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Westinghouse Astronuclear Laboratory

PHASE 1

PRELIMINARY SAFETY ANALYSIS REPORT

AEC Contract AT(30-1)-3800

Prepared for the

U.S.ATOMIC ENERGY COMMISSION

FEBRUARY 1970

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PRELIMINARY SAFETY ANALYSIS REPORT CORRECTIONS

<u>PAGE</u>	<u>CORRECTION</u>
xi	Add Table VI-1, "Handling, Test and Storage Absolute Accident Probabilities" - page VI-5
I-3	New table supplied identifying headings
III-1	3rd line: add right-hand parenthesis
III-5	(1) Reference III-2 should be: Federal Register Vol. 33, No. 230, November 26, 1968 Revision to Title 10, Atomic Energy, Part 71 "Packing of Radioactive Materials for Transport" (2) Reference III-3 should be: Federal Register Vol. 33, No. 194, October 4, 1968 Title 49 "Transportation, Radioactive Materials and Other Miscellaneous Amendments".
IV-12	2nd paragraph, 12th line: change reference to "titanium" to "titanate".
IV-13	2nd paragraph: change 0.045 to 0.405.
IV-15	1st line: change 300 years to 60,000 years and 10 half-lives to 2000 half-lives.
VI-4	Figure VI-3, left-hand column, last item: change to "Potential Environments Resulting from Accident".
VI-19	Table VI-4, 3rd column from left: change .11 to 11.0 and .22 to 22.0

<u>PAGE</u>	<u>CORRECTION</u>
VI-32	Figure VI-13: change ordinate to read; "Time to 1000 mr/hr at 3 ft, hrs".
VI-34	Last line: Add this sentence after sentence ending with 1.12×10^{-2} : "The probability that the SNAP-23A system is involved in a handling, storage, or test accident is approximately 3.14×10^{-2} ."
VI-35	Figure VI-14: change 4.16×10^{-2} to 4.26×10^{-2} .
VI-37	New Figure VI-15 supplied.
VI-38	Figure VI-16: Add to third box on left side column of boxes: <ul style="list-style-type: none"> ● Shallow water burial
VI-39	New page supplied.
VI-44	(1) 4th line: change $\frac{1}{\rho_e C_c}$ to $\frac{1}{\rho_c C_c}$ (2) Replace $\rho_i C_i$ with $\rho_1 C_1$ in: $\sigma = \rho_i C_i V_f \left[1 - \exp \left(\frac{-2 \rho_c C_c}{\rho_1 C_1} \right) \right]$
VI-45	Table VI-5: The first entry in 2nd column, "Accident Cases" should be changed to 60 mph.

PAGE

CORRECTION

VI-48

Line 14: change to read $\rho_c C_c \leq 40 \frac{\text{lb-sec}}{\text{in}^3}$ instead of $\rho_c C_c \leq 40 \frac{\text{lb-sec}}{\text{in}^3}$

VI-65

The maximum penetration depth, $D_{\max}$, should be changed to read 0.35 ft to correspond with an "N" value of 0.56. For N value approaching 1.4, $D_{\max}$ would increase to 0.87 ft. No conclusions on soil burial are altered by this change.

VII-2

Table VII-1: Enter the value of $3.5 \times 10^{-5} \frac{\mu\text{C}}{\text{cc}}$ in the space corresponding to maximum permissible concentration in shallow ocean.

VIII-11

Table VIII-4: Under heading of "Capsule Inventory" interchange contents of Capsule A and Capsule B.

VIII-18

Figure VIII-15: Change abscissa dimensions to amp/cm^2 .

VIII-43

References to $\gamma' + a$ should be $\gamma' + \alpha$.

VIII-44

References to $\gamma' + a$ should be $\gamma' + \alpha$.

IX-8

Figure IX-1:

- (1) Top left hand box corresponding to Step 1, item c: change titanium to tantalum
- (2) Step 3, item a: delete phrase "with platinum encased"
- (3) Step 3, item d: Change to read: "Assemble diffusion barrier around capsule"
- (4) Step 6, should read: "Assemble SNAP-23A System at WANL"
- (5) Step 8, delete phrase: "Mount SNAP-23A on special pallet"

PAGE

CORRECTION

IX-10

New Table IX-1 supplied.

D-11

Change Equation D-9 to: $D_{LL}(l, t) = \frac{E_L}{M_L} \frac{f_3 l}{\lambda_L} \left(e^{-\lambda t} \right)$

E-2

Change Equation E-2 to:

$$S_c(r, t) = \frac{q/D}{2\sqrt{\pi}\omega r} \times \frac{2}{\sqrt{\pi}} \int_{y=\frac{r}{\omega t}}^{y=\infty} e^{-y^2} dy$$

E-3

Change Equation E-3 to:

$$S_c(r, t) = \frac{q/D}{2\sqrt{\pi}\omega r} \times \frac{2}{\sqrt{\pi}} \int_{y=\frac{r}{\omega t}}^{y=\left[\frac{r}{\omega(t-t_0)}\right]} e^{-y^2} dy$$

F-9

New Figure F-4 Supplied.

F-13

New Figure F-6 Supplied.

F-14

4th line: change to $0.2625 \times 10^{+6}$ psi

G-32

Change both references "δ facing" to "σ facing".

