi (axial vibration). The yield strength of the truss material, Inconel 718, at operating temperature conditions is over 100,000 psi. During test 2 both the heat source and converter will be subjected to 10 G's. The truss stresses will be 25,000 psi (lateral shock) and 12,500 psi (axial shock).

Shock mounting of the assembly in the shipping container will be required for tests 3 and 4. Without shock mounts, the G loads in the support system selected would exceed 100 G's during test 3, and the combined rotational and translational effects in test 4 would be equivalent to over 200 G's translational. Soft shock mounts can reduce the loadings in tests 3 and 4 to levels equal to or less than those of tests 1 and 2.

In addition to the above requirements, the various subassemblies are subject to a bench handling test per MIL-STD-810A, Method 516.1 Procedure VI. Spring bumpers can be attached to the radiator when the radiator and converter subassembly is handled as a package. These bumpers can permit the subassembly to withstand this test and limit the accelerations of the converter and radiator to 10 G's in any direction.

Heat Source Support Structure

The maximum stress in the truss leg is 25,000 psi under a lateral 10 G shock (see Section 1). The load is 3,000 pounds at this condition. The buckling load of a solid rod with pinned ends would be 1,930 pounds (end bracket design is simplified if the leg is stable without end support). If the moment of inertia of the leg is increased to provide a safety factor of about 2 against buckling, a tube of at least 0.560 inch outside diameter should be used. In addition a superalloy material should be used (such as Inconel 600 as planned) since the strengths of the common stainless steel are not adequate at operating temperatures.

The maximum stress in the ring of the inner shell of the insulation assembly is less than 10,000 psi. A material such as 316 stainless steel is adequate in this area. The maximum stress

in the outer ring at the bottom of the outer shell is about 32,000 psi. Again 316 stainless steel is adequate since the temperature in this area is only slightly above ambient.

Radiator

The radiator hub and the radiator support are designed for heat transfer. To obtain good heat transfer, a contact pressure between the converter and radiator hub of at least 50 psi is required. However, the converter is limited structurally to an average of not more than 150 psi. A structural analysis of the hub for an appropriate clamping force was performed, and the corresponding pressure distribution was also obtained. Figure 3-28 shows the results. It is noted that, locally, high contact pressure at the base of the flange could exist; however, this result was obtained with the assumption that the converter is perfectly rigid. It should be noted that the high contact pressure can be prevented by limiting the initial flange gap to 0.010 inch or less. The effect of friction on this pressure distribution is indicated by Figure 3-29. With a coefficient of friction of 0.200 the contact pressure at 90 degrees, indicated in Figure 3-28 would be reduced by 35 psi.

The stresses obtained by the above analysis are shown in Figure 3-30. The bolt load was assumed uniformly distributed along the length of the hub and the effect of stress risers in the hub is not included in Figure 3-30. The effect of the concentrated bolt load on the stress in the vicinity of the flange is shown in Figure 3-31. This shows that the nominal 6100 psi stress in the vicinity of the first radiator fin grooves is increased by a factor of 4. The minimum radius at the base of the grooves was chosen to give a stress concentration not greater than 3. The resultant stress is well above yield; however, we see from Figure 3-31 that the stress drops 50 percent, one inch from the flange, so that no problem exists if the material is not notch-sensitive. The hub material is 6061-T4 aluminum which, according to Alcoa Technical Paper No. 18, has a very low notch sensitivity. It will tolerate a crack length of 2.3 inches at the yield stress without propagation. This also eliminates considering the possibility of an initial built-in-crack produced by a brazing flaw between the hub and the fins. Conservatively, a maximum acceptable braze-flaw size of 1/4 inch could be specified.

Figure 3-28. Radiator Hub Pressure Distribution (1500 lb/in. Bolt Load, No Friction)

Figure 3-29. Effects of Friction on Pressure Distribution

Figure 3-30. Typical Radiation Hub Stresses (1500 lb/in. Bolt Load)

Figure 3-31. Stress Profile Across Width of Hub Due to Bolt Eccentricity

The final area investigated is the hub to converter interface including the edge relief as pictured in Figures 3-32 and 3-33. Figure 3-32 shows the effect of the converter stiffness on the hub stress profile across the width at the hub. Note that this figure indicates that the high edge stress due to the concentrated bolt load indicated in Figure 3-31 is substantially reduced. Figure 3-33 shows that, theoretically, the overhang of the hub beyond the cold frame has a bad effect on the pressure distribution across the hub width. The tension in the overhang causes the edge of the band relief to dig in. This condition will be alleviated by providing a radius on the underside of the band edge. However, a locally concentrated load is not serious, even if it should occur, since the resulting converter stresses and deflections are not significant.

Figure 3-32. Stress Profile Across Width of Hub Due to Edge Relief

Figure 3-33. Pressure Distribution Across Width of Hub Due to Edge Relief

Subtask 340 - Thermal Analysis

Thermal analysis of the overall SNAP-23A system has been performed treating each major component individually. For this analysis, the system has been divided into three major components: the heat source, the thermoelectric converter, and the radiator. In the consideration of the major components, particular attention has been given to the thermal interfaces between the components and the instrumentation necessary for evaluation of component performance, both individually and when mated together.

All techniques used for the thermal description and analysis of the components and interfaces have been derived and adapted from applicable technology through careful literature studies. The only exception is the thermoelectric performance of the 3M converter which is analyzed by techniques provided by 3M (described in Subtask 350). The techniques analytically describe the temperature distributions and heat fluxes within the system components and are used in the design and development of the components for the 60-watt (e) system. After manufacture of the test system mock-up, the thermal analysis will be used in the evaluation of the system component test results.

After the thermal analysis has been correlated with the experiments, the analysis techniques will be used to study design changes and to recommend improvements to the 60-watt (e) system. The correlated analytical model will then be used for the conceptual design of the 25 and 100-watt (e) systems.

To accomplish the objectives of this subtask, the schedule shown in Figure 3-34 has been adopted and describes the major areas of effort.

Radiator Study

A model for predicting the heat rejection rate of a plate fin radiator was formulated and developed into a digital computer program. A parametric study of the effects of geometry variation on the heat dissipation capability was completed, and a fin configuration selected for the 60-watt (e) system. The effect of fin material variation was studied and an analysis of the power conditioning thermal rejection area requirements performed.

KEY: PROPOSED OR ORIGINAL

ACTUAL

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Figure 3-34. Thermal Analysis Schedule

The model for predicting the heat rejection rate of the plate fin radiator is based upon an analysis of one-dimensional heat conduction in a rectangular fin with heat transfer by natural convection to air. The basic shape of the fin for the model is shown in Figure 3-35. Heat transfer coefficients are evaluated by a correlation of Nu (Nusselt number) versus $GrPr$ (product by Grashof and Prandtl numbers) for laminar free convection heat transfer between vertical parallel plates developed by Elenbaas². The radial temperature profile in the fins is accounted for by a standard fin effectiveness correlation. Radial variation of the natural convection heat transfer coefficient with the fin spacing in the radial direction is accounted for by averaging the local coefficients. The effects of the outer shroud, power conditioner, and insulated source container on the free convection currents and the effects of axial and peripheral variation in the heat transfer coefficient are neglected.

Several recent papers^{3, 4, 5} have reported experimental and analytical investigations of natural convection heat transfer from rectangular vertical fins. These papers treat a variety of configurations and compare their results to those of Elenbaas. Their heat transfer coefficients were within 10 to 20 percent of those of Elenbaas¹. Discussions with the Westinghouse Research Laboratories⁶ indicated that the Elenbaas correlation gives conservative results when applied to the fin cooling of conventional electrical apparatus. However, these papers did not have fin length to spacing ratios as considered for the 60-watt(e) system radiator, and further, the effect of radial variation in fin spacing was not considered. The assumption of one-dimensional heat conduction in the fin is a realistic assumption for fins with a high fin effectiveness (greater than 80 percent). An overall reduction of 20 percent from the heat dissipation capability predicted by this model is being made to provide an allowance for uncertainties in the model, effect of power conditioning structure, inlet and exit plenum losses, and fouling.

The following range of parameters was evaluated in a study of the effects of geometry on the heat dissipation capability:

Figure 3-35. Fin Profile for Parametric Study

- 1) Base plate temperature, 120 - 240°F
- 2) Ambient temperature, 50, 75, and 100°F
- 3) Fin I.D., 9 inches
- 4) Fin O.D., 20 - 29.5 inches
- 5) Maximum fin length, 7.25 - 8.25 inches
- 6) Converter axial length, 3.75 - 4.00 inches
- 7) Fin thickness, 0.031 - 0.125 inch
- 8) Fin spacing, (1-4) x fin thickness
- 9) Fin material, aluminum 1100 ($k = 128$) and 6061 ($k = 90$)

Results of the parametric study are:

1) For any particular set of parameters, the rate of heat-dissipation "peaks" over a range of "number of fins" as shown in Figure 3-36. This indicates a trade-off between the heat transfer coefficient and the heat transfer area. The optimum fin number varies from 120 to 180 over the range of fin O.D.'s and fin S/δ 's*

2) Figure 3-37 shows that the rate of heat dissipation at optimum fin numbers "peaks" over a range of fin S/δ 's. The heat dissipation rate increases linearly with fin O.D.'s, fin lengths, and converter lengths; the latter two effects are not shown on the figure.

3) Figure 3-38 indicates that the ratio of heat dissipated at the optimum fin number over fin weight increased with larger S/δ 's and decreased with larger O.D.'s.

4) Figure 3-39 is a composite curve for fin O.D. = 22.75 inches. This figure indicates the effect of S/δ on heat dissipation capability, fin thickness, fin weight, and performance to weight ratio. Results are given for only the optimized number of fins at each S/δ ratio.

Design guidelines for the radiator selection can be inferred from these results, namely:

1) A minimum-weight radiator design (which does little to reduce system weight but minimizes the dynamic load on the converter) should have a fin ratio S/δ , approaching 4 with the fin O.D. selected to match the heat rejection requirements.

* S/δ - ratio of fin root spacing to fin thickness

Figure 3-36. Heat Transfer Rate versus Number of Fins

Figure 3-37. Optimum Heat Transfer as a Function of Fin Geometry

Figure 3-38. Ratio of Optimum Heat Transfer to Fin Weight versus Fin Geometry

587-D1

Figure 3-39. Fin Performance

2) Maximum heat dissipation radiator design should have a fin ratio, S/δ , of 1 to 2.5 depending upon the fin O.D. which is selected to match the heat rejection requirements.

Based upon system optimization study results reported in Subtask 310, overall dimensions of a fin for optimum generator operation are:

fin O.D. = 22.75 inches

fin length = 7.25 inches

converter length = 3.75 inches

A fin design with 138 fins and a fin thickness of 0.063 inch has been recommended. Referring to Figure 3-39, this very closely corresponds to the optimum configuration for a fin O.D. of 22.75 inches. For (6061 grade) aluminum fins, this configuration provides a thermal rejection capability, as calculated by the model just described, of 1080 watts or 80 watts more than required at beginning-of-life conditions with an ambient air temperature of 75°F. This extra heat removal capability should result in the converter operating at a slightly lower cold side temperature at design conditions.

Mechanical design of the power conditioner equipment has required removing portions of several fins. To compensate for this reduction in heat transfer area and restrictions in convection, the fin length has been increased to 8.25 inches (without changing overall dimensions of the device). Figure 3-40 is a sketch of the basic fin shape not including the cut-out areas for the power conditioning equipment. The dashed lines in this figure represent the profile of the fin used to obtain the parametric study results. Computer calculations of the performance of the fins in this sketch, with allowances for reduction in area for the power conditioning equipment, indicate no reduction in performance.

The use of (1100 grade) aluminum with a thermal conductivity of 128 Btu/hr-ft°R provides an increase in performance of 75 watts over (6061 grade) aluminum for this design as indicated in Figure 3-41. It has been recommended that this material be used in the radiator design.

FIN THICKNESS = .063

NUMBER OF FINS = 138

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Figure 3-40. Sketch of Fin for Final Design

Figure 3-41. Fin Material Comparison

A thermal analysis of pin-type fins indicated an increase in performance of 5 to 10 percent over the rectangular fin. However, this gain in performance increases fabrication complexities and costs, and is not recommended for this design.

The transistor in the power conditioning equipment is mounted on a flat plate that dissipates heat by natural convection on two sides and radiation on one side. A thermal analysis of this component indicated that it is necessary to thermally join the plate to the radiator fins to provide sufficient heat transfer area. A 3-inch long x 1/8-inch ring of 1100 grade aluminum brazed to the top portion of the radiator O.D. has been recommended. Consideration has been given to the design of the outer shroud, fin outer ring supports, and rain shield to minimize their effect on obstructing the convection of air to the fins and in reducing radiation heat transfer. Instrumentation on the fins for the component and system testing has been located to provide minimum obstruction of the air flow.

The radiator-design computer program was modified to calculate the heat transfer from tapered fins. A fin-effectiveness-relationship for triangular fins, reported in Eckert and Drake⁷, was used. The modified program is being verified. A separate topical report is planned which will present the detailed results of the radiator study.

Heat Source Study

Considerable effort has been directed toward the development of an accurate analytical model of the SNAP-23A heat source system. In addition, numerous approximate calculations have been made to guide the design decisions.

Because of the dominant position played by the heat supply system, a detailed two-dimensional model of the system, as conceived two months ago, was developed and is described in this report. Heat losses and temperature distributions have been obtained from this model and are summarized. The first system assembly appears to have heat losses greater than those forecast in the proposal study, indicating a firm requirement for system improvements as more is learned about the effects of possible design changes.

Initial substantiation of the Min-K 2000 thermal conductivity being used in these studies was obtained from an independent experiment. However, the largest single uncertainty in the thermal performance of the first SNAP-23A prototype is still the effective conductivity of Min-K 2000 when fabricated and installed into the insulation geometry of this system. Also, experimental data must be obtained for Min-K 2000 when backfilled with krypton or during the evacuated state.

The second major uncertainty lies in the estimation of heat losses introduced into the system by instrumentation and heater leads. Instrumentation designs are now in the formulation stage. They will be evaluated as design concepts are defined in sufficient detail to permit analysis.

Model Description and Parametric Analysis

An iterative two-dimensional computer model with azimuthal symmetry, based on a finite difference method of analysis (TOSS), has been constructed to assist in the parametric study of the heat supply system. Represented in the model are the SrTiO_3 fuel pellets, titanium fuel capsule, U-8 w/o Mo shield, copper heat accumulator, stainless steel heat jacket, Min-K 2000 insulation, stainless steel support rods, and the stainless steel outer container. The converter was modeled by substituting an equivalent radial resistance for the thermoelectric region and a 50 percent density aluminum cylinder for the cold frame. All exposed surfaces, except the fins, were considered to be cooled by natural air convection ($h = 1 \text{ Btu/hr ft}^2 \text{ }^\circ\text{F}$, $T = 70 \text{ }^\circ\text{F}$). Fin cooling was modeled as an equivalent heat transfer coefficient connecting the converter cold-frame fin-assembly interface with the ambient air temperature.

Conductivity as a function of temperature for the materials used in SNAP-23A was obtained from published data. Convergence problems in the analytical model, however, required constant values for the material properties. Conductivities based on estimates of typical material temperatures were used in the first convergence calculation. Based on the results of this calculation, conductivity values were revised to reflect the better thermal information in later runs. A summary of the values used in the analysis is given in Table 3-5.

TABLE 3-5 CONDUCTIVITY OF MATERIALS USED IN ANALYSIS

Material Name	Conductivity (B/hr ft °F)
Fuel (SrTiO_3)	0.50
Jacket (Hastelloy C)	10.0
Shield (U 8 M_O)	20.0
Copper	200.0
Stainless Steel	14.0
Min-K (Air)	0.015
Min-K (Argon)	0.012
Tie Rod (Stainless Steel)	11.3
Aluminum	120.0
Mica	1.56

Parametric analyses were performed to obtain an understanding of the SNAP-23A system and to assist in developing an optimum system design. In addition to the primary design, variant cases were run with larger and smaller diameters than the 24-inch nominal case. This alters the available space for side insulation as well as changing the truss length and the areas of the bottom and rain cover. The rain shield clearance was varied from 1.0 inch in the nominal case to 1.5 inches and 2.0 inches. The Min-K 2000 insulation was changed from the base line calculation containing one atmosphere of air to calculations with argon backfill and also krypton backfill. In addition, the nominal case was run at end-of-life conditions to determine the effects of source power degradation. Results of these cases are summarized in Table 3-6. Figure 3-42 and 43 report the detailed temperature distribution of the nominal system at beginning-of-life and end-of-life respectively. The previous analyses were based upon a design envisioned in mid-March. The model, therefore, differs in some respects with the present design.

The model for the above analyses assumes a fuel capsule four inches in diameter and seven inches long surrounded by one-half inch of containment material and three inches of shielding (1.5 inches at the top). Power at the beginning of life was 1167 watts. There was six inches of thermal insulation on the side, two inches on the bottom, and one inch between the rain cover and the heat source insulator seal. The insulation seal was modeled as a stepped serpentine disk 0.010 inch thick connecting the shield container with the outer can. All exterior surfaces, except those of the fin, were connected to the 70°F atmosphere by a heat transfer coefficient of $1.0 \text{ B/hr-ft}^2\text{-}^\circ\text{F}$. The temperature of the converter cold-frame fin assembly was specified. Only radial heat flow was permitted in the converter. Assembly gaps or interface thermal resistances between components were neglected and no breaks or imperfections in the insulation material were assumed.

A new model has been prepared which will account for design changes made prior to May 10, 1967. The changes involve the regions above and below the fuel capsule, the stud region, and the insulation gas seal design.