G-50

Change equation $\frac{\sigma_r}{\sigma_m} = \frac{\epsilon_p}{\epsilon_m}$ to: $\frac{\sigma_p}{\sigma_m} = \frac{\epsilon_p}{\epsilon_m}$

PAGE
CORRECTION

G-52 Change equation G-2 to $Z = \frac{\text{Force}}{\text{Velocity}}$

G-55 Change the equation

$$U_1 = U_o - U_2 = U_o \left[1 - e^{\left(\frac{-2 \rho_2 C_2}{C_o} \right)} \right] \text{ to: } U_1 = U_o \left[1 - e^{\left(\frac{-2 \rho_2 C_2}{\rho_o C_o} \right)} \right]$$

(That is, add " ρ_o " to denominator of exponent)

G-58 Figure G-17: Change abscissa to " $M, (IN^2)$ "

G-61 Table G-2

- (1) Column headed "Receiver Composite Impedance" : change first entry to 68 (was 88)
- (2) Column headed Capsule Stress (psi): change fifth entry to 619000 (was 61900)

G-64 Last line: change equation

$$\rho_6 C_6 = \sqrt{\frac{K I \rho}{A}} \quad \text{to} \quad \rho_6 C_6 = \sqrt{\frac{K I \rho}{A}}$$

" ρ " is not a subscript to "I" in this equation.

PAGE

CORRECTION

G-65

Change equation $\rho_6 C_6 = \frac{4600 \text{ lb/in.} \times 60 \text{ in.} \times 1620 \frac{\text{lb}}{\text{in}^3}}{386 \times 60^2 \text{ in.}^2 \times 60^3}$ to:

$$\rho_6 C_6 = \sqrt{\frac{4600 \text{ lb/in.} \times 60 \text{ in.} \times 1620 \text{ lb/in.}^3}{386 \times 60^2 \text{ in.}^2 \times 60^3}}$$

H-1

Replacement page supplied.

H-3

Replacement page supplied.

TABLE I-1

CONFORMANCE WITH SAFETY CRITERIA

<u>Safety Criterion (1)</u>	<u>Criterion Satisfied by Analysis</u>	<u>Criterion Satisfied by Test</u>	<u>Further Evaluation Required in Phase II</u>	<u>Requires Procedural Controls</u>
<u>A. Fuel Capsule</u>				
1. Containment	X			
2. Hydrostatic Pressure	X	X		
<u>B. Heat Source</u>				
1. Ship by Truck				X
2. Conform to AEC/DOT Regulations for Transport	X (2)			
3. Evaluate Heat Source as Integral Unit	X			
4. Survive 150 fps Vehicle Impact	X (3)			
5. Survive Overpressure Environment	X			
6. Survive 1850°F Fire	X			
7. Preclude Earth Burial				X
<u>C. System</u>				
1. Conform to AEC/DOT/IAEA Transport Requirements		X (5)	X (4)	
2. Satisfy Dose Rate Standards	X	X		
3. Survive Open Circuit Condition	X			
4. Reasonable Converter Assembly Time	X			X

TABLE I-1 (CONTINUED)

<u>Safety Criterion (1)</u>	<u>Criterion Satisfied by Analysis</u>	<u>Criterion Satisfied by Test</u>	<u>Further Evaluation Required in Phase II</u>	<u>Requires Procedural Controls</u>
5. Satisfy Transport Impact Accidents			X (4)	
6. Survive 1850°F Fire	X	X (5)		
7. Preclude Earth Burial a) during transport b) on site	X (4)			X
8. Survive Overpressure Environment	X			
9. Safe Site Operation			X	

(1) Detailed listing of safety criteria can be found in Section III.

(2) License to transport FPMS type heat source assembly between WANL and ORNL received on November 7, 1969 (USA-DOT-RAM-6123).

(3) This criterion is equivalent to that specified in Criterion B. 2.

(4) Conceptual design of system shipping container indicates feasibility of satisfying criterion.

(5) Simulated converter fire tests provided data on response of unprotected converter to 1475°F and 1850°F fires (see Section VIII-F).

* Site Probability Depends Upon Site Selection and Cannot Be Estimated at This Time.

Figure VI-15. SNAP-23A System Handling, Storage, Test, and Site Accident Network

distribution. Assuming the binomial distribution applies to this discrete process, the probability of an accident occurring in 3000 miles is determined as follows:

$$P \left\{ \begin{matrix} n \\ r \end{matrix} \right\} = \frac{n!}{r! (n-r)!} (p)^r (q)^{n-r}$$

where $P \equiv$ Probability of r successes in n trials
 $n \equiv$ number of independent events (in this instance, the number of miles)
 $p \equiv$ probability of success for one trial
 $r \equiv$ number of successes (in this case, number of accidents)
 $q \equiv 1-p$

Thus with the following values for the variables:

$r = 0$
 $n = 3000$
 $p = 3.5 \times 10^{-6}$ accidents/mile

the probability of no accidents occurring in 3000 miles is, from the above equation:

$$P \left\{ \begin{matrix} 3000 \\ 0 \end{matrix} \right\} = \frac{3000!}{0! (3000)!} (3.5 \times 10^{-6})^0 (0.9999965)^{3000}$$

$$P \left\{ \begin{matrix} 3000 \\ 0 \end{matrix} \right\} = 0.9895$$

The probability for one accident in 3000 miles is:

$$P \left\{ \begin{matrix} 3000 \\ 1 \end{matrix} \right\} = \frac{3000!}{1! (3000-1)!} (3.5 \times 10^{-6})^1 (0.9999965)^{(3000-1)}$$

$$P \left\{ \begin{matrix} 3000 \\ 1 \end{matrix} \right\} = 0.0150$$

In Appendix H, the data are shown for subdividing the aircraft accident into three types: takeoff or landing, mid-air collision, and other freight accidents. These are shown on Figure VI-16 for a 3000 mile shipment. Division of the truck accidents into three specific categories is also shown on Figure VI-16. The probabilities use the same data described in Section VI-B Paragraph 1.