Figure 3-42. Nominal Design Beginning of Life 1167 Watts Fuel

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Figure 3-43. Nominal Design End of Life 910 Watts Fuel

TABLE 3-6 PRELIMINARY THERMAL ANALYSIS SNAP 23A SYSTEM

Power - Watts Temp. - °F	Available Power	Losses					Heat To T/E	Max. Fuel Temperature	Min. Shield Temperature	T/E Hot Side Temp.	T/E Cold Side Temp.
		Bottom	Side	Rain Shield	Cap	Total Losses					
Proposal Design Min-K (Krypton) 24" D. 1/2" Neck	1168	30	57	92	0	179	989	1589	1190	1122	180
Phase I Design Min-K (Air) 24" D. 1.0" Neck	1167	53	74	75	6	208	960	1797	1130	1044	180
Phase I Design Min-K (Argon) 24" D. 1.0" Neck	1167	62	50	74	6	192	975	1836	1174	1105	180
Phase I Design Min-K (Air) 22" D. 1.0" Neck	1167	65	80	72	6	223	944	1768	1103	1019	180
Phase I Design Min-K (Air) 30" D. 1.0" Neck	1167	71	46	87	6	210	957	1688	1061	981	180
Phase I Design Min-K (Air) 24" D. 1.5" Neck	1167	69	64	64	6	203	964	1797	1132	1037	180
Phase I Design Min-K (Air) 24" D. 2.0" Neck	1167	70	67	57	6	200	967	1796	1131	1027	180
Insulation Seal Removed, Min-K(Air) 24" D. 1.0" Neck	1167	52	69	56	6	183	984	1807	1140	1053	180

The primary changes in the new model were made to: increase the shield thickness to three inches overall, change the design of the copper heat accumulator and stud to reflect latest mechanical designs, consider the effects of interface gaps and resistances, and create a more detailed representation of the converter region which includes axial heat flow as well as radial. The effect of the cooling fins was represented by an equivalent heat transfer coefficient linking the converter cold frame with the atmospheric temperature. This allows the cold frame temperature to vary with the converter heat through-put resulting in improved temperature distributions. Variant cases similar to those discussed herein will be studied as well as cases involving safety considerations such as absence of radiator and converter.

Interfaces

1. Cold Side Interface

A thermal analysis was performed to determine the thermal effects of mating the 6061 grade aluminum radiator base plate to the converter cold frame outer cover (304 grade stainless steel). The thermal performance of a conical-tapered outer-converter cover was compared with a cylindrical outer converter cover. To eliminate the possibility of the tapered outer converter cover stressing and distorting the converter aluminum cold frame, 20 percent of the tapered material was considered removed to make the material web area the same as in the cylindrical outer cover design.

Also affecting the thermal performance of the converter is the thermal resistance due to the interface that exists between the converter and radiator. An examination of the interface between the aluminum and stainless steel showed that the thermal conductance of this interface varies from $900 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ at 20 psi surface contact pressure to $1100 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ at 100 psi surface contact pressure with negligible variation due to surface finish. A 40 psi surface contact pressure was selected for the analysis so that the surface pressure during environmental conditions would never be greater than the maximum 120 psi limitation on the converter cold frame cover. At 40 psi surface pressure, the thermal conductance was $965 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ and produced an interface temperature drop of 4.7°F across the cylindrical

interface and 5.6°F across the tapered interface. In any case, without a minimum contact pressure, the interface temperature drop can be excessive, reducing the thermal efficiency of the conversion system.

The net effect of the overall thermal resistance is to reduce the usable temperature differential across the thermoelectric converter. For small temperature variations, the power output of the converter is directly proportional to the square of the temperature differential across the converter. The effect of the reduced temperature difference on the converter output for the tapered configuration is a 2.5 percent reduction in converter output whereas the cylindrical configuration yields a negligible reduction of 1.0 percent or less.

On this basis, the cylindrical converter outer cover with an interface material was selected since this configuration could have the least effect on the converter performance. A final selection of interface material will be made after further study of material compatibility and environmental conditions.

2. Hot Side Interface

The hot side interface is formed by the conically tapered (approximately 3 degrees per side) converter hot frame mating with the similarly tapered Inconel 718-covered copper heat-supply stud. Because of the nature of this configuration, problems would be anticipated when attempts were made to remove the converter (stainless steel hot frame) from the stud. Mechanical considerations dictated that an interface material for relatively easy release of the converter should be used. Thermal considerations required that a conformable interface should be included to eliminate a possible 2 mil void due to manufacturing considerations. A 2 mil void would cause a local thermal impedance causing a 40°F local temperature drop and have a significant effect on the converter performance. On the basis of manufacturing simplicity and material compatibility, mica was selected as the interface material. To allow for adequate conformity to guarantee the elimination of any gaps with a minimum thermal effect, 5 mils was selected as the mica sleeve thickness. A uniform temperature drop of 10 to 18°F is anticipated across the mica sleeve with an additional 10°F expected for mica to metal interfaces.

3. Interface Thermal Resistance

For the calculation of the thermal resistance between heat transfer surfaces, a number of references are available^{8, 9, 10, 11, 12, 13}. An approximate curve of interface thermal conductivity can be obtained pertaining to the converter to radiator interface (aluminum to stainless steel) and the converter to hot stud interface (stainless steel to stainless steel). The data used for the calculation of the approximate interface thermal drop is shown in the accompanying Figure 3-44. For complete verification of this information, it is necessary to run thermal contact resistance tests as proposed in Subtask 510.

4. Interface Materials

A number of possible interface materials were examined to reduce the temperature drop of the interface between the radiator and the converter. The effect of these materials in the interface is given in Table 3-7 below.

TABLE 3-7 TEMPERATURE EFFECT ON INTERFACE MATERIALS

Material	Interface Temperature Drop (°F/mil)
Teflon	2.6
Nylon	3.4
Silicone Rubber	2.8
Plastics	3.7 - 7.4
Silicones	3.5
Filled Silicones	0.3 - 1.0
Gold	0.03
Graphite	0.1
Silver	0.02
Platinum	0.1

Figure 3-44. Thermal Conductivity at Interface Versus Pressure

From the standpoint of ease of handling and material compatibility, the most likely candidate would be the filled silicones. These materials have been used successfully in similar interface applications for years in the semiconductor industry.

At the interface between the hot stud and converter where a very high temperature is anticipated, more stable interface materials are being considered. In particular, alumina, zirconia, magnesia, mica, gold, silver, graphite, platinum, boron nitride and molybdenum diselenide all appear attractive. Although mica has been selected for the first mockup, the other materials are being evaluated in various combinations from the standpoint of material compatibility and relative simplicity of manufacture for possible use in future systems.

Instrumentation

The objectives of the system instrumentation have not been changed from those described during the proposal effort. This area has been more thoroughly investigated and tentative locations defined.

1. Radiator-Instrumentation

The method of radiator test and instrumentation was defined to verify the design analysis. Under test conditions the following are to be determined:

- 1) Heat input to the base plate of the heat exchanger.
- 2) Temperature drop across the simulated thermoelectric converter heat-exchanger interface at the top, bottom, and center of the interface.
- 3) Thermal profile (axial and radial) of the radiator configuration from the fin base to the fin tip.
- 4) Thermal profile of the fins and base at four locations in four separate radiator quadrants. Full instrumentation on two diametrically opposite fins.
- 5) Test data acquisition at both BOL and EOL design heat rates into the base plate.
- 6) Temperature instrumentation at locations where the power conditioning is in thermal contact with the radiator.

Tentative instrumentation locations for the radiator are shown in Figure 3-45.

Figure 3-45. Radiator Instrumentation

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2. Converter Instrumentation

WANL and 3M have agreed to a number of thermocouple locations to determine the thermal mapping of the thermoelectric converter. Figure 3-46 schematically indicates the location of these thermocouples. From the thermocouples, information is expected to determine:

- 1) temperature profiles within the converter,
- 2) interface temperature drops,
- 3) temperature non-uniformity at the hot and cold junctions, and
- 4) converter and thermal losses.

The information will be used to achieve a consistency between the converter analysis and the test data. From this, improvements can be recommended for the 60-watt (e) system and the techniques can be applied to the design of the 25 and 100-watt (e) systems.

3. Heat Supply Instrumentation

The tentative thermocouple locations desired for thermal instrumentation of the heat supply are shown in Figure 3-47. The instrumentation is necessary to achieve a correlation between the thermal analysis and the actual system test performance. The instrumentation is expected to yield data to determine:

- 1) effectiveness of the heat source insulation,
- 2) support system thermal losses,
- 3) heat source temperature profile,
- 4) thermal losses of rain shield and heat source insulation seal,
- 5) heat source components interface thermal performance, and
- 6) temperature non-uniformities.

Since this instrumentation is the most difficult to install, some deletions may be necessary. However, much valuable information is expected (from this instrumentation) for system improvement recommendations and for the design of the 25 and 100-watt (e) systems.

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Figure 3-46. T/E Converter Instrumentation

○ THERMOCOUPLE LOCATIONS
ONE AT AXIAL LOCATIONS

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Figure 3-47. Heat Source Instrumentation

Subtask 350 - Thermoelectric Analysis

The thermoelectric converter for the SNAP-23A system is provided by the 3M Company utilizing the manufacturing and fabrication processes developed by 3M. From information provided by 3M, WANL has developed a computer code for comparative performance analysis of the thermoelectric converter. The computer code has been used to verify a consistency between WANL and 3M calculations and will be used for the following:

- 1) Predict thermoelectric converter efficiency as a function of hot and cold side temperatures
- 2) Describe the thermal resistance of the converter
- 3) Determine temperature limits
- 4) Examine the internal temperature profile of the converter
- 5) Correlate analysis and performance
- 6) Recommend design changes
- 7) Determine converter specifications
- 8) Perform design and analysis of the 25 and 100-watt (e) systems

To accomplish the objectives of the thermoelectric analysis, the schedule shown in Figure 3-48 describes the major areas of effort. The 3M performance analysis model, insulation, and instrumentation losses have been programmed into a code to predict the thermoelectric converter performance. The code is a one-dimensional analysis depending upon the balance of no-current heat flow between the hot and cold leg segments to determine the operating leg segment temperature. Two-dimensional heat flow and surface temperature non-uniformities are not included in the analysis. The instrumentation and insulation losses of the converter are considered to be heat flows parallel to the flow within the thermoelectric elements with no interchange of heat flow between the two. A simplified logic and flow diagram of the computer code is shown in Figure 3-49.

The code has been used to achieve a consistency in calculation of converter performance between WANL and 3M. A tabulation of the calculated performance to achieve the consistency of calculations is given in Table 3-8.

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Figure 3-48. Thermal Analysis Schedule

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Figure 3-49. Simplified Flow and Logic Diagram of Converter Computer Code

TABLE 3-8 CONVERTER ANALYSIS COMPARISONS SAMPLE CASE

Input Data	Information Value	
Number of couples	252	
Area of P-leg	0.0276 in. ²	
Area of N-leg	0.0276 in. ²	
Total length of N-leg	0.502 in.	
Length of N-leg cold segment	0.125 in.	
Total length of P-leg	0.356 in.	
Length of P-leg cold segment	0.125 in.	
Hot shoe resistance	4.4 x 10 ⁻³ ohms/couple	
Cold shoe resistance	0.5 x 10 ⁻³ ohms/couple	
Miscellaneous resistance	79.4 x 10 ⁻⁶ ohms/couple	
Hot junction temperature	1100°F	
Cold junction temperature	190° F	
	WANL Calculation	3M Calculation
N-leg segment temperature	407.8°F	408.3°F
P-leg segment temperature	578.8°F	576.7°F
Total generator resistance	9.723 ohms	9.747 ohms
Matched load current	2.68 amps	2.67 amps
Load voltage	26.04 volts	26.04 volts
Open-Circuit voltage	52.07 volts	52.05 volts
Load power	69.8 watts	69.56 watts
Efficiency	8.58%	8.59%

The slight difference in calculations can be attributed to the convergence criteria adopted by 3M for the determination of the segment temperature of each leg. The criteria for convergence is that the heat conducted through the leg hot-segment must equal the heat

conducted through the leg cold-segment within one milliwatt. Initial indications are that this convergence criteria may result in a leg segment temperature variation of plus or minus 4°F and could cause the discrepancy of calculation noted above.

An attempt was made to correlate the WANL code performance for the 3M proposed 280 couple configuration. This was based on information received from 3M. The end-of-life case was examined and the comparison with the 3M data received is tabulated in Table 3-9.

TABLE 3-9 280 COUPLE CONVERTER COMPARISONS

	WANL	3M
Number of couples	280	280
Diameter of P-leg	0.187 in.	0.187 in.
Diameter of N-leg	0.187 in.	0.187 in.
Length of P-leg	0.327 in.	0.327 in.
Length of N-leg	0.479 in.	0.479 in.
Cold segment length of P-leg	0.114 in.	0.114 in.
Cold segment length of N-leg	0.120 in.	0.120 in.
Hot junction temperature	890°F	890°F
Cold junction temperature	180°F	180°F
Load voltage	24.56 volts	24.72 volts
Open-Circuit voltage	42.71 volts	42.78 volts
Load power	54.5 watts	61.4 watts
Total heat required	788 watts	796 watts
Overall converter efficiency	6.93%	7.7%

The magnitude of the inconsistency between the calculations of WANL and 3M led to an immediate investigation of the cause.

The efficiency of the thermoelectric converter is

$$\eta_{\text{converter}} = \frac{\text{Power to load}}{\text{Heat in}}. \quad \text{Under operating conditions this becomes}$$

$$\eta = \frac{V_L \times I_L}{Q_{\text{cond}} + Q_{\text{pelt}} - 1/2 (Q_T + Q_J) + Q_{\text{shunt}}}$$

where

$V_L \times I_L$ = Electrical power delivered to the load

Q_{cond} = Heat conducted through thermoelements

Q_{pelt} = Peltier heat absorbed at the hot junction

Q_T = Thompson heat generated in thermoelements

Q_J = Joulean heat generated in thermoelements

Q_{shunt} = Heat loss through insulation, converter wall, etc.

The terms $V_L \times I_L$, Q_{pelt} , Q_T and Q_J , are all proportional to the current flowing in the thermoelectric elements. There is very good agreement of voltage calculations between WANL and 3M. There is, however, a discrepancy in the load power and total heat input to the converter. An increase in the current would increase the load power and efficiency. The logical conclusion is that the current produced at the load voltage is lower in the WANL calculation than the 3M calculation. The only way the current can increase while maintaining the same voltage is to decrease the internal resistance since

$$I_L = \frac{V_{\text{oc}}}{R_{\text{int}} + R_L}$$

with V_{oc} = Converter open circuit voltage

R_{int} = Converter internal resistance

R_L = External load resistance

The internal resistance of the converter is made up of two components, the thermoelectric material resistance, and the extraneous resistance. The thermoelectric material

resistance, is fixed by the choice of N and P type thermoelectric materials on the element geometry and operating temperature, since $R = \rho \frac{T_h - T_c}{A}$. The extraneous converter resistance is made up of the hot shoe resistance, the hot button resistance, the cold shoe resistance and the contact resistance, all of which may vary depending upon materials, geometry, temperature and contact pressure. This extraneous resistance must have been considerably lower in the 3M calculation than the value used in the WANL calculation. Verbal verification was received from 3M that the extraneous resistance values anticipated for the SNAP-23A converter have been considerably reduced from the values previously supplied by 3M. Also, 3M indicated that the method of calculation for predicting converter performance was changed from the description supplied to WANL. Rather than calculating the hot shoe, cold shoe and miscellaneous resistance directly, the total of these values is taken as a fixed percentage of the thermoelectric material resistance. The fixed percentage of extraneous resistance used in 3M's SNAP-23A calculation is 6.33 percent* or considerably less than the 15.3 percent** previously achieved in the SNAP-23 program.

There is a justification for using a fixed percentage extraneous resistance calculation since there is considerable variation of resistance with temperature. The extraneous resistance is expected to reduce rather than remain constant with decreasing temperature, especially when temperature extremes of the beginning-of-life and end-of-life conditions are expected. WANL is modifying its code to include the option of accepting either a fixed percentage or direct calculation extraneous resistance.

3M is preparing the applicable SNAP-23A Converter No. 1 information and is expected to transmit it to WANL on May 24, 1967. When this information is received, a calculation can be performed to confirm the Converter No. 1 performance forecast being quoted by 3M and a complete converter performance analysis can then be generated and reviewed. It is WANL's interpretation that the achievement of the required end-of-life converter efficiency will depend in large measure on 3M's ability to make significant reductions in converter extraneous resistance and/or reductions in shunt heat loss relative to previous SNAP-23 experience.

*Verbal information received from 3M on 5-22-67

**"SNAP-23 Program Terrestrial Radioisotope Thermoelectric Power Supply System, Parametric Study Thermoelectric Converter, " Report No. MM 3418-33, Minnesota Mining and Manufacturing Company, Dec. 20, 1966.

Subtask 360 - Materials and Processes - Design Support

Objective

The objective of this subtask is to provide support for the materials selection aspect of system design and performance analysis by supplying materials and process specifications; determine known or available materials data including elevated temperature mechanical, compatibility and thermal stability as they relate to long time system reliability; investigate and resolve materials process problems occurring in fabrication, testing, and analysis.

Materials Selection

In the SNAP-23A system, each component performs a specific function which is dependent to a large degree on a combination of particular materials' properties. Materials were selected which possessed the optimum desired properties consistent with availability and cost. With this approach, the program objectives of minimum system volume, weight, and cost can be realized. In the following discussion, each component will be reviewed with respect to function, materials selection, and potential problem areas.