2. Handling, Storage, and Test Accidents

a) Impact. Two configurations are involved for this phase of SNAP-23A operation: The bare system and the system in its shipping container.

TABLE IX-1
RADIATION DOSE RATE DATA FOR VARIOUS
PHASES OF HEAT SOURCE AND SYSTEM HANDLING

Configuration	Measured or Calculated Dose Rate (mrem/hr)**	Scaled Dose Rate*** (mrem/hr)	Allowable Radiation Worker Exposure Time per Calendar Quarter		
			Whole Body 1.25R	Hands and Forearms 3.0R	18.75R
Bare Fuel Capsule					
50 cm	808×10^3	903×10^3	N/A	N/A	N/A
100 cm	208×10^3	232×10^3	N/A	N/A	N/A
Heat Source Assembly					
Surface	400	447	2.8	6.7	42
50 cm	40	45	28.0	67.0	420
100 cm	12	13.4	93.3	224.0	1400
SNAP-23A System*					
Surface	160	179	7.0	16.8	105
50 cm	25	28	44.6	107.0	670
100 cm	9	10	125	300.0	1875
Heat Source Shipping Container*					
Surface		53	23.6	56.6	345
50 cm		15	83.0	200.0	1250
100 cm		7	178.0	430.0	Continuous
System Shipping Container*					
Inner Container					
Surface		45	27.8	67.0	415
50 cm		13	96.0	231.0	1440
100 cm		6	208.0	500.0	Continuous
Air Transport Configuration					
Surface		13	96.0	231.0	1440
50 cm		6	208.0	500.0	Continuous
100 cm		4	313.0	Continuous	Continuous

* Extrapolated from test data

**Reference is 165,700 Ci

***Based on 185,000 Ci

Figure F-4. Effect of Yield on Peak Overpressure and Peak Reflected Overpressure

Figure F-5. Side Fin Pressure Profile

613157 3B

Figure F-6. Equivalent Elastic-Plastic Curve for 1100 Aluminum

APPENDIX H

AIRCRAFT ACCIDENT RISK

An analysis of aircraft accident data which yield accident occurrence probabilities is presented herein. Probability data are based on a series of documents published by the Civil Aeronautics Board for the period 1960 through 1962 and the National Transportation Safety Board from 1964 through 1967 entitled "Annual Review of U. S. Air Carrier Accidents." The accident per mile statistics used for SNAP-23A were selected from the category of aircraft called "nonscheduled revenue service - all cargo" defined in the references as follows:

"All cargo revenue flights that are not operated in regular scheduled service such as charter flights and all nonrevenue flights incident to such flights."

All aircraft accident data are summarized in Table H-1. The probability of an accident involving the generator and its shipping container in a chartered cargo flight was computed as 3.3×10^{-8} accidents/mile. This value is an average of the annual statistics tabulated for domestic carriers (2.7×10^{-8} accidents/mile) and international carriers (3.9×10^{-8} accidents/mile). Since transportation of the SNAP-23A system by air is assumed to be from WANL to any site in the world, the average (accident/mile) statistic for the domestic and international territorial classes has been presented.

Three specific accident types have been defined and comprise the total aircraft accident probability. The accident can occur:

- During takeoff or landing
- As any mid-air collision between aircraft
- As any other inflight or cruise accident.

Their total is one or 100 percent. The probability of the specific event occurring is also given in accidents/mile. Each was determined by multiplying the frequency factor by the total aircraft accident probability, 3.3×10^{-8} . When these event probabilities are multiplied by 3000 miles, the values appearing in Section VI-C result.

The frequency factors have been determined after reviewing the detailed accident resumes for each year of the nonscheduled revenue - all cargo carriers. For the seven years reviewed there were a total of 48 accidents by these special carriers. There were 40 accidents during the take-off, or landing phases, five accidents en route or during cruise, and three mid-air collisions. In 18 cases, or 40 percent of the accidents, fire resulted. Fire was present in:

- (1) Thirty-two percent of the take-off or landing accidents
- (2) Sixty percent of the accidents en route
- (3) Two of the three mid-air collisions

Deployment of the generator at or near the site may be by helicopter. Based on the annual statistics for the helicopter domestic carriers non-scheduled revenue service, the accident probability for this transportation mode has been computed to be 0.76×10^{-6} .