1. Isotope Capsule

The function of the capsule is to contain the isotope fuel for the life of the system and to satisfy nuclear safety requirements. The major requirements for the SNAP-23A fuel capsule material are compatibility with the fuel material, resistance to seawater corrosion and air oxidation, compatibility with the biological shield, and fabricability in the required sizes and shapes. Hastelloy C was selected as the isotope containment material because of its excellent seawater corrosion and air oxidation resistance. Sea water burial tests conducted by the International Nickel Company and NRDL indicate negligible weight loss after ten years submergence.^{14, 15} Limited compatibility tests with non-radioactive Sr Ti O_3 and the apparent successful operation of the SNAP-7 series of generators indicate that there is no apparent compatibility problem with Hastelloy C in contact with Sr Ti O_3 .^{16, 17} However, the compatibility of Hastelloy C with the biological shield represents a major area of concern. Nickel, the base

metal of Hastelloy C, and uranium, the major constituent of the shield, form a low melting eutectic at about 1350°F and thus require a barrier layer to prevent contact of the two materials while at or near the eutectic temperature.

In lieu of the lack of experimental data concerning the compatibility of various potential barrier materials with uranium alloys and Hastelloy C at the temperatures of interest, 900 to 1400°F; selection of a barrier material was made on the basis of available thermodynamic data in the form of constitutional phase diagrams.¹⁸ On the basis of these data, titanium, tantalum, and tungsten appear capable of performing the function of a barrier layer. Cost and handling considerations favor the use of titanium, since experimental data concerning the interaction of each of these metals with U-8 Mo at high temperatures are lacking.

2. Biological Shield

The purpose of the biological shield is to attenuate the radioactivity of the fuel in accordance with nuclear safety requirements. For the SNAP-23A design, it is desirable that the shield materials accomplish the nuclear safety objectives utilizing a minimum weight and volume of material. Of all the heavy metals considered for the shield, uranium or a uranium alloy is the optimum material with respect to the volume and weight requirements.

Unalloyed uranium, however, undergoes an allotropic phase transformation at approximately 1225°F which is accompanied by a significant change in volume.¹⁹ The addition of molybdenum in concentrations above 4 to 6 w/o stabilizes the high temperature body-centered cubic γ phase down to room temperature provided the cooling rate is rapid. The γ phase is metastable and transforms to $\gamma^1 + \alpha$ phases during prolonged thermal exposure at temperatures below the eutectoid temperature of 1064°F.²⁰ The time required for complete transformation and the extent of the accompanying volume change is a function of the alloy composition and holding temperature. U-8 w/o Mo was selected as the shield alloy because it represents the best compromise between metallurgical behavior, mechanical properties, availability, and fabricability. Since most applications of U-8 w/o Mo alloy are for use at temperatures below 700°F, the metallurgical behavior as well as mechanical properties at the higher temperatures, 900 to 1400°F, have not been well defined.

In the heat path of the SNAP-23A system, the biological shield is located between the fuel capsule and the copper heat conductor. The compatibility problem between the capsule material, Hastelloy C and U-8 Mo has been discussed in the previous section concerning the heat source capsule. Although phase diagrams of the binary titanium-uranium, titanium-molybdenum systems indicate that the formation of a low melting phase is not likely, interdiffusion and the formation of intermetallic phases may produce difficulties. At the U-8 Mo/Cu interface, a similar condition exists. Since the uranium-copper eutectic temperature is above 1700°F, the compatibility of this interface will be controlled by the rate of interdiffusion and generation of intermetallic phases.¹⁸ These data will have to be determined experimentally for the temperature of interest. If it be demonstrated experimentally that adverse effects are produced by the interdiffusion of copper and U-8 Mo, an intermediate barrier layer such as tungsten may be required to retard the interaction.

3. Heat Accumulator

The purpose of the heat accumulator is to transfer heat from the source to the thermoelectric generator. Unalloyed copper was selected as the heat conductor material because of its superior thermal conductivity relative to copper alloys. The heat source design will be modified in such a way that the copper conductor will not be required to be a structural member because of the low creep resistance of copper at the initial operating temperature. Of the commercial grades of copper available, OFHC copper was selected as the heat conductor material because (1) it possesses the highest thermal conductivity, (2) it is the most fabricable, and (3) joining presents the least difficulties.²¹

Compatibility with the materials on each side of the heat accumulator is an area of uncertainty as described in previous sections. The interface with the outer jacket which will be made of Inconel 718 is not as critical as the other interfaces discussed. Except in the area of the thermoelectric generator, efficient transfer of heat across this interface is not critical; however, compatibility is still of initial concern for the integrity of the system.

4. Heat Source Jacket

The heat source jacket is intended to protect the copper accumulator and U-8 Mo shield from atmospheric contamination during shipment of the capsule from the loading site (ORNL to WANL), for final assembly and during normal operation of the system. An austenitic stainless steel was initially intended for use as the heat source jacket; however, strength and assembly requirements favor the use of a higher strength material with a lower coefficient of thermal expansion. To reduce gaps in the heat path due to manufacturing tolerances, a jacket of low thermal expansion material would put a compressive load on the copper accumulator as the temperature of the system goes from ambient to operating temperature. High creep strength is desired to prevent relaxation of the tension in the jacket and thus permit the jacket to follow the copper accumulator as the system cools during the life of the system. Heat treatable nickel base appears to satisfy these requirements. The problems of fabrication common to these materials such as their high work hardening characteristics and tendency toward loss of strength when welded must be considered. Tentatively, Inconel 718 appears to be the most promising because of its high strength and resistance to overaging when welding in the solution treated condition.²²

5. Insulation

To achieve an effective SNAP-23A design, it is necessary to minimize heat losses. This can be accomplished by judicious selection of insulating materials. The ideal insulating material for this application must have: low thermal conductivity which does not change with time at temperature, dimensional stability with time at temperature, and fabricability. Unfortunately, no one insulator possesses all of the desired characteristics; therefore, the final selection must ultimately be a compromise. To select the most suitable insulation, a survey of potentially attractive insulations was conducted. The survey was limited to materials which have demonstrated chemical, mechanical, and physical stability under continuous operational conditions in a 500 to 1800°F environment, and to materials with thermal conductivities of less than 1 Btu/hr-ft²-°F.

Of the 15 insulating materials reviewed, Linde Super Insulation and Johns-Manville MIN-K-2000 have by far the lowest thermal conductivity. Because of uncertainties concerning the shock and vibration characteristics, long term vacuum requirements, and directional properties which presents a developmental problem for adoption of this material to a particular system, the super insulation was not considered for use in the SNAP-23A system at this time.

The MIN-K-type insulations which possesses the next best thermal conductivities have been used successfully in a number of isotope systems. Current work at Johns-Manville under AEC sponsorship is being conducted to improve outgassing, handling, and mechanical and thermal properties of this class of materials. Commercially available MIN-K-2000 was selected for use in the prototype systems which are being built in Phase I of this program. As the improved formulations are developed and become commercially available, they can be readily incorporated into the SNAP-23A design with a minimum of difficulty.

In view of the range of service temperatures anticipated for the insulation, MIN-K-2000 exhibits satisfactory thermal conductivity, dimensional and chemical stability, and low degradation for long-term operation. There also exists the possibility of further reduction of the thermal conductivity of MIN-K-2000 through evacuation and backfilling with an inert low conductivity gas such as krypton. However, the reduction of thermal conductivity using an inert gas backfill is predicted on the basis of calculated values which have not been verified experimentally. In order to utilize the inert gas backfill, techniques for replacing air entrapped in the insulation with the inert gas without causing an adverse effect on thermal conductivity would have to be developed and experimentally verified.

Utilization of MIN-K-2000 does present a fabrication problem. Commercial MIN-K-2000 has several aspects which must be understood in order to fully appreciate the fabrication problem.

MIN-K-2000 contains a binder and asbestos fibers to increase handling ease and machinability. Previous experience at Martin with the SNAP-11 system, at the Westinghouse Aerospace Electrical Division with a 100 watt RTG system for buoy application, and at Westinghouse

Astronuclear with a five watt RTG program has shown that MIN-K insulation must be heat treated to remove the phenolic binder prior to high temperature use in a confined enclosure.²³ The organic binder is removed by heating the MIN-K slowly in air to 1100°F and holding for an extended period of time. The binder is oxidized and evolves as gas. The reaction is exothermic, thus care must be taken lest the center of the MIN-K overheats causing sintering and a resultant degradation of thermal conductivity properties. The heat treatment produces a material which is more amenable to machining; however, the material is friable and not conducive to rough handling. This situation does present a fabrication problem since MIN-K-2000 is normally produced as a 2 in. x 18 in. x 36 in. sheet. With this type of material, the insulation components are formed by rough cutting, heat treating to remove the binder, stacking them to an oversized configuration, and clamping and machining to final dimensions. The insulation component is then machined to allow for the heat source support assembly. The insulation is then fitted into the insulation can assembly; the end plates are welded into place; the container is outgassed, then backfilled with a partial pressure of inert gas. The latter step is required since voids cannot be excluded during assembly.

An alternate approach which is being considered for Phase II manufacture of SNAP-23A systems, involves the direct compaction of the insulation into the insulation assembly container. This procedure is possible only with insulation which does not require an outgassing step such as the new formulation of MIN-K now being developed at Johns-Manville.²⁴ A low temperature bake-out at 300 to 400°F would be required to reduce absorbed gases.

6. Heat Source Support Assembly

To provide support for the heat source, a support assembly was incorporated into the insulation assembly. The support assembly positions the heat source relative to the insulation assembly and provides support during shock loading while causing a minimum heat loss. Since one end of the support system will be at the temperature of the outer heat source jacket, the support assembly material must have sufficient mechanical strength to sustain short-time high-stress loads as well as creep resistance at low loads.

Inconel 718 in the heat treated and aged condition possesses the desired mechanical strength in addition to sufficient weldability to permit its use as the support member.²²

7. Radiator

The primary function of the radiator assembly is to dissipate heat from the thermoelectric generator and power conditioning system. The primary requisite of the fin material is high thermal conductivity at low weight. The radiator assembly is attached to and supported by the thermoelectric generator and, therefore, must be minimized with regard to weight. Design requirements indicate that the minimum allowable thermal conductivity for the radiator fin material is 90 Btu/hr-ft-°F. These requirements preclude the use of all materials except aluminum or an aluminum alloy.

Data published by the G1 corrosion committee of the ASTM indicate that aluminum and its alloys when properly treated can withstand a humid marine environment for periods of time well beyond the ten year life of the SNAP-23A system.^{25, 26, 27} The use of aluminum with stainless steel in a marine environment is not as serious a problem as indicated by the position of these materials in the electromotive series. Galvanic corrosion of aluminum and stainless steel can be minimized through design by the elimination of crevices and through the use of protective coatings produced by anodizing and chromate sealers. Tests of unprotected aluminum in contact with stainless steel at the various test stations around the world indicate that a natural passivity is developed by the materials which tends to minimize the galvanic effects that occur in a marine environment.

Specifications

One of the specified responsibilities of this subtask is to review and recommend specifications which identify materials and processes required to fabricate the SNAP-23A system. As the design of the system progresses, government and commercial specifications are reviewed for applicability. In those instances where certain materials and processes are not covered by existing specifications, new specifications are to be generated. Table 3-10 lists the components, materials, and processes identified with appropriate recommended specifications.

TABLE 3-10 RECOMMENDED MATERIALS AND PROCESS SPECIFICATIONS

Component	Material or Process	Recommended Specification
Fuel Capsule	Hastelloy C, sheet Hastelloy C, Bar	AMS-5530 AMS-5750
Diffusion Barrier	Titanium	MIL-T-9046, Type 1 Composition A, unalloyed
Accumulator	Copper, OFHC	ASTM-B-170
Heat Source Jacket	Inconel 718 * Heat Treatment, Inconel 718 * Welding, Inconel 718	AMS-5596A WANL-PS-294641-4 WANL-PS-597930
Shield	*† U-8Mo-, Casting	WANL-PDS-30107-1
Insulation	† MIN-K-2000 † Heat Treatment, MIN-K-2000	WANL-PDS-30186-1 WANL-PS-597918-1
Rainshield	† Elastomer	MIL-STD-417 (Cord) R615, A, B ₁ , C ₂ , D, L (10% Vol/change)
Radiator	Aluminum 1100 Aluminum Alloy 6061 Dip Brazing of Aluminum Anodizing of Aluminum Anodized Coating Sealer	QQ-A-250/1C QQ-A-250/11C AMS-2673 MIL-A-8625, Type II, Class 1 MIL-P-8585 A

*These specifications were generated for this program.

† These specifications are included in Appendix

As additional materials are designated and requirements are delineated, applicable specifications will be reviewed and recommended or generated as required.

Materials Properties

An additional requirement of this subtask, is to provide required materials data to the design section. Standard materials data sources were reviewed, data compiled and transcribed to the standard WANL Materials Manual format. Table 3-11 indicates the status of the materials data review and evaluation.

TABLE 3-11 STATUS OF MATERIALS DATA

Component	Material	Thermal Expansion	Thermal Conductivity	Tensile Strength	Yield Strength	Elongation	Modules of Elasticity	Creep Properties	Wt loss vs Temperature
Capsule	Hastelloy C	X	X	X	X	O	O	X	
Barrier	Titanium	X	O	X	X	O	O	X	
	Tartelium	X	X	X	X	X	X		
	Tungsten	X	X	X	X	O	X		
Shield	U-8 Mo	☒	☒	☒	☒	☒	☒	☐	
Accumulator	Copper	X	X	X	X	O	X	O	
Heat Assembly Jacket Truss Support	Inconel 718	X	O	X	X	X	X	X	
Insulation	MIN-K-2000		☒					☐	X
Radiator	Aluminum - 1100	X	X	X	X	X			
	Alloy 6061	X	X	X	X	X	X		
Outer Jacket	Inconel 600	O	O	O	O	O	O	O	

X DATA COMPLETED

O DATA STILL TO BE COMPILED

☐ DATA TO BE DETERMINED EXPERIMENTALLY

Subtask 370 - Shield Design

The overall objectives of this subtask include the following:

- 1) To design a minimum weight biological shield for the 60 watt(e) generator system which is a practical hardware configuration, readily fabricable and compatible with the system requirements.
- 2) To conceptually design the biological shields for the 25 watt (e) and the 100 watt (e) generators.
- 3) To prepare test specifications for the gamma flux mapping experiments on a capsule-shield assembly to be performed at ORNL

During the subject reporting period, the entire effort was directed toward the 60 watt(e) design. Major emphasis was placed upon the shield parametric study. The following describes the shield parametric study in detail.

Scope of Shielding Parametric Study

A parametric shielding study was performed by Hittman Associates, Inc. consistent with the 60 watt(e) system weight optimization study. The parameters of primary concern in the study were the SrTiO_3 pellet diameter, the thermal power of the source, and the overall RTG dimensions which were varied by changing the thermal insulation thickness. The ranges over which these parameters were varied are given in Table 3-12.

TABLE 3-12 RANGE OF VARIABLE PARAMETERS

Fuel Diameters (SrTiO_3)	3, 4, 5 inches
Thermal Power	1200 - 2000 watts
Insulation Thickness-Radial	5-8 inches
Top (includes Cu Stud)	6.0 - 9.0 inches
Bottom	2.0 - 4.0 inches

The dose rate requirements used to establish the required U-8 Mo shield thickness were based upon nuclear safety, shipping, and handling considerations. As indicated in the nuclear safety section, the dose rate must not exceed 200 mrem/hr at the generator surface or 10 mrem/hr at a distance one meter from the generator surface.

Analytical Method

All dose rate calculations were made using either the SCIS or QAD computer codes. Both of these codes are capable of performing multigroup gamma ray shielding calculations for multiregional cylindrical systems.

The SCIS code²⁸ is a multi-energy, multi-region computer code written to perform gamma shielding analysis for isotopic sources in the form of a right circular cylinder. SCIS employs the methods and procedures described in the Reactor Shielding Design Manual²⁹ for replacing a cylindrical source by a line source for radial calculations, and by a disk source for axial calculations.

The code computes the energy dependent uncollided particle flux, the energy dependent dose rate, and the total flux and total dose rate for gamma rays in both radial and axial directions. The code is limited to a maximum of ten radial and axial materials (including the source region) and to eleven gamma ray energy groups.

The QAD code was written to compute both fast neutron and gamma ray penetrations through shielding assemblies and materials. The gamma ray computations are based on point kernel ray tracing techniques which use a polynomial buildup factor. Sources may be specified in either rectangular, cylindrical, or spherical geometry. Zones (or space regions) are formed by surface boundaries which intersect and establish the desired volume enclosures. For gamma rays the code calculates and prints out fluxes and doses with and without buildup factors and energy absorption rates.

Materials Properties

The materials properties of most interest in the shielding parametric study were the nuclear radiation spectrum from the SrTiO_3 source and the gamma ray attenuation properties of the SrTiO_3 and the U-8 Mo shield.

The nuclear radiation properties of the SrTiO_3 are listed in Table 3-13. All photon emissions listed in the table are derived from Reference 30 located in the Bibliography. These photons result from bremsstrahlung only. All emission rates are calculated using Wyard's formulation for bremsstrahlung photon production.

TABLE 3-13 NUCLEAR RADIATION PROPERTIES OF SrTiO_3

	Max Energy Mev	Avg Energy Mev
Sr-90	2.27	0.545
Y-90	0.944	0.201
Photon Energy (Mev)	Photon Emission Rate photons/watt.sec	
0.25	1.705×10^{11}	
0.50	6.612×10^{10}	
0.80	1.958×10^{10}	
1.10	6.103×10^9	
1.40	1.650×10^9	
1.70	3.027×10^8	
2.00	2.082×10^7	

Photon attenuation coefficients for the materials used in the RTG are given in Table 3-14. In the shielding calculations, in both QAD and SCIS, buildup factor coefficients for depleted uranium were used to calculate buildup factors for U-8 Mo.

RTG Geometry Model

While the basic shape of the SNAP-23A system is known, all dimensions are subject to minor changes. Since some of these changes are very small and do not appreciably change the required shielding thicknesses, these system dimensions were fixed for purposes of the shielding parametric study. The key parameters which significantly affect the shield design were varied over reasonably ranges. The dimensions used in the shielding study are given in Table 3-15.

As seen in Table 3-15, the only variables considered parametrically were fuel diameter, thermal power level, and insulation thickness. Also, as is noted in the table, no shielding credit was taken for the copper stud on top of the system.

TABLE 3-14 PHOTON ATTENUATION COEFFICIENTS FOR MATERIALS USED IN RTG

Material	0.25 Mev	0.5 Mev	0.8 Mev	1.10 Mev	1.40 Mev	1.70 Mev	2.00 Mev
SrTiO ₃	0.520	0.3392	0.2696	0.2332	0.206	0.1872	0.1728
Hastelloy-C	1.2963	0.7358	0.5748	0.4935	0.4354	0.3916	0.3656
Titanium	0.5445	0.3870	0.3078	0.2700	0.2385	0.2160	0.1939
U-8 Mo	13.2292	2.9354	1.6147	1.2215	1.01094	0.89610	0.8317
Cu	1.4112	0.73390	0.5853	0.50389	0.4448	0.3947	0.3741
Stainless Steel	0.9821	0.67057	0.5385	0.46449	0.41055	0.3735	0.3437
Insulation	0	0	0	0	0	0	0

TABLE 3-15 RTG DIMENSIONS USED IN PARAMETRIC SHIELDING STUDY

Region	Material	Dimensions
Source	SrTiO_3	3, 4, 5, in. diameters 12,000 - 20,000 th(w) 0.85 w/cc
Clad	Hastelloy-C	0.35 in. radial thickness 0.60 in. axial thickness
Diffusion Barrier	Ti	0.04 in. thickness
Shield	U-8 Mo	Thickness as required to give surface dose rate 200 mrem/hr and 1 m dose rate 10 mrem/hr
Heat Accumulator	Cu	0.37 in. radial thickness 0.25 in. bottom thickness 0.75 in. top thickness height 5.26 in. diameter 4.25 in.
Clad	Stainless Steel	0.05 in.
Insulation	Min-K	5 in. - 8 in. radial thickness 2 in. - 4 in. bottom thickness 2 in. - 4 in. top thickness
Insulation Can	Stainless Steel	0.125 in.

The shield weight calculations were performed using the following geometrical assumptions. Radial insulation and shield thickness was assumed as one third of the radial insulation thickness (thus giving a bottom shield thickness) and the top thickness was chosen so that the distance from the top of the shield to the RTG surface was 8.875 inches.

Discussion of Results

The results of the shielding parametric study are summarized in Figures 3-50 to 60. The data include shield thickness as a function of power level, source diameter, and insulation thickness. Shield weights are also presented.

Radial shield thicknesses for the 3, 4, and 5 inch diameter fuel pellets are given in Figures 3-50, 51, and 52. It is clear from an examination of these figures that the 10 m rem/hr dose rate at one meter from the RTG surface is the more stringent requirement for these cases. For a given power level and insulation thickness, these curves show that the required shield thickness decreases with increasing source diameter due to the larger amounts of attenuation within the source.

Figures 3-53 and 3-54 give the required shielding thickness below and above the source. Below the source the surface dose rate of 200 mrem/hr is the more stringent requirement while above the source the 10 mrem/hr requirement is the more stringent. Since the copper stud is neglected as a shielding material, the only difference between the top and bottom is the distance from the shield to the generator surface. The distance from the source to the bottom surface is shorter. This difference in separation distance shifts the more limiting dose rate constraint from the surface dose to the one meter point at the base of the generator. These figures also show only one curve for each diameter source. The reason for this is that the axial shield thicknesses are independent of power level in the range of interest for this study. This result is as expected for source heights which are greater than or equal to $3/\mu_s$ (where μ_s is the photon linear attenuation coefficient for the source) for all photon energy levels. For sources of these heights, increasing the power level while holding the diameter constant makes the source longer. This new portion of the source then makes very little contribution to the photon flux leaving the opposite end of the source.

Figure 3-50. Radial Shield Versus Insulation Thickness

Figure 3-51. Radial Shield Thickness

Figure 3-52. Radial Shield Versus Insulation Thickness

Figure 3-53. Shield Versus Insulation Thickness for Bottom

Figure 3-54. Shield versus Insulation Thickness for Top

Figures 3-55 to 3-60 summarize the shield weights which have been calculated utilizing shield thicknesses presented in Figures 3-50 to 3-54, the geometrical assumptions previously delineated and a shield density of 17.4 g/cc. Figures 3-55 to 3-57 give shield weights as a function of generator outer diameter and thermal power for a constant pellet diameter, while Figures 3-58 to 3-60 present shield weights as a function of source diameter for a given constant power level.

Summary of Results

From Figures 3-58 to 3-60 it is evident that a substantial shield weight penalty results from the use of a 3-inch diameter fuel pellet over the entire 1200-2000 watt power range considered. There is, however, an inconsequential weight penalty associated with a 4-inch diameter pellet for the 1200 watt case, but significant penalties result from the use of this size pellet for the 1600 and 2000 watt cases. A five-inch pellet leads to the lowest weight shield over the total power range considered, with the weight savings increasing as power level increases. These results can be understood when it is noted that for the case where the radial, top and bottom shield thicknesses are identical, the ratio of shield inner surface length to diameter (L/D) for minimum shield weight is exactly unity. As a first order approximation, it can be assumed that the minimum weight SNAP-23A shield for a given power level is the one having a fuel pellet with an L/D of unity. This is a reasonable approximation since SNAP-23A radial and axial shield dimensions are comparable and the L/D of the fuel pellet is very close to the L/D of the inner shield surface or outer fuel capsule surface. Figures 3-58 to 3-60 include the fuel pellet L/D and are useful in interpreting shield weight results. For example, for the 1200 watt case, Figure 3-58, the 3-inch diameter pellet leads to an L/D of 4.09 which is quite unfavorable compared to the L/D of the 4 and 5-inch cases. This explains the substantial weight penalty associated with the use of a 3-inch pellet. Similar trends can be identified and interpreted for the 1600 and 2000 watt cases.

Figure 3-55. Shield Weight Versus Outside Diameter

Figure 3-56. Shield Weight Versus Outside Diameter

Figure 3-57. Shield Weight Versus Outside Diameter

Figure 3-58. Shield Weight Versus Outside Diameter

Figure 3-59. Shield Weight Versus Outside Diameter

Figure 3-60. Shield Weight Versus Outside Diameter

Subtask 380 - Electrical Design

The objective of this subtask is to provide the detailed design of an electrical power conditioning subsystem which controls system output to 24 volts.

The activity in Subtask 380 includes the design and breadboarding of an electronically simulated thermoelectric generator, the breadboarding of the shunt regulator shown in the proposal, breadboard testing on the preliminary 60 watt electrical design, and preliminary thermal calculations and tests to establish the heat sinking requirements for the power conditioning subsystem.

The electronically simulated thermoelectric generator can be used to checkout and test the electrical power conditioning system by linearization of the generator output voltage versus output current. Depending on where the linear characteristic intercepts the actual load voltage curve, an error of up to 12.5 percent occurs. This error does not affect the operation of the power conditioning system; however, it will affect the ultimate temperature profile of the unit. Prior to the temperature profiling measurements, the simulation will be upgraded.

The shunt regulator shown in the proposal was breadboarded utilizing available parts, and preliminary operational tests were conducted. Results of these tests indicate that the design system is operational. As shown in Figure 3-61, the proposed circuit utilizes a "quad" arrangement for the shunt transistors. Failure of any one of the shunt elements, open circuit or short circuit, will not cause the entire system to malfunction.

The symmetrical arrangement is the prime advantage of this circuit. The four legs of the quad arrangement are identical. Parts were substituted to check load-sharing of the quad elements, and no problems were observed for step changes in load or input. No efforts were made to cause the four elements to load share since the main function of the circuit multiplicity is for the purpose of reliability. The possibility of forcing some load-sharing without interconnecting the four legs electrically is being investigated; however, the

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reliability requirement states that one of the four elements must be able to dissipate all of the power in the event of a system or load malfunction when one of the quad elements fails.

Heat transfer calculations were made, including the effects of the radiation and thermoelectric generator on the heat transfer coefficient for the three inch wide ring around the circumference of the radiator. As a result of these calculations, the ring is to be brazed to the radiator in order to provide additional area required for the dissipation of heat in the power conditioning system under maximum input power (no load conditions). The effective heat transfer coefficient of the ring is reduced by approximately 50 percent when the overall system is evaluated. This is attributed to a reduction of air flow on the inside surface of the ring. Further thermal calculations must be conducted on the proposed power transistor connection before a definite means of connecting the power transistor to the shroud is established.

Subtask 390 - Capsule Design and Development

The overall objectives of this subtask are to design and demonstrate the fabricability of the SNAP-23A fuel capsule. As a reference design, the capsule consists of strontium titanate fuel doubly encapsulated in Hastelloy-C cans. The fuel is in the form of strontium titanate pellets four inches in diameter and approximately 3.25 inches long. Two of these pellets are contained within a sealed inner can, or liner, supplying a fueled length of 6.5 inches. The liner is cylindrical in shape, fabricated from Hastelloy-C tubing, with a wall thickness of 0.030 inch, and welded end caps of 0.060 inch thickness. The sealed liner fits within an outer cylindrical can or shell designed to withstand high pressures of sea water burial. The outer Hastelloy-C shell has a wall thickness of 0.400 inch. The end caps for the shell are of a trepan design with a cap thickness of 0.585 inch. The outer shell is encased within a thin titanium diffusion barrier for metallurgical compatibility between the sealed Hastelloy-C shell and the surrounding uranium-molybdenum shield. The entire capsule assembly is about 8 inches long.

The finished capsule must be capable of operating for 10 years and demonstrate materials compatibility, structural integrity, and resistance to both oxidation and corrosion. In addition, it must satisfy various nuclear safety requirements for structural, chemical, and metallurgical integrity against impact, sea water, and ground burial and fire.

The effort on this subtask was initiated early in the program and will continue throughout Phase I of the program.

Preliminary Capsule Design

1. Design Constraints

The overall objective of this effort is to design a radioisotope fuel capsule which meets all of the system hot cell assembly and nuclear safety constraints. The major nuclear safety constraint, which affects the capsule design, is sea water burial resulting from a transportation or operational accident. For this potential accident situation, ocean burial depths

of 20,000 feet can be postulated. This environment imposes two constraints upon the capsule design; sea water corrosion and external pressures approaching 10,000 psi.

The capsule hot cell assembly and fabrication operations also impose certain restrictions on the capsule design. To facilitate capsule assembly operations, a minimum diametral clearance of 20 mils is required between the capsule and each liner during fueling. In addition, ORNL recommends that weld penetration depths should be limited to 0.15 inch or less and the thickness of each capsule liner should be less than approximately 0.2 inch. Both of these requirements have been imposed to ease hot cell closure welding operations while maintaining acceptable closure welds free of thermal stress cracking.

2. Capsule Configuration Survey

Several capsule end cap configurations were investigated and discarded for various reasons. Among these were the spherical or parabolic head design. This design permits minimum cylinder and head wall thicknesses but requires an increase in capsule length due to the voided round end configuration. Capsule height could be decreased by utilizing the voided ends for fuel volume but at the necessity of developing contoured SrTiO_3 fuel forms. Contoured fuel form development for SrTiO_3 was not considered feasible for the subject design.

Another head configuration investigated was the "coolie hat" head design which consists of a thick cylindrical disk transitioning to a shallow cone on the outer face. This design also increased the overall capsule length beyond that of the selected model. The selected end cap is shown in Figure 3-62. It consists of a moderately thick flat head design. This design minimizes the overall capsule length and facilitates the transfer of heat through the capsule end cap.

Variations of the selected flat head model were investigated to determine the practicality of assuming internal fuel support and interaction of vessel walls for performance of the stress analysis. Two discarded models which bear mentioning are the dual wall capsule model and the radial fuel support model.

Figure 3-62. SNAP-23A Fuel Capsule

The dual wall model uses two equally thick vessels, one inner and one outer, acting together to provide containment. Although this model reduced the requirements for welding and material thicknesses, it introduced extreme dimensional tolerances to assure that the inner liner and outer capsule act in unison. Preliminary deflection calculations showed that with the 20 mil diametrical clearance, specified by ORNL for fueling, the outer capsule liner would fail before it deflected sufficiently to pick up support from the inner liner. From this calculation, it was concluded that a single thick walled outer capsule is mandatory. The second item considered was radial fuel support, however, diametral tolerance considerations also precluded the use of this concept.

As indicated above, the flat end cap design was selected. The detailed capsule dimensional requirements are based upon the stress analyses.

3. Stress Analysis

A detailed stress analysis has been performed to develop a capsule design which meets the 20,000 feet sea water burial environment of approximately 10,000 psi. The actual pressure environment is 9360 psi; however, for conservatism, 10,000 psi has been assumed.

The initial model used to establish the capsule structural design details was an elastic stress model. Using this model, the critical stress and moment equations⁴⁰ were identified. These equations were then programmed for the UNIVAC 1108 computer to facilitate parametric evaluation of the capsule as a function of head thickness (t_1) and cylindrical wall thickness (t_2). Figure 3-63a illustrates the critical stresses evaluated. The stresses in the end cap of the capsule may be established by superimposing the following loading conditions:

- 1) uniform pressure over a simply supported circular plate,
- 2) uniform moment at the boundary of a circular plate, and
- 3) uniform radial force at the boundary of the circular plate.

Stresses in the circular cylinder can also be determined by the superimposing of various loading conditions, namely:

- 1) uniform pressure over surface of cylinder,
- 2) uniform moment on the edge of the cylinder, and
- 3) uniform radial shear force at the edge of the cylinder.

Figure 3-63a. Critical Capsule Stresses

Before the stresses in the closed cylinder can be determined, using the above superposition technique, the relationship of the moment and shear with respect to the thicknesses of the end cap and the cylinder wall must be solved. The relation of the moment and edge shear load (compatibility equations) are determined by equating the angular and radial displacement of the two adjoining structures for the above loading conditions. An inspection of the stress and compatibility equations show that the discontinuity stresses resulting from the edge moment and shear load decrease exponentially. As a result, the maximum moment and shear occur at the interface of the end cap and the cylindrical liner.

The stress acting at the mid-point of the end cap is given by:

$$S_1 = \frac{6M}{t_h^2} - \frac{V}{t_h} - \frac{3PR^2}{8t_h^2} (3 + \mu) \quad (1)$$

Figure 3-63b. Stress Profile

where:

- S_1 = stress acting at mid-point of end cap, psi
 P = the external pressure, psi
 R = capsule radius, in.
 t_h = thickness
 V = transverse shear, lb per linear inch
 M = moment, in -lb per linear inch
 μ = Poisson's ratio

The discontinuity stress, S_2 , which acts in the weld region of the end cap is:

$$S_2 = \frac{6M}{t_h^2} - \frac{V}{t_h} \quad (2)$$

S_3 is a similar discontinuity stress which occurs at the cylindrical liner-weld interface and is described by:

$$S_3 = \frac{2VR\lambda_c}{t_c} - \frac{2M\lambda_c^2 R}{t_c} - \frac{PR}{t_c} \quad (3)$$

where

- t_c = thickness of cylindrical liner, in.
 λ_c = liner flexibility parameter

The hoop stress, S_4 acting at the center of the cylindrical liner is:

$$S_4 = \frac{1}{t_c} \left[2V(\lambda_c R e^{-\lambda_c X} \cos \lambda_c X) - PR - 2\lambda_c^2 R M e^{-\lambda_c X} (\cos \lambda_c X - \sin \lambda_c X) \right] \quad (4)$$

where

X = the axial distance from the cap-liner interface to the mid-point of the liner, in.

The compatibility equations for the moment and shear force at the joining structures used in the above equations are:

$$M = \left[\frac{\frac{2 PR^2 \lambda_c^3 E t_h D_c}{E t_c (1 - \frac{\mu}{2}) [E t_h + 2 R D_c \lambda_c^3 (1 - \mu)]} + \frac{PR^3 \lambda_c^2 D_c}{4 D_h (1 + \mu)}}{2 \lambda_c + \frac{2 R \lambda_c^2 D_c}{D_h (1 + \mu)} - \frac{\lambda_c E t_h}{E t_h + 2 D_c \lambda_c^3 R (1 - \mu)}} \right] \quad (5)$$

$$V = M \left(2 \lambda_c + \frac{2 R \lambda_c^2 D_c}{D_h (1 + \mu)} \right) - \frac{PR^3 \lambda_c^2 D_c}{4 D_h (1 + \mu)} \quad (6)$$

where flexibility parameters are:

$$D = \frac{E t^3}{12 (1 - \mu^2)} \quad (7)$$

$$\lambda = \frac{(3 (1 - \mu^2))^{1/4}}{R^2 t^2} \quad (8)$$

The parameter E is the modulus of elasticity and the subscripts h and c refer to the head and cylinder, respectively.

As indicated previously, the above equations were programmed for the UNIVAC 1108 computer. The code has been designated CAPSAP. Using this code, the stress profile for a large number of capsule configurations have been calculated. Cylinder wall thicknesses ranging from 0.050 inch to 1.0 inch and end cap thicknesses ranging from 0.5 inch to 0.7 inch have been evaluated. Figures 3-63, 64, and 65 include some typical data generated.

Figure 3-63b shows how the critical stresses, S_1 , S_2 , S_3 , and S_4 vary as a function of cylindrical wall thickness for a specified end cap thickness of 0.550 inch. The compressive and tensile yield strengths as well as the ultimate strength of Hastelloy-C have been superimposed on to the graph. To achieve an acceptable design, the four stresses must be below the yield strength of the selected capsule material simultaneously. It is apparent from the curves presented in Figure 3-63b that the simultaneous solution of these requirements is impossible for the selected end cap thickness. For the end cap thickness selected, 0.55 inch, a cylinder wall thickness of 0.325 inch appears to be the best compromise. For this configuration the maximum stress is reduced to approximately 91,000 psi. All four of the critical points, however, would have stress values exceeding the elastic limit of Hastelloy-C. When the stress exceeds the yield strength of the material, local yielding occurs and a redistribution of stresses results. At that point, the model used is not applicable.