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SNAP-23A PROJECT MANAGER
SYSTEMS AND TECHNOLOGY

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I. SUMMARY

A preliminary nuclear safety assessment of the SNAP-23A system has been completed as part of the Phase I effort. The studies indicate that the system, based on the FPMS reference design, is generally in accord with the safety criteria evolved at the start of the SNAP-23A program. The significant results of this assessment are as follows:

- Six tests have been completed on the SNAP-23A program which have a direct bearing on assessing nuclear safety. These are: (1) radiation measurements on the heat source; (2) hydrostatic tests of the fuel capsule; (3) seawater corrosion test of the heat source materials; (4) an electrically heated mockup by the system (EMS); (5) materials compatibility tests at normal and elevated temperatures; (6) a test to determine post fire heat conductance capability of the converter. The results of the tests have been favorable, and generally correlate analytical estimates of system behavior and response to adverse accident environments.
- Various impact analyses have indicated that the heat source container can safely meet handling, storage, test, and transport impact environments. The heat source shipping container offers adequate protection. Preliminary studies show that it is feasible to design a shipping container for the SNAP-23A system that will meet safety requirements in the event of many types of transport accidents—including aircraft.
- Specific attention has been given to the environment, probabilities, and responses associated with air transport of the SNAP-23A system. The results obtained to date indicate that air transport is practical but the system will require a specially designed shipping container, and airplane tiedown provisions.
- The radiological consequences of a hypothetical airborne fuel release were evaluated with a source term based upon fuel vapor pressure and a postulated fire environment. Using a simple body model which defines the uptake of the Sr-90 and the dose to critical human organs, radiation exposures were found to be well below the allowable critical organ doses for the general public.
- Shallow and deep water continuous fuel releases were evaluated on the basis of maximum permissible continuous concentration (MPCC) as the index to assess the severity of a fuel release and it was concluded that the underwater fuel release would not present a severe safety hazard.

- Analytical studies were made to determine the response of the SNAP-23A system to 1475°F and 1850°F one half-hour fires. The studies showed that the post-fire converter conductance has a major effect in determining whether the steady state temperatures achieved following the fire will result in fuel capsule and shield integrity. Converter tests in a simulated fire environment have demonstrated that converter conductance is not adversely affected as a result of either fire. This test result, combined with analytical studies, indicates that the system alone can meet safety criteria. That is, post-fire steady state temperatures will be within acceptable limits without engineered safeguards or artificial cooling. The system shipping container will provide additional protection from the fire environment during transport.
- Siting problems and hazards have been investigated. The general scope of siting accidents can be related directly to the detailed accident analyses contained in this report. Other than earth burial, siting poses no specific safety problem. If burial in a very low conductivity soil is a potential hazard (and this is doubtful) then simple engineered safeguards can be used.
- With one exception, the safety criteria evolved at the start of the program still appear to be attainable design standards. A thorough survey of the explosion-overpressure accident environment has indicated that the original criterion was unrealistic. The overpressure accident model defined in this report is based on conditions at 30 feet from a tank truck with a two percent equivalent TNT yield explosion.

Table I-1 presents a checklist that indicates the extent to which the FPMS design satisfies the safety criteria as determined from analyses and supporting test results contained in this report. Those criteria which require further evaluation are so indicated as well as those that will necessitate procedural controls and/or engineered safeguards.

TABLE I-1
CONFORMANCE WITH SAFETY CRITERIA
A. Fuel Capsule

1. Containment	X		
2. Hydrostatic Pressure	X	X	

B. Heat Source

1. Ship by Truck			X
2. Conform to AEC/DOT Regulations for Transport	X(2)		
3. Evaluate Heat Source as Integral Unit	X		
4. Survive 150 fps Vehicle Impact	X(3)		
5. Survive Overpressure Environment	X		
6. Survive 1850°F Fire	X		
7. Preclude Earth Burial			X

C. System

1. Conform to AEC/DOT/IAEA Transport Requirements		X(5)	X(4)
2. Satisfy Dose Rate Standards	X	X	
3. Survive Open Circuit Condition	X		
4. Reasonable Converter Assembly Time	X		X
5. Satisfy Transport Impact Accidents			X(4)
6. Survive 1850°F Fire	X	X(5)	
7. Preclude Earth Burial
a) during transport
b) on site	X(4)		X
8. Survive Overpressure Environment	X		
9. Safe Site Operation			X

-
- (1) Detailed listing of safety criteria can be found in Section III.
- (2) License to transport FPMS type heat source assembly between WANL and ORNL received on November 7, 1969 (USA-DOT-RAM-6123).
- (3) This criterion is equivalent to that specified in Criterion B.2.
- (4) Conceptual design of system shipping container indicates feasibility of satisfying criterion.
- (5) Simulated converter fire tests provided data on response of unprotected converter to 1475°F and 1850°F fires (See Section VIII-F).

II. INTRODUCTION

The SNAP-23A radioisotope fueled thermoelectric generator system features a lightweight, compact design that incorporates the latest improvements in nuclear and materials technology. The system is being developed for varied terrestrial applications.

A definitive SNAP-23A application has not yet been specified, but it is anticipated that the system will be utilized for a variety of missions. It is clear that a SNAP-23A unit can be subjected to a wide spectrum of both normal and accident-induced environments during manufacture, transportation, and on-site operation. These environments and their effects upon the generator and/or heat source must be thoroughly assessed to determine whether the system is safe for its intended uses. Hence the system design is influenced strongly by these postulated environments, especially with regard to the fuel containment design.

The fuel of the reference 60 watt system consists of approximately 170,000 curies of strontium titanate (SrTiO_3) and is an intense source of beta and bremsstrahlung radiation. Since the allowable body burden of strontium-90 is very low, and the gamma dose rate adjacent to the bare fuel is approximately 40,000 rem/hr, assurance must be provided that deployment of the SNAP-23A generator will not endanger the health and safety of the operating personnel or the general public.