Increasing the end cap thickness to 0.6 inch yields the stress profiles shown in Figure 3-64. From these data, it is apparent that the maximum stresses can be reduced to approximately 75,000 psi for a cylinder wall thickness of 0.355 inch. This is still unacceptable since the elastic limit of Hastelloy-C has been exceeded. Figure 3-65 illustrates stress profiles for capsules with a 0.700 inch end cap. It is apparent from these curves that capsules cylindrical thicknesses of 0.38 to 0.44 inch satisfy the elastic constraint.

Implicit in the assumed elastic model is the fact that the cylindrical liner and the end cap must form a continuous section. This means that full penetration welds are required at the end-cap liner interface. As indicated above, this is beyond present hot cell closure welding accomplishments. To avoid a costly and long term development program, ORNL recommends that the cylindrical wall thicknesses should be limited to approximately 0.2 inch and weld penetrations should be limited to 0.15 inch or less. It is apparent that both of the above restrictions have been violated.

Plastic Hinge Model .

The most critical of the closure welding constraints is the depth of weld penetration. To meet this constraint, the end cap must be judiciously designed to minimize stresses in the weld area. This can be accomplished by providing axial support by the fuel pellet. Axial

Figure 3-64. Stress Profile

Figure 3-65. Stress Profile

a) CAPSULE LOADING

b) DETAIL LOADING DOTTED LINE ON HEAD REPRESENTS DEFLECTION PROFILE

611852-28B

Figure 3-66. Fuel Capsule Loading and Details

fuel support can be obtained using shims. The deflection calculations associated with the CAPSAP elastic model show, however, that the end moment on the end cap is sufficiently large to prevent the cap from resting on the pellet in the weld area. In fact, assuming elastic deflection conditions prevail, only center support is achieved even with shims. It is apparent from these results that design innovations must be provided to reduce the moment in the weld area. To achieve the desired heat balance required to achieve defect free, structurally sound welds, a trepan can be provided in the end cap near the region to be welded. If properly designed, the reduction in the end cap thickness in the trepan area can be used to reduce both the moment and resulting stresses in the weld area. Under hydrostatic pressure of ocean burial, the stress in trepan area will be greater due to the reduced thickness. If the stress in this area is allowed to rise beyond the yield strength of the material, it will flow locally or deform plastically. Using this technique, the deflection profile of the end cap can be revised to pick up the desired fuel support while limiting the stress in the weld area. Figure 3-66a is the overall loading imposed on the capsule during sea water burial. Part b of the same figure is a detailed view of the loading and deflection profiles at various points.

The maximum moment transmitted into the end cap may be established by assuming that the cylindrical liner is fixed at each end as shown in Figure 3-67.

Figure 3-67. Fixed End Loading

For this condition, the maximum moment developed at the point of discontinuity is:

$$M_f = \frac{P}{2\lambda^2} \quad (9)$$

and the shear force is:

$$Q_f = \frac{P}{\lambda} \quad (10)$$

where the variables are as defined previously.

For a pressure loading of 10,000 psi, the maximum end moment developed for the subject Hastelloy-C capsule is 2740 inch-lb/linear inch. The shear force is 7440 lb/linear inch. Assuming that both the above moment and the shear force are transmitted to the end cap, the maximum stress developed in the region of the trepan is:

$$\sigma_h = \frac{6M_f}{t_p^2} + \frac{Q}{t_p} \quad (11)$$

where

t_p = thickness of end cap in the trepan, in.

The depth of the trepan required can be established by solving for t_p in equation (11) using the previously established moment and shear values, assuming an allowable stress for σ_h . Solving for t_p yields:

$$t_p = \frac{-Q}{2\sigma_p} \pm \frac{1}{2} \sqrt{\left(\frac{Q}{\sigma_p}\right)^2 + \frac{24M}{\sigma_p}} \quad (12)$$

As indicated previously, the end cap will be designed to yield in the area of the trepan. For conservatism it is assumed that the material is perfectly elastic-plastic. This means that the material behaves elastically up to its yield point and behaves in a completely plastic manner above this point. Hastelloy-C, however, retains some of its strength characteristics above its yield point. Now, to achieve a controlled deformation where only the outer fibers of the material exceed the yield point, the stress⁴² in the weld area must be no greater than 3/2 times the yield stress or 98,400 psi. Applying this stress value and solving for t_p in equation 12 yields a thickness of 0.435 inch.

A rough calculation was made to predict the amount of fuel support which is achieved just as the stress in the trepan approaches the yield strength. The maximum end moment at that point is:

$$M_o = \left(\sigma_y + \frac{Q}{t_p} \right) \frac{t_p^2}{6} \quad (13)$$

where:

$$\sigma_y = \text{the yield strength}$$

and

$$Q = 2 \lambda M_o \quad (14)$$

Substituting Equation (14) into (13) gives:

$$M_o = \frac{\sigma_y}{\frac{6}{t_p^2} + \frac{2\lambda}{t_p}} \quad (15)$$

In this case, the fibers on the compressive side of the head will reach the yield strength prior to the fibers under tension. The value of the moment at the point where the compressive fibers reach their yield strength is 1710 in.-lb per linear inch. Based upon Reference⁴¹ the $\frac{b}{a}$ ratio of fuel support is 0.4. The parameters "a" and "b" are defined in Figure 5b of the reference.

The thickness of the head within the hinge circle is conservatively calculated by using the rupture disc formula ⁴³. The loading condition assumed is a uniformly loaded circular plate with fixed ends which are allowed to deflect sufficiently to make the center and circumferential stress equal. For this condition:

$$t_h = \left[\frac{0.38 P z^2}{\sigma_y} \right]^{1/2} = 0.538 \text{ in.} \quad (16)$$

where

z is the radius of the end cap.

The final end cap thickness was increased to 0.585 inch to satisfy the welding heat balance requirement.

The weld must be analyzed to establish its ability to support the moments arising from the fixed end conditions discussed earlier. The depth of weld must also be established. Figure 3-68 is a free body diagram of the forces acting on the weld area.

Figure 3-68. Weld Free Body Diagram

Assuming no yielding in the cylinder wall, the end cap, and the horizontal foundation, and taking summation of moments about point A:

$$\Sigma M_A = 0 = M_f - F_w d \quad (17)$$

The resultant force acting on the weld is:

$$F_w = \frac{M_f}{d} = 5270 \text{ lb/linear inch} \quad (18)$$

The horizontal and vertical components of F_w are:

$$F_H = \frac{h}{d} F_w = 5180 \text{ lb/linear inch} \quad (19)$$

$$F_V = \frac{c}{d} F_w = 677 \text{ lb/linear inch} \quad (20)$$

The horizontal force F_H produces a tensile stress in the weld of:

$$\sigma_H = \frac{F_H}{A_w} = 51,800 \text{ psi} \quad (21)$$

where the weld area, A_w , is assumed to be a weld penetration of 0.1 inch over a one inch circumferential length.

The vertical force F_V produces a shear stress of

$$\sigma_S = \frac{F_V}{A_w} = 6,770 \text{ psi} \quad (22)$$

where:

$$A_w = 0.1 \text{ in.}^2 \text{ per linear inch}$$

Taking an incremental area as shown in Figure 3-69 and applying Mohr's circle, the maximum shear stress and the principle stresses are determined.

Figure 3-69. Incremental Weld Area

The maximum shear stress σ_{SM} is

$$\sigma_{SM} = \left[\left(\frac{\sigma_n}{2} \right)^2 + \sigma_s^2 \right]^{1/2} = 26,730 \text{ psi} \quad (23)$$

and the principle stress in tension, σ_n , is 52,630 psi.

To check the weld for the stress imposed on it prior to the yielding of the hinge, the free body diagram as shown in Figure 3-70 is used.

$$\begin{aligned} h &= 0.51 \text{ in.} \\ d &= 0.52 \text{ in.} \\ c &= 0.067 \text{ in.} \\ M_o &= 1710 \text{ in.-lb} \\ &\text{per linear inch} \end{aligned}$$

Figure 3-70. Weld Free Body Diagram

Using the conservative assumption that the reaction forces per linear inch share the load equally:

$$R_L + R_w = R_F \quad (24)$$

and the weld equally shares the load; i.e.,

$$R_L = R_w \quad (25)$$

the force per unit length of the reactions is

$$R_F = \frac{P (a^2 - (0.4a)^2)}{2\pi(a + .4a)} = 6,000 \text{ lb/linear inch} \quad (26)$$

Based upon Equations (24), (25) and (26):

$$R_w = 3,000 \text{ lb/linear inch}$$

The force F_w in the weld is determined by taking the summation of moments about point A.

Using the worst case value of $V_L = 6,000 \text{ lb/linear in.}$, the force F_w is:

$$\Sigma M_A = R_w c + F_w d - 1.35 V_L - M_o = 0 \quad (27)$$

Inserting the appropriate value and solving for F_w :

$$F_w = 4460 \text{ lb/linear inch}$$

The horizontal and vertical components of F_w are:

$$F_h = \frac{h}{d} F_w = 4380 \text{ lb/linear inch} \quad (28)$$

$$F_v = \frac{c}{d} F_w = 575 \text{ lb/linear inch} \quad (29)$$

Summing the horizontal and vertical forces in the weld

$$F_H = 4380 \text{ lb/linear inch}$$

$$F_V = F_v + R_w = 3575 \text{ lb/linear inch}$$

For an assumed weld area of $0.1 \text{ in.}^2/\text{linear inch}$, the tensile stress, σ_H , and the shear stress, σ_S , acting on the weld are:

$$\sigma_H = 43,800 \text{ psi}$$

$$\sigma_S = 35,750 \text{ psi}$$

Taking the incremental area as before and applying the stress vectors as in Figure 3-71, the principle shear and tensile stress are again found using Mohr's circle.

Figure 3-71. Incremental Weld Area

The principle shear stress σ_{SM} is 42,000 psi and the principle stress in tension, σ_n , is 63,900 psi.

The values determined above by assuming the reaction R_w in the weld shares the load equally with the ledge plus the worst case assumption that $V_L = 6000$ implies that the 0.1 inch weld penetration may be questionable. Using the same method or procedure, and calculating the principle stress for a weld penetration of 0.11 inch, the principle shear and tensile stresses are found to be:

$$\sigma_{SM} = 38,000 \text{ psi}$$

$$\sigma_n = 57,900 \text{ psi}$$

Based on the preceding calculations, the loading conditions imposed on the weld prior to the hinging action of the trepan appears to be the worst case loading condition. The weld

penetration depth of 0.135 inch as shown in Figure 3-62 is a design goal. The minimum acceptable weld penetration is 0.11 inch.

It is shown, in the analysis presented above, that considering the end cap alone, yielding will occur under the trepan before it occurs in the weld, thus protecting the weld from high stresses. A third mode of yielding is also possible, namely, yielding of the cylindrical wall of the capsule just below the weld where the wall thickness is reduced to 0.135 inch. Preliminary calculations indicate that this region represents a site of early yield, thus further protecting the weld and permitting the end cap to transfer load to the pellets. This mode of deformation will be analyzed further.

The hoop stress in the cylindrical liner at a distance sufficiently removed from the point of discontinuity so that the load effects of the shear and moment have been reduced to zero is described below. The maximum hoop stress considering the thick wall formula occurs at the inner surface of the cylinder and is:

$$\sigma_{\text{Hoop(max)}} = \frac{2 P (R_o^2)}{R_o^2 - R_i^2} \quad (\text{Reference 40}) \quad (30)$$

where:

$$\begin{aligned} R_o &= \text{outer radius of liner, 2.476 in.} \\ R_i &= \text{inner radius of liner, 2.076 in.} \\ P &= \text{external pressure, 10,000 psi} \end{aligned}$$

Substituting the appropriate numbers into Equation (30) gives a hoop stress of 67,000 psi.

The hoop stress at the midplane is:

$$\sigma_{\text{Hoop}} = \frac{P R_o^2 (R_i^2 + r^2)}{r^2 (R_o^2 - R_i^2)} \quad (\text{Reference 40}) \quad (31)$$

where:

$$r = 2.276 \text{ in.}$$

Evaluating the above gives:

$$\sigma_{\text{Hoop}} = 61,300 \text{ psi}$$

The fixed end condition has considerably more effect on the longitudinal stress. With fuel support taken into account, the highest localized longitudinal stress σ_1 , is:

$$\sigma_1 = \pm \frac{6M_f}{t_c^2} \pm \frac{1.932 Q_f}{\beta_{tc}^2} - P \quad (32)$$

or:

$$\sigma_{1(\max)} = -45,500 \text{ psi}$$

4. Weld Configurations

In establishing the design criteria, weld configurations were evaluated for the numerous proposed design concepts and materials. Welding parameters were considered based on hot cell welding techniques, thermal stress cracking, and uniform penetration requirements. The selected designs (liner and capsule) incorporate weld configurations considered to be applicable to remote weld operations. A uniform volume of material is present in the immediate weld area in both the capsule side wall and the end cap due to the side wall configuration and the end cap trepan design. The trepan design allows a uniform heat distribution during welding, thus reducing thermal stresses and affording uniform penetration. Figure 3-72 summarizes the preliminary capsule design.

5. Tolerance Study

The objectives of the dimensional tolerance study were to establish preliminary fuel pellet, liner, and capsule dimensions and tolerances (dimensional and form) that would assure positive internal fuel support, as required by the fuel capsule stress analysis. In addition, this study was performed to facilitate fuel capsule remote assembly and minimize assembly difficulties.

Dimensional and form tolerances were determined for the fuel pellet. A diametral tolerance of $\pm .000, - .005$ inch was established for the fuel pellet in order to assure a single direction deviation from the design fuel pellet diameter. Parallelism of the pellet end faces

NOTES:

- 1 HOT CONDITION DIAMETERS AND LENGTHS DETERMINED FOR $\Delta T = 1000^\circ F$
- 2 COLD CONDITION DIAMETERS BASED ON ORNL REQUIREMENT OF .020 DIAMETRAL CLEARANCE FOR THE FOLLOWING ASSEMBLIES:
 - a HOT FUEL PELLET INTO COLD LINER
 - b HOT LINER ASSEMBLY INTO COLD CAPSULE
 - c HOT CAPSULE ASSEMBLY INTO COLD DIFFUSION BARRIER-SHIELD ASSEMBLY

Figure 3-72. SNAP-23A Fuel Capsule Interface Dimensions

and perpendicularity of the end faces to the pellet side wall were established to be within 0.005 inch. This tolerance provides a maximum pellet end face incline which corresponds to an angle of $0^\circ - 5$ inches or $1/12$ degree incline. In order to assure internal fuel support, various wedge shaped shims may be inserted. Assuming the case of a double fuel pellet stock, a maximum of four 0.005 inch wedge shaped shims could be required. Additional flat shim stock may be inserted to attain any given fuel height dimension. The result indicates that column support of the fuel is considered feasible particularly when shim insertion is considered in obtaining a predetermined height.

The ORNL diametral clearance requirement of a 0.020 inch minimum for remote assembly was incorporated into all phases of the dimensional tolerance study. Since ORNL has not established a fuel pellet length tolerance, a conservative tolerance of $+ .000, - .100$ inch was selected for the tolerance study.

The liner and capsule length dimensions at operating temperature were determined by the standard maximum-minimum dimensional stack-up method assuming that all resulting deviations will be eliminated through the use of shim stock.

Thermal expansion values were calculated assuming a Δt of 1000°F in order to determine operating temperature dimensions. Machined or room temperature dimensions will be determined to assure correct component stock-up for a precise temperature differential at each surface in question. The precise temperature profile has not been firmly established.

The liner and capsule diameters were determined allowing the required 0.020 inch minimum diametral clearance between the "at temperature" liner and between the "at temperature" fueled liner assembly and the "room temperature" capsule. Figure 3-72 summarizes the results of the tolerance study.

6. Materials Evaluation

A brief study was initiated to investigate the corrosion resistance and compatibility of the various heat source materials. There is ample evidence that Hastelloy-C possesses exceptional stability in sea water. USNRDL has recently measured a corrosion rate of 5×10^{-5}

mils per year by radioactive tracer techniques³¹, and 2×10^{-2} mils per year by electro-chemical measurements.³² International Nickel Company has exposed samples of this alloy to sea water for years without measurable corrosion.

Similarly, the corrosion rate of unalloyed titanium in sea water is about 2×10^{-2} mils per year. Several titanium alloys, including Ti-6 Al-4V, Ti-5 Al-2.5 Sn and Ti-3 Al-11 Cr-13V, were completely resistant to deep ocean exposures for times up to 197 days in stressed or unstressed conditions³³.

Quantitative data on the corrosion of U-8 Mo alloy is being obtained. This alloy is not believed to be exceptionally corrosion resistant. The compatibility of the titanium with Hastelloy-C and U-8 Mo is also being considered. Also under study are the possible galvanic effects that might influence corrosion rates should the multi-metal generator be submerged in sea water. These results will be included in future reports.

Capsule Development

1. Pellet Fabrication Liaison

Liaison was initiated early in this period with Oak Ridge National Laboratory on fuel pellet design and fabrication capabilities, current technology, materials compatibility, etc.

Oak Ridge personnel felt that the four inch diameter pellets of the reference design can be fabricated by extrapolation of current technology. This size, however, is larger than any strontium titanate pellets which they have made to date. They are proceeding with the required development effort to supply finished pellets for this program.

The current technique for titanate pellet fabrication consists of hot pressing strontium titanate powder. To minimize contamination, this pressing is performed in platinum sleeves and caps. Oak Ridge personnel suggest two individual pellets to obtain the 6.5 inch of fuel per capsule.

2. Procurement and Capsule Materials

Capsule material procurement activities were initiated early in the program. A complete review was made to establish potential suppliers of Hastelloy-C bar, sheet, plate, and tubing stocks. Due to a six month plant shutdown at Stellite Division of Union Carbide Corporation (the only known U. S. supplier of this material), available stock of this alloy is very scarce and delivery schedules for new material are long. The review related to availability, cost, and delivery schedules. All known or potential United States and foreign suppliers were contacted, including: Eastern Stainless Steel, Universal Cyclops, Alvac, Continental Metals, Alleghany-Ludlum, Armco Steel, A. D. Mackay, Frasse, Whitehead, Mitsubishi International (Japan), Samuel Osborne (England), Compagnie-Des A Teliers Et Forges La Loire (France), and all known East and West Coast warehouses. The only organizations that could or were willing to supply the material were Stellite Division of Union Carbide and Mitsubishi International.

Some small pieces of stock of interest to this program were found through a warehouse supplier. This material was ordered for welding studies.

Following the completion of the preliminary mechanical design and the review of component manufacture methods, Hastelloy-C stock was ordered for the liner, shell walls and the end caps. All of the stock was ordered from Stellite Division on comparable delivery schedules, lowest cost and under the Buy American Act.

3. Component Fabrication Evaluation

In the second month of this period, work was initiated on an evaluation of feasible methods for forming, machining, and/or fabricating the liner and shell components for the SNAP-23A capsule. Four available methods were evaluated. These are described and discussed below.

(a) Casting the Liner and Shell Bodies (Including Bottom Ends).

This method was immediately rejected for this application. Casting (investment or otherwise) will contain pores which would be the source of leakage. In addition, castings, as a class, have poorer strength characteristics than wrought stock. Methods

(b) Boring the Center Out of Solid Bar Stock.

Boring the capsule out of solid stock appears initially to be one of the simplest methods of fabrication. Depending upon the radius allowed, capsules could be bored with or without solid bottoms. However, with a four inch diameter requirement, this is a wasteful method to obtain cylinders with 0.030 and 0.400 inch thick walls since the majority of the stock must be machined away. This material is difficult to machine and long machining times would be required for each component. An evaluation of the potential suppliers of the required bar stock showed that long delivery schedules (at least 30 weeks) would be encountered due to the bar size (at least 5 inches in diameter) and the difficulty of machining. This method does not appear attractive as an easy or economical method of manufacture even for the first few components required for Phase I studies. The long term procurement is even less attractive for off-the-shelf components.

(c) Seamless Pipe with Welded End Caps.

The use of seamless pipes is quite attractive due to the known structural stability of seamless pipe. However, the sizes required for this application are not available from commercial sources. The largest seamless pipe made by Stellite Division of Union Carbide Corporation is No. 3 Schedule 40 with an 0.216 inch thick wall and 3.500 inch outside diameter. Both dimensions are below the requirements for the outer shell and the diameter is too small for the inner liner. Checks with other suppliers were unsuccessful. Developmental seamless tube would be very expensive and of unknown reliability.

(d) Welded Pipe with Welded End Caps.

The subject method is second in preference to the seamless pipe concept since the longitudinal weld joint is a potential source of leakage or failure. It is believed, however, that mechanical work and heat treating, followed by 100 percent inspection of the weld area, will yield acceptable components. Again, the size of pipe required for this job was not available as an off-the-shelf or normal procurement item. The largest size pipe in this category is No. 4 Schedule 80 (4.500 inch outside diameter with an 0.337 inch thick wall). These dimensional sizes can be supplied by a reputable firm within reasonable delivery

schedule (approximately 4 weeks following delivery of plate stock). As a result, this fabrication technique was selected and the material has been ordered. Delivery of the liner and shell units are scheduled for mid-September 1967.

Following manufacture of the Hastelloy-C pipe, final machining of outside and inside pipe diameters will be required to maintain the close tolerances required by the preliminary design. In addition, end caps for both the liner and shell cans must be machined to shape and size. Evaluation of potential machining contractors is now in progress.

4. Fabrication of Capsule Subassemblies

This section of the subtask relates to the welding of end caps in the bottoms of the inner liner and outer shell. Work in this area was initiated during the latter portion of this reporting period.

(a) Equipment Design, Procurement, and Setup.

Work was initiated on layout and setup of the tooling, jigs, and equipment required for this phase. The initial work related to assembly and checkout of available units for flat plate stock welding from jigs available in-house. Some additional tooling, chills, and parts are being ordered.

Work was also started on the design, procurement, and setup of cylindrical welding equipment for actual liner and shell welds.

(b) Flat Plate Welding Studies.

Late in the period, welding studies were initiated on available flat plate stock to establish preliminary welding parameters prior to delivery of cylindrical parts (not due until mid-September). This study will be used to establish required welding speed, current, electrode size and arc gap, joint configuration, and weld properties.

4. COMPONENT DEVELOPMENT

Subtask 410 - Heat Source Materials Test Program

Summary Objectives

The design of the heat source requires the unique combination of a number of materials to operate at high temperatures for a ten-year period. It is, therefore, not surprising that a deficiency of data concerning the behavior of these materials under the required systems conditions exists. Under Subtask 360, the function of each component is described and the criteria for material selection are given along with potential problem areas. In most cases, the potential problems involve areas where data are incomplete or non-existent. The objective of this subtask is to plan and conduct heat source materials properties evaluation tests to develop mechanical and physical properties, oxidation, and compatibility data for system design to permit ten-year operation of the heat source in the 900°F to 1400°F temperature range.

Program Status

The major areas of concern with regard to the heat source involve the oxidation behavior as well as the mechanical and thermophysical properties of the U-8 w/o Mo shield material and the compatibility of each material at each interface along the heat path in the heat source.

The shield material, U-8 Mo, undergoes a phase transformation as its operating temperature drops from 1340°F at BOL to less than a 1000°F at EOL. When the shield temperature drops below 1064°F, the eutectoid temperature for the U-Mo system, transformation of the high temperature $\gamma + \alpha$ takes place. Changes in mechanical and physical properties very often accompany such transformations. In order to characterize the mechanical behavior of the U-8 Mo in the 900°F to 1400°F temperature range, tensile and creep rupture tests will be conducted for material in the γ and $\gamma' + \alpha$ condition. These conditions can be attained by proper heat treatment and quenching procedures derived from existing TTT curves as shown in Figure 4-1.

Figure 4-1. TTT Diagrams for the U-8 Mo Alloy Quenched to Temperature from 900°C

Accompanying the phase change in the U-8 Mo system is a volume change, the extent of this is temperature dependent. Dilatometric studies are required to completely define the volume change as it occurs during the life of the system. Because of the phase change, thermal expansion data in the literature are uncertain and require verification.

Because the shield lies directly in the heat path, thermal conductivity data for the γ and $\gamma' + \alpha$ phases in 900°F to 1400°F temperature range are a vital necessity for the system design.

At each interface in the heat path, a potential compatibility problem exists. The incompatibility of uranium and Hastelloy-C was recognized and a suitable barrier material was selected to prevent catastrophic failure of the capsule. At each interface, the extent of material interactions must be determined to permit prediction of conditions as the life of the system approaches ten years.

Of the materials which make up the heat source assembly the one most susceptible to corrosion by oxidation is the U-8 Mo shield. Although, essentially, this component is hermetically sealed in the system during operation, and oxidation should not occur, it is possible that serious oxidation could occur due to the combination of air exposure and temperature increase when assembled with the isotope capsule. It is necessary, therefore, to characterize the oxidation behaviour of U-Mo alloy in the temperature range of possible exposure to determine both the tolerable time and temperature limits.

Materials Test Program

1. Mechanical Properties Evaluation

As-cast bars of U-8 Mo will be machined to round standard ASTM tensile specimens. A number of specimens will be heat treated to produce a $\gamma' + \alpha$ phase microstructure. Specimens of the γ and $\gamma' + \alpha$ phase will be tensile in an inert atmosphere in the temperature range room temperature to 1400°F at 200° incremental steps.

Similarly creep rupture specimens will be prepared and tests will be conducted at 900°, 1000°, 1200° and 1400°F using four stress levels to characterize the creep rupture behavior of the two metallurgical conditions of U-8 Mo. Tests will be conducted in vacuum to minimize contamination from the atmosphere.

Quotations have been solicited and an order for fifty machined tensile and creep specimens of as-cast U-8 Mo has been placed with National Lead. Initial delivery of specimens is scheduled during the next quarter of this program. Testing will begin immediately upon receipt of materials.

2. Thermophysical Properties

Thermal conductivity of U-8 Mo will be measured using the "comparison bar" technique. The technique consists of measuring the temperature gradient along the test bar and determining the heat flux through the test bar by using standards with known conductivity at each end of the test bar. With the heat flux known for a given temperature gradient, the thermal conductivity of the test material can be determined.

Dilatometric measurements will be used to determine the thermal expansion and the dimensional stability of the U-8 Mo. Isothermal dilatometric measurements will be used to determine the volume change in U-8 Mo as it transforms from γ to $\gamma' + \alpha$ in the temperature range 900° to 1085°F. Dilatometric measurements will also be used to determine the thermal expansion of γ and $\gamma' + \alpha$ U-8 Mo from room temperature to 1400°F.

These tests will be initiated upon receipt of material which is expected during the beginning of the next quarter.

3. Oxidation Behavior

The oxidation behavior of the U-Mo alloy will be characterized by determining the weight gain per unit area and the description of the oxide film as a function of time and temperature.

Because of the difficulty in ordering and the long delivery times for U-8 Mo, readily available material in the form of hot rolled U-10 Mo bar has been substituted for this test. It is believed that the oxidation behavior of the U-8 Mo alloy will not vary significantly from those obtained using material with slightly higher Mo content.

Metallographic characterization of the substitute alloy bar has been completed. The material is essentially single phase, fine grained and equiaxed as shown in part A, Figure 4-2. Some stringer, presumably oxides, oriented in the bar rolling direction, are present. Examples of these are shown in part of Figure 4-2. Hardness, resistivity and X-ray diffractometer measurements on the bar all correspond with literature values describing the metallurgical condition as being 100 percent metastable gamma, i.e., the single phase structure is body-centered-cubic (gamma) a condition wherein this elevated-temperature-stable allotrope is retained at room temperature by rapid cooling. Rapid cooling of the cast U-8 Mo shield should result in retention of the metastable gamma condition for this component, and a fine grain structure should develop due to chill cooling in the vicinity of the casting surfaces. Therefore, in these respects (fine grain and crystal structure) the U-10 Mo oxidation test material corresponds to that most likely to be present in the shield.

Oxidation coupons, nominally 1/4 inch thick by 3/4 inch diameter have been cut from the U-10 Mo bar. To keep the sample surface condition consistent throughout the tests all coupons have been hand lapped on No. 320 emery and ultrasonically cleaned in methanol. Tests are being run in horizontally positioned tubular resistance furnace retorts open at one end, with the degree of oxidation studied principally by determining the gain in sample weight for given time-temperature exposure conditions. Air movement is minimal. Thus far preliminary data has been gathered for exposure times of from 1 to 7 days at temperature in the range 100 to 1112°F. These results are entered in Table 4-1.

Oxidation appears to be very slight below about 390°F for up to 7 days exposure while catastrophic reaction occurs in roughly 1-2 days at temperatures above about 930°F.

(a) Grain Structure

Longitudinal View 100X

Transverse View 100X

(b) Inclusions

Longitudinal View 100X

Longitudinal View 500X

Figure 4-2. Microstructure of U-10 w/o Mo Alloy Used for Oxidation Studies

A black scale develops at exposure times of 25 hours or longer at 425°F or higher, and becomes non-adherent to the alloy somewhere in the .002 to .0113 gram oxide/square centimeter reaction range. Photographs of test coupons representing the types of scaling observed are given in Figure 4-3. Further examination of these samples and additional oxidation testing are planned.

TABLE 4-1 PRELIMINARY STAGNANT AIR OXIDATION DATA ON U-10 Mo ALLOY

Oxidation Conditions Time (hrs) Temp (°F)		Weight Grain/Unit Area (mgms/cm ²)	Manner of Oxidation
64.5	266	less than 1	slight blue surface discoloration development on the samples.
112.5	302	less than 1	
168.5	346	less than 1	
294.5	360	less than 1	
25.0	428	2	Deep black adherent, oxidation product developed.
50.8	482	8	
100.0	536	20	
64.5	616	12	
112.5	680	113	Deep black non-adherent oxidation product developed.
168.3	788	560	
25.0	932	141	
50.8	1022	-	Catastrophic oxidation occurred with complete sample disintegration.
100.0	1112	-	

4. Compatibility

To determine the compatibility of materials which make up the heat path of the heat source assembly, tests will be conducted using washer type material specimens clamped together in series as they appear in the heat source. The test specimen composites will be

Unexposed

100 hrs/536°F
Adherent Scale

25 hrs/932°F
Non-Adherent Scale

100 hrs/1112°F
Catastrophic Oxidation

Figure 4-3. Photographs of U-10 w/o Mo Oxidation Coupons

sealed in quartz capsules which have been evacuated and backfilled with an inert gas. Encapsulated test specimens will be exposed for 100, 500, 1000, and 10,000 hours at 900°, 1000°, 1200° and 1400°F. Metallographic examination of the interdiffusion zone will be conducted. The rate of growth of the interdiffusion zones will be determined as a function of time and temperature to permit prediction of interfacial conditions as the life of the system approaches ten years. The material combinations which will be evaluated include Hastelloy C/Ti/U-8 Mo/Cu/Inconel 718. In addition, Hastelloy C/Ta/U-8 Mo and Hastelloy C/W/U-8 Mo will also be investigated to provide back-up data in the event the titanium barrier between the Hastelloy C and U-8 Mo proves ineffective.

Due to the urgency of initiating these long term tests and the previously discussed problem of procurement of U-8 Mo, the U-10 Mo material has also been substituted in this test program. It is believed that the test results obtained using the alloy with the slightly higher Mo content will not vary significantly from those expected using the U-8 Mo composition. Thick washers approximately 3/8 inch thick of each heat source material has been machined to size and composite test specimens are being assembled on Monel bolts which will be torqued to provide contact pressure at each interface during the initial stages of the compatibility test. The composite specimens will be encapsulated in a quartz tube with a partial pressure of argon to prevent reaction with the atmosphere during exposure.

Four composite sets of material washers will be exposed to air at a moderate temperature (approximately 400°F) prior to assembly and thermal exposure to evaluate the effect of oxidation at the interfaces on the interdiffusion process. These specimens will be thermally exposed at 1400°F for 100, 500, 1000 and 10,000 hours to fully assess the effect of the oxide film on the compatibility of the various heat source materials.

Subtask 420 - Insulation Assembly Materials Test Program

Summary of Objectives

The efficiency of the thermal insulation assembly has a pronounced effect on the physical size of the SNAP-23A system. Under subtask 360, the function of each component is described and materials selection criteria are given with the potential problem areas. In the case of the insulating material, potential problems concern areas where data are incomplete or non-existent. The objective of this subtask is to plan and conduct insulation materials processing and property evaluation tests to develop mechanical and thermophysical properties data for system design to permit ten-year operation in the temperature range 200° to 1200°F.

Program Status

Min-K-2000 is produced commercially in sheet form, 2 x 18 x 36 inch. To utilize this material in the SNAP-23 insulation assembly, shaped segments must be cut from the sheet, heat treated, fitted together, final machined, and then installed in the insulation container. This assembly procedure assumes that the thermal conductivity of the Min-K-2000 is isotropic; this however, may not be the case since the sheet is manufactured by pressing normally to the large dimensional plane of the sheet. No data are available to indicate whether the thermal conductivity of the material is orientation dependent. This determination will be made as part of this program.

Thermal conductivity data published for Min-K-2000 with inert gas backfill are calculated values. Procedures for evacuating entrapped air in the pores of the insulation and replacing it with inert gas atoms have not been developed. The feasibility of such a procedure has yet to be determined. As part of this program, the feasibility of evacuating and backfilling Min-K-2000 with an inert gas will be investigated.

Because of continued interest in the load carrying ability of Min-K-2000 from a design consideration, the dimensional stability of Min-K-2000 as a function of compressive load will be determined.

Two boards of Min-K-2000 have been purchased and heat treated according to WANL specifications. Because of the laminar appearance of the heat treated material, it is suspected that properties may vary somewhat with orientation. This condition will be taken into consideration in the test program.

Test Program

1. Thermal Conductivity

The thermal conductivity of heat treated Min-K-2000 in air in the temperature range 200° to 1200° F will be experimentally determined for directions perpendicular and parallel to the pressing direction. Thermal conductivity will be determined using the guarded hot plate method as described in the ASTM specification ASTM-177-C-63. While at the highest temperature the thermal conductivity apparatus will be evacuated to the best attainable vacuum and the thermal conductivity will be measured. Krypton will then be introduced into the system to $1/3$, $2/3$ and one atmosphere; the thermal conductivity at each pressure will be measured. Thermal conductivity of modified Min-K-2000 will be determined by the procedure described above.

Thermal conductivity samples as shown in Figure 4-4 have been cut from Min-K-2000 slabs heat treated according to WANL specifications. Solid discs $1/2$ inch thick by 8 inches in diameter were cut by band sawing and hand finishing to ensure uniform surface and specimen thickness. Samples of identical overall dimensions composed of butted band-saw slices of material were produced for measurement transverse to the pressing direction.

Testing is currently in progress. To date, preliminary data has been generated for thermal conductivity in the pressing direction up to approximately 600° F. These results are entered in Table 4-2 where comparison is made with data published by Johns-Manville. For practical purposes the preliminary conductivity value determined by Westinghouse is constant

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Figure 4-4. Sectioning of Min-K-2000 Board to Produce Thermal Conductivity Samples Both Parallel and Perpendicular to the Material Processing Direction

over the temperature range from 75° to 625°F, averaging approximately .023 BTU/hr-ft °F. A best comparison of this value can be obtained at 625°F, since extrapolation was necessary to yield comparison data for lower temperatures. The results check within 10 percent at this temperature.