This document is a revised preliminary safety analysis report (PSAR) containing a comprehensive summary of the analyses, appropriate design data, and applicable test information that has been generated on a 60-watt SNAP-23A system based on the Fueled Prototype Mockup System (FPMS) design. The basic purpose of this report is to demonstrate assurance that the design meets the established safety objectives before proceeding with fabrication of additional fueled units, and to update the original PSAR issued in April 1968.

The attempt has been made to minimize the need for referencing topical reports, and where possible, detailed calculations have been placed in Appendices. The Appendices are not supplemental data; they substantiate and form a part of the accident analysis.

III. NUCLEAR SAFETY CRITERIA

Design, development, and test of the SNAP-23A system, heat source subassembly and shipping containers have proceeded on the basis of safety design criteria which have evolved from an operations analysis (Reference III-1 of the SNAP-23A system. A summary of these criteria are listed below which incorporates a change in the previous criterion on over-pressure as well as further amplification on the criterion concerning impact velocity.

A. Fuel Capsule

1. The complete fuel inventory of the SNAP-23A system shall 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 shall be accounted for in the capsule structural analysis. This criterion shall be verified by hydrostatic testing of the fuel capsule.

B. Heat Source

1. Shipment of the heat source assembly shall be via truck or railroad.
2. Shipment of the heat source assembly shall conform to regulations set forth by the AEC (Reference III-2) and the Department of Transportation (Reference III-3), (DOT). It shall be possible to demonstrate analytically that the hypothetical accident conditions posed in Appendix B of Reference III-2 can be met. These conditions include the following:
 - 30 foot free drop onto a flat rigid surface
 - 40 inch free drop onto a mandrel
 - Exposure to 30 minute 1475°F fire (this requirement is less severe than Criterion B.6.)

3. In all instances of impact, explosion, and fire accidents, fuel capsule and shielding shall be considered an integral unit. For cases other than water burial, potential damage to the fuel capsule shall reflect the protection provided by the outer layers of the heat source.
4. The heat source, enclosed in a shipping container, shall withstand vehicle impacts at net velocities up to 150 ft/sec. The complete fuel inventory shall 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.

Note: This criterion defines the potential impact environment for a transportation accident in terms of the transport vehicle rather than the shipping package. Investigations (References III-4, III-5, III-6) of impact-type accidents for vehicle, rail, and aircraft modes of transport concluded that it was most unlikely that a package being transported by any of these carriers would be subject to an impact exceeding that equivalent to a 30 ft drop upon an unyielding surface, as specified in federal regulations.

5. The heat source assembly shall withstand the overpressure environment created by the explosion of 876 lb of TNT located 30 feet from the heat source. There shall be no loss of radioisotope fuel containment or rise in dose rate levels beyond 1000 mrem/hr at three feet from external surface of the assembly.

Note: The rationale behind this new overpressure criterion is developed in Appendix E.

6. The heat source, in a shipping container, shall 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 fire has terminated and the temperature at the center of the package has stabilized or has begun to fall.
7. Procedures used in transporting the heat source from Oak Ridge National Laboratory (ORNL) to Westinghouse Astronuclear Laboratory (WANL) shall, insofar as possible, prevent earth burial.

C. SNAP-23A System

1. The SNAP-23A system design shall meet the transportation and/or licensing requirements of the AEC, DOT, and International Atomic Energy Agency (IAEA) and shall be transportable by all usual modes, including airplane, helicopter, ship, train, and truck.
2. The maximum radiation dose rate at the surface of the SNAP-23A system shall be 200 mrem/hr. The maximum radiation dose rate at one meter from the surface of the system shall be 10 mrem/hr.
3. Thermal design of the SNAP-23A system shall be capable of accommodating an open circuit (no electrical load) situation without compromising the structural integrity of the heat source.
4. The design of the SNAP-23A system and/or its assembly procedures shall allow a reasonable time in which to assemble the converter onto the heat source stud without compromising the safety of the SNAP-23A system.
5. The SNAP-23A system with its shipping container, shall be designed to withstand impacts resulting from all modes of transportation. The impact shall 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. Ground transportation impact shall include net vehicle velocities of up to 150 ft/sec. Airplane accident evaluations shall consider the plausible configurations evolved as the result of an accident. Regardless of the mode of transportation, the energy absorption capabilities of the carrier and shipping container will be considered. (See Note B.4)
6. It shall be analytically demonstrable that the SNAP-23A system will meet the hypothetical accident conditions posed in Appendix B of Reference III-2. These conditions include:
 - 30 foot drop onto flat rigid surface
 - 40 inch free drop onto mandrel
 - Exposure to 30 minute 1475°F fire (this is less severe than the following criterion:

7. The SNAP-23A system inside its shipping container shall 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 fire has terminated and the temperature at the center of the package has stabilized.
8. Procedures and the shipping container used in transporting the system to an operational or test site shall, insofar as possible, prevent earth burial.
9. The SNAP-23A system shall withstand the overpressure environment created for the explosion of 876 lb of TNT located 30 feet from the system. Heat rejection capability shall not be impaired to a degree where heat source integrity is compromised shielding shall be sufficient to maintain radiation levels below 1,000 mrem/hr at three feet from the external surface of the package. (See Note B. 5).
10. A site evaluation shall be performed to assure that the fuel and its radiations will be contained under credible site accident environments.

REFERENCES

- III-1. SNAP-23A Operations Analysis and Nuclear Safety Design Criteria, HIT-248 Revision B, March 1968.
- III-2. "Packaging of Radioactive Material for Transport" USAEC Rules and Regulations, Title 10, Atomic Energy Part 71 dated July 27, 1966.
- III-3. "Agent T. C. George's Tariff No. 19", ICC Regulations for Transportation of Explosives and other Dangerous Articles ... Effective date 5 Sept. 1966.
- III-4. Notes on Certain Aspects of the Regulations (Regulations for the Safe Transport of Radioactive Materials), Safety Series No. 7 Section 4.2.3, IAEA, 1961.
- III-5. The Safe Transport of Radioactive Materials, Edited by R. Gibson-UKAEA, Pergamon Press-1966, page 98.
- III-6. The Transport of Radioactive Materials, Interim Recommendations for the Application of Environmental Tests to the Approval of Packaging, UKAEA, AHSB(S) R19, 1963.

IV. SYSTEM DESCRIPTION

The SNAP-23A system described in this report is the current reference design which has evolved from a continuing series of design improvements accomplished during Phase I of the SNAP-23A development program. Changes have been made primarily as a result of extensive component and performance data from the electrically heated Experimental Mockup System (EMS). Design changes to this system may be accomplished during Phase II of the development program in which case conformance with safety criteria will be reevaluated.

Emphasis is placed on delineating those design features and procedures that have been incorporated into the system to assure that it will meet the established safety criteria. A more detailed description of the SNAP-23A systems from the performance standpoint will be found in Reference IV-1.

A. SYSTEM REQUIREMENTS

The criteria used in designing the SNAP-23A system, which were derived primarily from Reference IV-2, are classified as design requirements and system constraints. The former are basic to the system design and remain unaltered in conducting the detailed design and development program. The latter influence the design over and above basic design requirements, and cannot be defined in quantitative terms. In addition to these design/performance requirements, the SNAP-23A system must also be able to satisfy the Safety Design Criteria defined in Section III.

Design Requirements

● Unattended Operating Lifetime (minimum)	10 years
● Power Output (minimum-end-of-life)	60 watts (e)
● System Efficiency (minimum-end-of-life)	6.6 percent
● Voltage (with resistive load)	24 ($\pm$ 1) volts dc
● Maximum Dose Rate (surface of system)	200 mrem/hr
● Maximum Dose Rate (1 meter from surface)	10 mrem/hr

- Nominal Environment Free circulation of air at 70°F, 1 atm, and 50-100 percent humidity in a salt water corrosion atmosphere
- Shock Resistance Capability Per MIL-STD-810A, Method 516.1, Procedure 1, as defined in Reference IV-3. This includes peak shock inputs of 5 g's in each direction along the vertical axis and one horizontal axis.
- Vibration Resistance Capability Per MIL-STD-810A, Method 514.1, Equipment Class as defined in Reference IV-1. This includes excitations from 1.3 to 5 g's at frequencies from 5 to 50 cps along the same axes used for shock testing.

System Constraints

- Ease of field maintenance and replacement of the thermoelectric/electronic subsystem(s).
- Ease of handling, transportation, and installation.
- Adaptability to transportation under world-wide environmental extremes, including temperature extremes of -80°F to + 160°F.
- The reliability and quality of the system are to be assured by programs established and conducted generally in accordance with MIL-STD-785 and MIL-Q-9889A.
- The design BOL maximum converter hot junction temperature shall not exceed 1040°F with a 70°F ambient. This constraint is calculated to maximize EOL power output.
- The converter shall be designed to produce 61 watts at EOL conditions in order to assure the availability of 60 watts output from the system.

B. SUMMARY DESCRIPTION AND GENERAL DESIGN FEATURES

The SNAP-23A power system, shown in Figure IV-1, is designed to yield at least 60 watts of available power at an overall efficiency of at least 6.6 percent in an ambient air temperature of 70°F at the end of its 10-year life. The system consists of four major sub-assemblies, viz, the heat source, the lower base, the upper insulation and the power converter-radiator.

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Figure IV-1. SNAP-23A System

The assembly drawing in Figure IV-2 shows the fuel capsule in the center of the bottle-shaped heat source assembly. The fuel consists of four disk-shaped pellets of strontium titanate, SrTiO_3 . These pellets are contained within a Hastelloy-C capsule liner which in turn is fitted inside the fuel capsule. The capsule wall is relatively massive in order to meet the criterion for withstanding accidental submergence in 20,000 feet of sea water. The capsule is surrounded by a uranium-8 percent (by weight) molybdenum (U-8Mo) biological radiation shield, which is designed to reduce the radiation levels external to the system within the limits of 200 mrem at the surface and 10 mrem one meter from the system surface. The thin tantalum member is a diffusion barrier to prevent interactions between the shield and fuel capsule materials.

The shield is enclosed radially and above by the copper heat accumulator. This member conducts heat to the thermoelectric converter surrounding the neck of the bottle at the top of the assembly, and is also a diffusion barrier between the shield and the heat source jacket. A tantalum barrier plate located beneath the U-8 Mo shield plug reduces undesirable heat conduction out the bottom of the system.

An Inconel and stainless steel jacket surrounds the assembly, sealing it against oxidation and maintaining the heat source mechanical integrity under normal and accident conditions. The upper portion of the assembly is a tapered cylindrical sector or stud which provides a relatively uniform heating surface for the thermoelectric converter.