TABLE 4-2. A COMPARISON OF PRELIMINARY THERMAL CONDUCTIVITY DATA OBTAINED AT WESTINGHOUSE R AND D CENTER WITH THAT PUBLISHED BY JOHNS-MANVILLE

Westinghouse Data		Comparison Data*	
Temperature (°F)	K BTU/hr-ft °F	Temperature (°F)	K BTU/hr-ft °F
149	.0225	150	.0178
212	.0210	200	.0190**
247	.0225	250	.0192
364	.0245	350	.0196
472	.0245	475	.0202
625	.0225	625	.0210

* Johns-Manville Aerospace Products Thermophysical Data Sheet

** Obtained by extrapolating the published curve.

2. Mechanical Properties

To evaluate the dimensional stability of Min-K-2000 under load, compressive creep tests will be conducted to determine the deformation characteristics as a function of temperature. Tests will consist of the application of a compressive load and the measurement of creep as a function of time. At each given temperature, a number of stress levels will be evaluated to determine the stress dependence of the observed creep. Tests will be conducted in the low (600°F) intermediate (600° to 900°F) and high temperature (900° to 1200°F) ranges.

Compression test blocks, 1 x 1 x 2 inch, representing material directions, both perpendicular and parallel to the stress axis have been cut from heat treated Min-K-2000 slabs. Compressive creep testing is currently in progress.

Subtask 430 - Radiator Thermal Tests

Radiator thermal testing is required to evaluate the performance of the radiator of BOL and EOL. Work started on March 15 and should be completed by October 1, 1967.

Chromalox industrial electric ring heaters were tested at 1000 watts for 16 hours and have been selected as heating elements for radiator testing. Four heaters will supply 750 to 1000 watts of heat necessary to simulate EOL and BOL operating conditions. These ring heaters will be placed in a stainless steel housing with an insulated core. Power and internal instrumentation leads will be routed to the center and then out the bottom of the housing. The housing will be placed in a rain shield configuration to simulate the thermoelectric converter in the proposed SNAP-23A system. Guard heaters will be incorporated to ensure the proper heat flux at the radiator inner ring.

The radiator and outer cover for radiator testing will be supplied by Mechanical Design. Power conditioning will be supplied by Electrical Design. Engineering Mechanics will provide the simulated thermoelectric converter, rain shield, and test fixture.

Instrumentation in the housing will be adjacent to the radiator inner hub instrumentation so temperature drop across the interface can be measured. Additional instrumentation will be incorporated to monitor heater performance and heat losses from the housing to the test fixture. This heater design is in the final layout stage. Test fixture design will be completed pending final mechanical design of the system.

To eliminate the possibility of a.c. thermocouple interference, d. c. power has been selected for the heaters.

Subtask 440 - Power Conditioning System

The objective of this subtask is to conduct developmental tests on the component parts and mock-up assembly of the power conditioning system.

The work done in Subtask 440 on the SNAP 23A Program has been directed to the construction of prototype modules for the electrical power conditioning system and a conceptual physical layout for the E.P.C. system, including the system cabling.

The electrical power conditioning system will consist of eight separate packages. Four of the packages are power transistors and the remaining four are modules containing the pre-regulator junctions. The eight packages are alternately mounted on the three inch shroud at 45 degree intervals. A sketch of the cabling system is shown in Figure 4-5. Provision is made for six thermocouples to be carried via the harness and the external connector for diagnostic purposes on the deliverable equipment. These may not be routed in this manner on the prototype units.

Preliminary pre-regulator module layouts have been made and a prototype unit produced. The module layout has been modified as a result of the original prototype, and modules are being made to substitute into the breadboard unit. The size of the module is 1 x 1 x 2 inch.

Unit testing was performed on one leg of the breadboard "quad" arrangement. The entire quad breadboard was then completely assembled and tests on the overall system are now in progress. At present the unit is being temperature cycled to determine long term stability.

NOTE: THERMOCOUPLE LEADS FROM T. E. TO EXT. CONN. AND POWER LEADS FROM T. E. TO EXT. CONN. IN RADIAL PATH. (THE LEADS WILL BE FURNISHED AS PART OF T. E., HARNESSSED AND RUN DIRECTLY TO EXT. CONN.) POWER LEADS FROM EXT. CONN. CIRCUMFERENTIAL CONN. TO R₁ THRU R₄ AND T₁ THRU T₄ AS SHOWN. CONNECTION WILL BE MADE TO THE POWER CONNECTOR AT BOTH ENDS OF 360 DEGREE RUNS.

(LEAD SIZE - 20 GAUGE)

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Figure 4-5. Physical Wiring Layout SNAP-23A

Subtask 470 - Component Development Engineering Follow-up

Radiator Subassembly Thermal Performance Test

Instrumentation and test condition requirements have been established for the radiator subassembly thermal performance test. This test will provide a means not only for experimentally verifying the radiator thermal performance at the design point but will be an aid in correlating measured temperatures with the thermal analysis for evaluation of off-design points.

Provisions have been made in the radiator subassembly design to accommodate the required thermocouple instrumentation. Small holes will be bottom-drilled in pre-selected fins before the fin assembly is brazed. These holes would be difficult if not impossible to provide on the brazed assembly due to the relatively close fin spacing. Layout work was performed to establish the routing of thermocouple leads.

On the basis of previously transmitted requirements for the radiator thermal performance test, decisions were made pertaining to the heater assembly design required to simulate the thermoelectric converter heat dissipation characteristic. The external geometry of the heater assembly will be basically that of the thermoelectric converter unit to simulate converter-radiator interface conditions. Electric heaters will be internally mounted within the heater assembly to duplicate the waste heat rejection rate of the converter. Redundant heaters and heater circuitry will be provided to minimize the probability of test interruption in the event of heater burnout.

60 - Watt Systems Mockup Environmental Test

Specifications are being established for the fuel capsule simulator to be used in the 60-watt systems mockup test. The fuel capsule simulator will reproduce the fuel capsule external geometry and material, and contain electric heaters to duplicate the heat output of the radioisotope fuel capsule assembly (1132 watts). After installation of the capsule simulator (heater assembly) into the heat source subassembly, hermetic sealing and backfilling with helium will be accomplished.

Instrumentation requirements with regard to the required number and location of thermocouples has been established. Studies were initiated during May to establish routing of the thermocouple leads and define special provisions that must be made in the design of the system components to accommodate the required instrumentation. It is presently anticipated that metal sheathed thermocouples will be used and that a major portion of the leads will penetrate through bosses located at the top of the heat source sub-assembly below the neck portion. In this scheme, hermetic sealing through the closure would be accomplished by brazing the metal sheathed leads to a disk-like closure member which in turn will be brazed or welded to the feed-through boss. It is presently anticipated that the heat source heater power leads will be brought out through ceramic type hermetic feed-throughs.

5. SYSTEM DEVELOPMENT

Subtask 510 - System Development

Sequential Assembly Tests

The sequential assembly tests will consist of a component-by-component assembly from the heat source outward with a thermal survey at every boundary. Work started on March 16, 1967 and should be completed by January 15, 1968.

Conceptual heat source design is in progress. Six-inch Watlow Firerod cartridge heaters will be used to build up two or three circuits of heaters. These cartridge heaters will be housed in a stainless steel cylinder with holes to accommodate each of the cartridge heaters. Ten gauge nickel wire will be used in the circuit in all high temperature areas ($T = 500^{\circ}\text{F}$). The heaters sealed in the stainless steel cylinder will be operated in an inert atmosphere.

The layout of this design will begin shortly pending finalized design of the heat source region of SNAP-23A and accurate definition of assembly procedures.

Initial instrumentation studies concerning sequential assembly testing were completed jointly by Thermal Analysis and Engineering Mechanics. The results of this study were submitted to Mechanical Design.

Finalized instrumentation definition will be completed pending information on assembly procedures and a response from Mechanical Design on the initial instrumentation study.

Thermal Contact Resistance

The design for the thermal contact resistance apparatus (Dwg. WT701681) has been completed and procurement is currently in progress. For reasons of safety and lack of test versatility, planned use of the existing test apparatus which measures thermal contact resistance was abandoned. This structure was not designed for the pressure range of interest. In addition, the number of modifications to change the apparatus from measuring the number of parameters of the planned tests make such an effort impractical.

A final decision was made concerning the choice of the material for the heat meters. These meters are the items which allow the determination of the heat flux across the contact surfaces. The restrictions imposed by the generality of the proposed testing requires the meter material to possess a small oxidation rate for the test temperature range, a high melting point, known thermal conductivity values for the test temperature range, and several other properties. The material chosen is Hastelloy R-235 produced by the Union Carbide Corporation.

6. SAFETY ANALYSIS

Subtask 610 - Nuclear Safety

The objectives of the nuclear safety effort are to:

- 1) Perform an operations analysis of the SNAP-23A generator from component manufacture to the end-of-mission recovery.
- 2) Develop comprehensive nuclear safety criteria for AEC approval
- 3) Perform a safety analysis of the SNAP-23A 60 watt(e) design.
- 4) Define the Phase II program plan for performance of a comprehensive safety analysis and test program.
- 5) Prepare a preliminary SNAP-23A safety report summarizing the results of the Phase I safety effort.

Operations Analysis and Nuclear Safety Design Criteria

An operations analysis of the 60 w(e) SNAP-23A generator was completed and is contained in Hittman Report HIT-248. The analysis supplies the background information for the nuclear safety criteria which are presented in the same report. The philosophy used in defining criteria was that of arriving at goals applicable to the complete heat source or generator. If the final assemblies can be shown to conform with the criteria, then the SNAP-23A generator will be classified as a safe system having maximum mobility and minimum dependence on handling or operating procedures. The preliminary safety criteria evolved are summarized below.

1. Fuel Capsule

- 1) The complete fuel inventory of the SNAP-23A generator will be absolutely contained within the Hastelloy-C fuel capsule during normal operation and after a maximum credible accident.
- 2) The unshielded fuel capsule shall not fail structurally under externally applied hydrostatic pressure equivalent to 20,000 feet of seawater for a period of 280 years. The compressive strength of the fuel pellets will be accounted for in the capsule structural analysis.

2. Heat Source Assembly

- 1) Shipment of the heat source assembly will be via truck or railroad.
- 2) Shipment of the heat source assembly will conform to regulations set forth by the Atomic Energy Commission and the Interstate Commerce Commission^{34, 35}. It will be possible to demonstrate analytically that the hypothetical accident conditions posed in Appendix B of Reference 34 can be met. These conditions include the following:
 - (a) 30 foot free drop onto flat rigid surface
 - (b) 40 inch free drop onto mandrel
 - (c) exposure to 30 minute 1475°F fire (this requirement is less severe than criterion 6 below)
- 3) In all instances of impact, explosion, and fire accidents, the fuel capsule and shielding will be considered an integral unit. For cases other than water burial, potential damage to the fuel capsule will reflect the protection provided it by the outer layers of the heat source.
- 4) The heat source enclosed in a shipping container will withstand vehicle impacts at net velocities up to 150 ft/sec. The complete fuel inventory will remain contained, and radiation dose rates after impact will be less than 1000 mrem/hr at three feet from the external surface of the container.
- 5) The heat source assembly will withstand shock overpressures up to 28,000 psi without loss of radioisotope fuel containment or rise in dose rate levels beyond 1000 mrem/hr at three feet from the external surface of the assembly. The overpressure will be a positive square wave pulse with a duration of 0.050 second.
- 6) The heat source, packaged for shipment, will survive a one-half hour 1850°F black body fire. Survival shall be interpreted to mean no loss in fuel containment. The package shall not be cooled artificially until the 30 minute exposure period is terminated and the temperature at the center of the package has stabilized or begun to fall.
- 7) Procedures used in transporting the heat source from ORNL to WANL will, insofar as possible, preclude earth burial.

3. Complete Generator

1) The SNAP-23A generator design will meet the transportation and/or licensing requirements of the AEC, ICC, FAA, Coast Guard, and IAEA and will be transportable by all usual modes.

2) The maximum radiation dose rate at the surface of the generator will be 200 mrem/hr. The maximum radiation dose rate at one meter from the surface of the generator will be 10 mrem/hr.

3) Thermal design of the generator will be capable of accommodating an open circuit (no electrical load) situation without damaging structural integrity of the heat source.

4) The design of the generator and/or its assembly procedures will allow a reasonable time in which to assemble the converter onto the generator without compromising either the nuclear safety of the generator or generator performance.

5) The complete generator, enclosed in a shipping container, will withstand impact on a hard rigid surface at terminal velocity of the generator package or 150 ft/sec, whichever is more. The impact will not cause breach of the fuel capsule or compromise shielding integrity beyond that which gives a dose rate of 1000 mrem/hr at three feet from the external surface of the package.

6) It shall be demonstrable analytically that the complete generator will meet the hypothetical accident conditions posed in Appendix B of Reference 34. These conditions include:

- (a) 30 foot drop onto flat rigid surface
- (b) 40 inch free drop onto mandrel
- (c) exposure to 30 minute 1475°F fire (this is less severe than criterion 7).

7) The complete generator, packaged for shipment, will survive a one-half hour 1850°F black body fire. Survival shall be interpreted to mean no loss in fuel containment. The package shall not be cooled artificially until the 30 minute exposure period is terminated and the temperature at the center of the package has stabilized.

8) Procedures used in transporting the heat source to an operational or test site will, insofar as possible, preclude earth burial.

9) Shock overpressures of up to 28,000 psi must not compromise generator heat rejection capability to a degree where heat source integrity is compromised. Shielding will contain fuel radiations to less than 1000 mrem/hr at three feet from the external surface of the package.

10) A site evaluation shall be performed in order to assure that the fuel and its radiations will be contained under credible site accident environments.

Safety Analysis of the 60 W(e) SNAP-23A System

1. Thermal Environment

Analyses of the effects of adverse thermal environments are being made with the aid of a 72 node transient thermal model of the generator. The basic makeup of the model is discussed in the Systems Analysis section (Subtask 315) of this report. To date, the transient problems investigated considered the generator and bare heat source exposed to an 1850°F one-half hour fire. Limited results are available for the generator in a 1475°F fire defined as one of the AEC hypothetical accident situations³⁴.

Some of the results obtained for the complete generator are shown in Figure 6-1. The temperature profile presented starts at the middle of the fuel capsule (node 13) and extends radially outward through all layers until the outer shell is reached (node 48). The insulation effectively limits the heat source thermal input to that which comes through the converter subassembly. The variation of some of the key generator temperatures as a function of time is shown in Figure 6-2. Note that the aluminum radiator melts in about 0.2 hour, after which the major part of the fire heat flux enters the generator through node 67, the cold shoe sleeve. In all generator analyses, the initial temperatures are for an open circuit generator in order to introduce some conservatism to the results.

The generator thermal model has also been used to estimate the effects of a fire on the heat source. This has been accomplished by considering all nodes external to the heat

Figure 6-1. SNAP-23A Complete Generator in 0.5 Hour 1850°F Fire Radial Temperature Profile

Figure 6-2. SNAP-23A Complete Generator in 0.5 Hour 1850°F Fire

source as isothermal and at a temperature of 1850°F. Heat input from the fire is by radiation only. Temperature results for an unprotected heat source are shown in Figures 6-3 and 6-4. The temperature profile in Figure 6-3 is a radial cut through the generator center. The significant result from the analysis conducted thus far is that the bare heat source is not compromised by the 1850°F half-hour fire. The heat source temperature transients are given in Figure 6-3. The shield and fuel temperatures will continue to climb after 0.5 hour; however, the magnitude of the thermal lag will not approach the melt temperatures of any material component or eutectic.

To date, results for a fire environment are generally favorable. It appears that the thermal safety criteria for either the heat source or generator can be achieved.

2. Ground Burial

An analytical model for ground burial of the generator is currently under preparation. The model analytically surrounds the generator with a low conductivity soil by adding 14 soil nodes to the basic 72 generator nodes. The first computer run will consider the generator in an upright position with 12 inches of soil over the top of the generator.

3. Water Release

A water release and diffusion analysis for the SNAP-23A fuel inventory has been completed. The analysis has considered the case of instantaneous release of 187,500 curies of Sr-90 and also the case of a slow continuous release of the titanate fuel form. The maximum permissible continuous concentration (MPCC) of Sr-90 in seawater has been equated with allowable body burdens and is estimated to be $3.5 \times 10^{-5} \mu\text{C}/\text{cc}$. A slow continuous release of the total beginning-of-life fuel inventory will result in seawater concentrations which never exceed the MPCC at radial distances beyond 1000 meters from a completely exposed source. Even these concentrations will effectively exist only at the depth of burial. The results for a slow release of the fuel inventory are shown in Figure 6-5. The analytical expression³⁶ for the time and distance dependent concentration are indicated on the figure.

The upper bound for continuous solutions is an instantaneous release of the total inventory. The instantaneous solution is also more nearly representative of seawater release of a strontium oxide fuel form. The concentrations for instantaneous release of 187,500 curies

Figure 6-3. Unprotected Heat Source in 1850°F Fire

Figure 6-4. Exposed Heat Source in 1850°F Fire

Figure 6-5. Seawater Concentration of Sr-90 Versus Time After Release

are given in Figure 6-6. The MPCC value is exceeded for a limited time and distance from the source. Concentrations are below the MPC_{sw} value for distances above 500 meters. The MPC_{sw} (maximum permissible seawater concentration) is another commonly used index and is based upon the relationship between radioactivity in fish consumed by man and the radioactivity ingested by drinking water with maximum permissible concentration of Sr-90.

Figure 6-6. Seawater Concentration of Sr-90 Versus Time After Release

Subtask 640 - Safety Review

The objective of this subtask is to provide overall program management concerning those aspects of the SNAP-23A effort which are pertinent to radiological safety. The following accomplishments were realized under this subtask during this period.