The heat source assembly is positioned in the base assembly, the latter being a hermetically sealed container that both insulates and supports the heat source assembly. The specified insulation is Min-K 2002 backfilled with krypton. A tubular truss array supports the heat source assembly during normal operation and also is capable of withstanding shock and vibration loads up to 5 g's. The hermetic seal across the top of the base assembly is a thin serpentine member designed to provide an adequate seal with minimum conduction heat loss. Assembly requirements prevent the use of a hermetic seal for the upper insulation, however, a rain shield prevents active fluid or gaseous interchange with the environment.

The thermoelectric converter, the radiator and the power conditioner comprise the converter-radiator assembly. The anodized aluminum radiator is manufactured in two halves

1. RADIATOR
2. T/E CONVERTER
3. POWER CONDITIONER HARNESS
4. FUEL CAPSULE
5. BIOLOGICAL SHIELD
6. HEAT ACCUMULATOR
7. HEAT SOURCE JACKET, UPPER SECTION
8. JACKET CLOSURE
9. HEAT SOURCE JACKET, LOWER SECTION
10. INSULATED BASE ASSEMBLY
11. INNER BUCKET
12. THERMAL INSULATION
13. HERMETIC SEAL
14. RAIN SHIELD
15. RAIN COVER
16. PROTECTIVE COVER
17. LIFTING LUGS
18. MICA CONE
19. STUD
20. CONVERTER HOT FRAME
21. STRUT MEMBER
22. SHIELD PLUG
23. BARRIER PLATE
24. CAPSULE DIFFUSION BARRIER

Figure IV-2. SNAP-23A Assembly Drawing

which are bolted together over the converter outer case to provide the necessary heat sink for the converter. The converter hot frame is tapered slightly to fit over the heat source assembly stud. A thin mica cone separates the converter from the stud in order to permit removal of the converter if desired. A hold-down bolt threads into the top of the stud and bears on a Belleville washer which in turn forces the converter onto the stud. The top of the converter and top of the stud are covered by additional Min-K 2002 which is retained by a cover that bolts to a flange on the converter. The power conditioner regulates the converter output voltage to 24 ± 1 volt and dissipates any excess power. Power transistors used for this purpose are bolted to the radiator to obtain a good heat sink.

After the assembly of the components described above, a perforated cover is placed over the converter-radiator assembly and bolted to a flange on the base assembly. The perforations permit air flow through the radiator with little impedance, but the cover is strong enough to prevent casual damage to the radiator and provides lifting eyes by which the system may be lifted.

The overall system is 24 inches in diameter and 32 inches tall. The estimated weight for the FPMS is 1288 lbs based upon calculated values for the system components as given in Table IV-1. The heat source has a nominal thermal output of approximately 1200 watts; system temperatures at end-of-life (EOL) and beginning-of-life (BOL) are shown in Figure IV-3.

C. HEAT SOURCE ASSEMBLY DETAILS

The SNAP-23A heat source, shown in Figure IV-4, is designed for absolute containment of the radioisotope fuel and effective shielding of the bremsstrahlung radiation. The design features of the heat source described below reflect the various performance and safety requirements imposed on the mechanical, thermal, nuclear, and metallurgical characteristics of the assembly. Heat source characteristic dimensions are given in Table IV-2 and temperature limitations of the critical portions of the Heat Source Assembly are given in Table IV-3.

TABLE IV-1
CALCULATED VALUES FOR SNAP-23A FPMS COMPONENT WEIGHTS

<u>Heat Source</u>		867.5 lbs
Fueled Capsule	34.3 lbs	
Shield	697.0	
Accumulator	8.0	
Barrier Plate	0.8	
Jacket	37.4	
<u>Base Assembly</u>		244.3
Inner Bucket	49.4	
Struts	11.6	
Outer Can	121.5	
Insulation	59.4	
Hermetic Seal	2.4	
<u>Top Assembly</u>		176.5
Rain Shield	8.2	
Insulation	13.7	
Instrumentation Rings	25.9	
Protective Cover	33.5	
Radiator	50.2	
Converter*	35.0	
Power Conditioner*	10.0	
<u>Total System Weight</u>		1288.3 lbs

*Predicted Values

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Figure IV-3. SNAP-23A System Temperature of Design
EOL and BOL Conditions

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Figure IV-4. FPMS Heat Source Assembly

TABLE IV-2
HEAT SOURCE ASSEMBLY CHARACTERISTICS DIMENSIONS

Overall Dimensions, inches	
Overall height	22.98
○ D (body), max	12.58
○ D (flange)	13.88
Stud diameter, top	3.295
Component Dimensions, inches	
Jacket	
Thickness, stud	0.100
Thickness, shoulder	0.190
Thickness, transition weld	0.090
Thickness, body	0.060
Barrier Plate	
Thickness	0.030
Diameter	7.75
Shield	
Length, overall	14.357
Diameter, overall	11.482
Thickness, top	2.835
Thickness, side	3.157
Thickness, plug	3.289
Length, cavity	8.233
Diameter, cavity	5.167
Diffusion Barrier	
Length	8.23
Diameter, outer	5.127
Thickness, wall	0.040
Thickness, cap	0.075
Capsule	
Length	8.108
Diameter, outer	4.974
Diameter, inner	4.163
Thickness, cap	0.592
Capsule Liner	
Length	6.928
Diameter, outer	4.094
Thickness, wall	0.032
Thickness, cap	0.065