Orientation

The nature and scope of the safety problems previously considered by the AEC and industry for isotopic power generators were identified from a literature survey of published safety reports. Particular emphasis was placed on SNAP-7 documentation although other programs were also included. It was established that the safety design of the terrestrial systems has been based on absolute fuel containment and that the specifications contained in the Title 10-Code of Federal Regulations were used to define the accident environments during transport. It was also evident that engineered safeguards depended strongly on the system application and that the safety design of SNAP-23A for a broad terrestrial application would generally tend to be somewhat more restrictive than an actual mission would warrant.

The current mechanical, thermal, and nuclear designs of the SNAP-23A system were reviewed in detail to gain an appreciation of the influence of safety constraints on system performance. A similar review of other RTG systems designed for specific missions revealed that significant design changes in the heat removal area would undoubtedly be required for certain applications of SNAP-23A. For example, heat removal from the SNAP-7D system employed as a power source for a floating weather station is achieved by conduction to an oil bath using full-length axial cooling fins. The response of the generator to some accident conditions could be materially affected by such changes, which must therefore be given careful consideration in the course of the program.

Applicable sections of the Code of Federal Regulations (10CFR20 and 10CFR71), the IAEA shipping regulations for radioactive materials, miscellaneous publications on transport requirements and accident environments, and numerous articles and texts on the ocean and its characteristics were also reviewed. It was determined that the CFR and IAEA

regulations were generally consistent with each other in the area of radioactive shipments and hence few problems should be encountered in the transport of a SNAP-23A system beyond the continental United States.

Design Review

The design effort was closely monitored during this period and three areas of safety significance were identified:

1. Radiation Streaming - The end-plug design for the biological shield was found to have undesirable contours which would substantially reduce its shielding effectiveness. The problem was brought to the attention of Hittman Associates who recommended changes that would preserve the required attenuation of bremsstrahlung radiation and enhance the ability of the capsule and shield to survive highly velocity impact.
2. Shield Quality - The procurement specifications for the radiation shield were found to be inadequate in terms of defining the acceptability of interval voids and the extent of repairs that can be made to a defective unit by the supplier. Hittman was requested to supply this information as input to the shield specifications.
3. Fuel Contaminants - The WANL thermal design effort and the Hittman shield design task is currently proceeding on the basis of a pure strontium-90 fuel form. An evaluation of the significance of fuel contaminants was initiated to determine if the heat load from strontium - 89 and/or the gamma activity from residual fission products could present safety problems related to capsule wall temperatures and shield effectiveness. Communication with Dr. S. Rimshaw of the ORNL Fission Products Development Laboratory yielded data on the proportion of strontium-89 in strontium-90 fuel at discharge for a wide range of irradiation periods which are summarized in Figure 6-7. This information was used to estimate the heat output from a 185,000 curie source of strontium-90 produced during a 30-day and a 200-day irradiation period as a function of time after discharge (Figure 6-8). It is evident that fuel which has been aged longer than 1.5 years does not present any problem. However, depending on the source of supply, the use of one year old fuel will involve 200-400 watts of extraneous

Figure 6-7. Atomic Ratio ($\text{Sr } 89 / \text{Sr } 90$) at Discharge for Various Irradiation Periods

Figure 6-8. Effect of Sr 89 on Heat Output of SNAP-23A Fuel

heat from strontium-89 at beginning of life. ORNL maintains an inventory of approximately 10^6 curies of "aged" strontium-90 which would probably be an available source of supply for SNAP-23A prototypes. Unless procurement of fuel is arranged well in advance of the production phase, however, the heat output of strontium-89 will require special consideration in the system thermal design.

With regard to gamma activity from fission product contaminants, Rimshaw claims that chemical reprocessing has been sufficiently refined that radiation shielding of SNAP-23A need only be concerned with the strontium-90 bremsstrahlung. This aspect will be explored further in the next reporting period.

Safety Review

In conjunction with the cursory safety analysis performed by Hittman Associates to develop Safety Design Criteria, the three following areas having significant influence on the generator design were given close attention:

1. Thermal Environment - The choice of a half-hour fire at 1475°F specified by 10CFR71 as an accident condition which must not compromise fuel containment or shield effectiveness was reviewed in light of a published report³⁷ indicating that large, free-burning, hydrocarbon-fueled fires can generate an overall temperature as high as 1850°F. It was established that the one-hour standard fire originally specified by 10CFR71 was a realistic representation of an unattended fire within a building of normal combustible loading; this fire specification involves a time-varying temperature reaching a peak value of 1700°F. Discussions with members of the AEC Regulatory staff revealed that tests with hydrocarbon-fueled fires and closer scrutiny into the mechanics of transport accidents prompted a change in this fire specification which now conforms to current IAEA regulations. The basis for selecting a half-hour fire at a constant temperature of 1475°F (800°C) is comprehensively discussed in a UKAEA report³⁸. It is, therefore, believed that safety design of the SNAP-23A generator using the thermal environment specified by the current 10CFR71 regulations will

neither present any undue risk to the general public nor require any significant modifications to the basic system design.

2. Shield Integrity - Consideration of the high radiation levels that exist in the vicinity of an unshielded capsule (~ 300 rem/hr at one meter) made it quite evident that loss of shielding integrity during a transport accident could expose the public to serious radiation doses. Hittman was therefore directed in the early stages of its definition of safety criteria to provide for shield integrity as well as fuel containment for all credible accidents, taking any advantage of protection afforded by the shipping container during transport phases. The ability of the shield to survive a fire was considered to present the greatest obstacle to meeting this basic safety philosophy and hence contact was made with Union Carbide at Paducah which is engaged in a test program to evaluate the performance of uranium shipping casks during transport accidents. Mr. S. Bernstein reported unpublished results in which a stainless steel clad annular cylinder of unalloyed uranium (15 inch I.D. x 18 inch high x 1 1/2 inches thick) exposed to an oil fire at temperatures exceeding 1800°F for a period in excess of one hour experienced no visible damage. It was judged that the effects would have been more severe if rupture of the cladding had permitted oxidation of the shield material. Application of this experience to the SNAP-23A heat source assembly, which is also a stainless steel clad uranium body, will depend on a more detailed safety analysis that will establish the integrity of the steel shell during an impact/fire accident and the extent of eutectic melting at the shield-accumulator interface.

3. Deep Sea Burial - The safety aspects of a deep-sea burial following a transport accident or sinking of a floating station was given considerable attention due to the strong influence of a deep-ocean environment on the fuel capsule design. An investigation into the topography of the deep ocean³⁹ revealed that beyond 100-200 nautical miles of the coast-line, a large fraction of the Atlantic floor lies at the maximum depth of 3000 fathoms (18,000 ft). The average depth of the Arctic and Pacific differs only slightly from the Atlantic. The presence of localized deep trenches in the Caribbean and Pacific ranging from 4000 to 6000 fathoms need

not be considered since these locations can be avoided as operational sites or shipping lanes without any serious constraint on system usefulness. Design of the capsule to withstand hydrostatic pressure equivalent to a depth of 20,000 feet is therefore considered to be a realistic design value.

The choice of a 0.1 mil/year ocean water corrosion rate for Hastelloy-C reported in the open literature does not include galvanic effects which depend on the detailed system design and which may significantly alter the currently assumed corrosion rate for the capsule material. This aspect of capsule performance will be given particular attention in a meeting with NRDL scheduled for the immediate future.

The intent of the absolute containment philosophy adopted for the SNAP-23A program is to prevent internal radiation exposure of the public by inhalation or ingestion of the strontium-90 fuel. This approach is obviously necessary when a breach of the capsule would release the radioisotope to the atmosphere or coastal/surface waters where marine life is harvested for human consumption. However, there is strong evidence that little or no communication exists between the deep ocean and surface waters so that loss of fuel containment following a deep-sea burial has little likelihood of contaminating the human biosphere. In a contribution by Broecker "Radioisotopes and Large-Scale Oceanic Mixing" for the book "The Sea" by Interscience Publishers, it was concluded that dissolved solids remain in the deep Pacific ocean more than 1000 years before returning to the surface while the corresponding time for the Atlantic is about two-thirds as great. The relatively rapid dilution that will occur following a capsule rupture and the low-efficiency exchange mechanisms that are required to transfer the radioisotope from the deep sea to the biosphere via the seafood chain are additional arguments which suggest that fuel release following a deep sea burial does not constitute a radiological hazard. It is therefore recommended that the safety implications of injecting high-level radioactivity into the marine environment be given more serious consideration in this program.

Documentation

As required by the SNAP-23A Program Plan, a Safety Design Criteria document is being prepared by Hittman Associates. Three WANL/Hittman interface meetings were held to resolve problem areas so that issue of the document is now planned for the vicinity of June 1, 1967. This document represents a significant improvement over prior efforts in this area yet is inherently unable to serve as a completely definitive guide to the system design. The liaison between safety analysis and system design is therefore expected to exert a role of greater importance in the remaining period of the Phase I effort.

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APPENDIX A

Westinghouse Electric Corporation

Astronuclear Laboratory
P. O. Box 10864
Pittsburgh, Pa. 15236
(Fed. Ident. Code No. 14683)

* PURCHASING DEPARTMENT SPECIFICATION 30107

April 7, 1967

CASTING, URANIUM ALLOY

1. SCOPE

This specification covers the requirements for uranium alloy intended for use as a radiation shielding material, designated as follows:

<u>PD Spec Designation</u>	<u>Description</u>
30107-1	An alloy of depleted uranium containing 8% molybdenum in the as-cast condition

2. APPLICABLE DOCUMENTS

2.1 The following documents, of the issue in effect on the date of invitation for bids, shall form a part of this specification to the extent specified herein.

Federal Test Method Standard 151

2.2 Copies of Federal Standards required by contractors in connection with specific procurement functions should be obtained as indicated in the Department of Defense Index of Specifications and Standards.

3. REQUIREMENTS

3.1 Chemical Composition: The material shall conform to the following composition:

<u>Element</u>	<u>Range or Limit, Per Cent</u>
Molybdenum	7.8 to 8.2
Carbon	.035 max.
Oxygen	.025 max.
Cobalt	.010 max.
Iron	.010 max.
Nickel	.010 max.
Uranium -235	.22% max.
Uranium (Total)	Remainder

3.2 Repair of Defects: If approved by the purchaser, defects revealed by machining, such as cold shuts and shrinkage cracks, may be removed by machining a straight sided cavity into which a plug of material conforming to Section 3.1 is inserted. The manufacturer shall submit a description of a proposed repair for purchaser's approval.

3.3 Soundness: The casting shall be sound as indicated by the gamma scanning technique. The technique and acceptance standards shall be negotiated between the manufacturer and the purchaser.

3.4 Surface Condition: After machining and repair of defects, if applicable, the surface of the casting shall be smooth, free of defects such as cracks and dents, and free of contamination such as grease, oxides and miscellaneous dirt.

3.5 Density: The density of the casting shall be 0.629 lb/in.³ (17.4g/cc) minimum.

4. QUALITY ASSURANCE

4.1 Source Surveillance

4.1.1 Material covered by this specification is subject to source surveillance by a Westinghouse representative. This may include surveillance of the product and of the manufacturer's systems, procedures, and facilities which relate to testing and inspection of this material. The manufacturer shall perform tests and supply data as required to illustrate compliance with all requirements of this specification.

PDS-30107

4.1.2 Material accepted by the purchaser's representative at the place of manufacture which is later found to be not in conformance with this specification, is subject to rejection.

4.2 Consistent Quality: No change shall be made in the quality of successive shipments of material furnished under this specification without first obtaining the approval of the purchaser.

4.3 Visual Inspection: The casting as supplied shall be visually inspected without optical aids for conformance to Section 3.4.

4.4 Chemical Analysis

4.4.1 Sampling: From each casting, a sample shall be obtained from each of the top and bottom locations and from another location if specified on the purchase order. Samples shall consist of clean cuttings obtained by any mechanical means.

4.4.2 Test Methods: The samples shall be analyzed in accordance with the applicable sections of Methods 111 and 112 of Federal Test Method Standard No. 151. Total uranium need not be determined.

4.5 Density Determination: Density shall be calculated as the ratio of weight in air to the volume computed from dimensions.

4.6 Records and Reports

4.6.1 Records: The manufacturer shall maintain complete records of the processing steps and retain them for two years. They shall be made available to the purchaser upon request.

4.6.2 Reports: The manufacturer shall furnish five copies of a certified test report showing the information and results of the tests specified in Sections 4.4 and 4.5 and bearing a statement that the material conforms to the requirements of this specification. The report shall show the Purchase Order Number; Specification Number, including dash number; Melt Number; Test Identification Numbers; Date of Test; Name of Manufacturer.

5. PREPARATION FOR DELIVERY

5.1 Packing: The castings shall be packed in sealed containers which have been evacuated or filled with an inert gas and cushioned to protect the material from physical damage during shipment.

5.2 Marking: Each shipping container shall be plainly marked with the Purchase Order Number; PD Specification Number (including dashes); Gross, Tare, and Net Weights; Name of Manufacturer.

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* PURCHASING DEPARTMENT SPECIFICATION 30186

March 2, 1967

THERMAL INSULATION, FIBROUS

1. SCOPE

This specification covers the requirements for a fibrous refractory material for use as thermal insulation with a maximum service temperature of 1800 F, designated as follows:

<u>PD Spec Designation</u>	<u>Description</u>
30186-1	Min -K 2000 thermal insulation

2. SUPPLIER: Johns-Manville
 22 East Fortieth Street
 New York, New York 10016

3. CONSISTENT QUALITY

The manufacturer shall submit a certified statement that no significant change has been made in the manufacture of the product that would alter the integrity of the product or result in a change in the typical characteristics of the product since March 1, 1967.

4. LOT IDENTIFICATION: The supplier shall mark each container with an identification of the lot or batch of material and the date of manufacture.

5. CHARACTERISTICS

5.1 Forms: Supplied in sheets, slabs and molded shapes.

PDS-30186

5.2 Typical Properties:

- (a) Density, lb./cu. ft.: 20
- (b) Transverse strength, psi: 70
- (c) Compressive strength, psi:
 - 80 (5% compression)
 - 170 (10% compression)
- (d) Linear shrinkage, per cent: 1
- (e) Thermal conductivity, Btu in/sq. ft. hr. F:
 - 800 F - 0.27
 - 1000 F - 0.30
 - 1200 F - 0.34
 - 1400 F - 0.39
 - 1600 F - 0.46
- (f) Specific heat, Btu/lb. F:
 - 400 F - 0.23
 - 800 F - 0.25
 - 1200 F - 0.27
 - 1600 F - 0.27

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* PROCESS SPECIFICATION 597918

April 7, 1967

THERMAL INSULATION, FIBROUS - THERMAL TREATMENT

1. SCOPE

This specification covers the requirements for the thermal treatment of thermal insulation, designated as follows:

<u>Designation</u>	<u>Description</u>
597918-1	Thermal treatment of Min-K2000 (Johns-Manville trade name)

2. APPLICABLE DOCUMENTS

2.1 The following documents, of the issue in effect, shall form a part of this specification to the extent specified herein.

PDS-30186

3. REQUIREMENTS

3.1 Material: The material shall conform to the requirements of PDS-30186.

3.2 Thermal Treating Equipment

3.2.1 The furnace shall be capable of being heated to 1350 F at the rate of 200 F per hour minimum.

3.2.2 The heat input control shall be capable of maintaining any selected temperature to ± 25 F.

3.2.3 The furnace shall be equipped with apparatus for automatically recording the temperature at intervals of 15 minutes maximum.

3.3 Thermal Treatment Procedure

3.3.1 Heat Treatment for Elimination of Volatiles

FS-597918

3.3.1.1 The Min-K2000, set on pins or rails to allow air circulation underneath the material, shall be placed in the center of the furnace which shall be 300 F maximum.

3.3.1.2 Heat the material as follows:

- (a) Hold at 300 ± 25 F for one hour
- (b) Heat from 300 F to 500 ± 25 F in one hour maximum
- (c) Heat from 500 F to 800 ± 25 F at 100 F per hour maximum
- (d) Heat from 800 F to 1200 ± 25 F at 200 F per hour maximum
- (e) Heat from 1200 F to 1350 ± 25 F in one hour maximum

3.3.1.3 Hold at 1350 ± 25 F for eight hours minimum.

3.3.1.4 Furnace cool to 800 F maximum, remove from the furnace and air cool to 130 F minimum.

3.3.1.5 For storage up to machining, the material at 130 F minimum shall be placed in a plastic bag, which shall be filled with dry nitrogen or argon (dew point -60 F maximum) immediately and sealed.

3.3.2 Drying Treatment

3.3.2.1 The machined insulation, set on pins or rails to allow air circulation underneath the material, shall be placed in a furnace at 300 F maximum.

3.3.2.2 The machined material shall be heat treated as follows:

- (a) Hold at 300 ± 25 F for one hour
- (b) Heat to 750 ± 25 F in two hours minimum
- (c) Hold at 750 ± 25 F for three hours minimum
- (d) Remove from furnace and air cool to 130 F minimum
- (e) For storage awaiting assembly, the material at 130 F minimum shall be placed immediately in a plastic bag which shall be filled with dry nitrogen or argon (dew point -60 F maximum) and sealed.

PS-597918

4. QUALITY ASSURANCE

4.1 Surveillance: Adherence to the provisions of this specification shall be under the surveillance of a Westinghouse representative.

4.2 Compliance: No change shall be made from this specification, or an approved procedure, without first obtaining the approval of Westinghouse.

4.3 Records and Reports:

4.3.1 Records: Temperature, times, weight changes, and other heat treating variables shall be recorded. Temperatures shall be read and recorded at 15 minute maximum intervals during heating and holding periods. Records including furnace temperature charts shall be kept available for two years minimum.

4.3.2 Reports: The manufacturer shall submit five copies of a certified report confirming that the requirements of this specification have been met. The report shall show the purchase order number, part, serial and lot numbers, and heat treat batch numbers if applicable.