TABLE IV-3

SNAP-23A HEAT SOURCE CRITICAL TEMPERATURES

<u>Critical Areas</u>	<u>Temperature Limitation</u>
Radioisotope Fuel	3470°F, Melting Point
Fuel - Fuel Capsule Interface	Compatibility data above 2000°F is unavailable
Fuel Capsule (Hastelloy-C)	2310-2380°F, Melting Point
Shield (U-8 Mo)	2100°F, Melting Point
Shield-Heat Accum. Interface	1742°F, Cu- U eutectic
Heat Accumulator (Copper)	1980°F, Melting Point
Inconel Components	~2350°F, Melting Point
Stainless Steel Components	~2550°F, Melting Point

Fuel Pellets. The use of SrTiO_3 was specified by the AEC for the SNAP-23A. There is a long history of experience in the use of this fuel form, particularly in the SNAP-7 series of RTG's. The original selection of this fuel form was based on considerations such as:

- (1) High melting point
- (2) Thermodynamic stability in air
- (3) Compatibility with the container material
- (4) Good thermal conductivity
- (5) Low solubility in water (a safety criterion)

The properties of the oxide and titanates, SrO , Sr_2TiO_4 , and SrTiO_3 , are relatively well established. Of these three forms, SrTiO_3 appeared to best satisfy the five criteria. More recent data show that there may be little difference in effective aqueous solubility among the three forms, since effects of radiation damage apparently change the solubility rate by a factor of 10^3 or more (Reference IV-4). However, the titanium remains a good choice based on the other four considerations: it melts at 2040°C (3600°F), is stable in air, and has the highest conductivity of the three forms, (Reference IV-5). Compatibility tests with the strontium titanate/Hastelloy-C system at 1000°C for 5000 hours produced an interaction zone of insignificant thickness.

The fuel pellets used in the SNAP-23A source are hot pressed (to approximately 85 percent theoretical density) furnished by the Isotopes Development Center of the Oak Ridge National Laboratory. In arriving at fuel pellet dimensions, it was recognized that the pellet diameter specified for the system (four inches) was significantly greater than that of any pellets previously made, and new equipment would be required to permit fabrication of this size pellet. To provide sufficient strength to avoid fracture of the pellets during handling, ORNL utilized nickel mesh disks (about ten per pellet) which are spaced along the pellet height prior to pressing. Approximately 1.0 percent SiO_2 is added to the fuel prior to pressing. This additive provides a binding quality necessary to the sintering process. Both the nickel mesh and the SiO_2 have been evaluated by ORNL and have thus far shown no deleterious compatibility effects with either the fuel or Hastelloy-C. The pellets made in accordance with this design have proved to be sufficiently strong to withstand normal handling. Nuclear

characteristics of the pellets used in the FPMS heat source are discussed in Section VIII-A.

Fuel Capsule and Liner. The specific material properties required for a fuel capsule and liner subassembly are:

- (1) Compatibility with the fuel form
- (2) Fabricability and weldability
- (3) Stability at elevated temperatures with regard to oxidation resistance

and resistance to seawater.

- (4) Strength to survive large hydrostatic pressures.

Figure IV-5 shows the fuel capsule-liner subassembly in sectioned elevation. The Hastelloy-C capsule liner is 0.030 inches thick. The fuel capsule cylindrical shell is 0.045 inch thick Hastelloy-C, and the trepanned end caps are 0.585 inch thick Hastelloy-C. The trepans permit the end caps to deform under high hydrostatic pressures and pick up support from the fuel pellet stack. The flexibility of the trepanned corner design has been further enhanced by incorporating a weld relief hole at the base of the weld. Capsule stress analyses (see Appendix-C) have established that the weld penetration at the end cap-to-cylinder joint must be a minimum of 0.110 inch to meet design conditions.

Integrity of the tungsten-inert gas (TIG) welded liner is checked by a leak test. Since the capsule shell is the prime fuel safety encapsulation, a Kr-85 leak test is performed following the final TIG closure weld. Ultrasonic tests on the weld regions are then carried out to check the weld penetration depths and soundness.

Of the oxidation resistant superalloys considered for SNAP-23A fuel containment capsules, Hastelloy-C is best qualified on the basis of long-term data relevant to the SNAP-23A environmental requirements. Tests have shown that 20 mils of Hastelloy-C will be oxidized in 50,000 hours at 1840°F (Reference IV-6). As detailed in Section VIII-C, the seawater galvanic corrosion rate of Hastelloy-C samples, tested in conjunction with other SNAP-23A materials, was measured using electrochemical techniques and found to be 1.8×10^{-3} mils per year in aerated seawater and 3.6×10^{-4} mils in deaerated seawater. Hastelloy-C was also tested in hot seawater vapor and at high hydrostatic pressure. The corrosion rate was determined to be about 7×10^{-4} mils per year (Reference IV-7). This indicates that the

Figure IV-5. SNAP-23A Fuel Capsule

capsule alone will contain the fuel for better than 300 years, or longer than 10 half-lives of Sr-90.

During the development of the SNAP-23A capsule it was noted that data for "as-cast" Hastelloy-C, which are applicable to the weld region material, show the existence of an aging phenomenon that causes the room temperature ductility to drop essentially to zero after 500 hours of aging at 1475°F (Reference IV-8). This is close to the BOL operating temperature of the lower end of the capsule. A modification in the manufacturing process for Hastelloy-C, which limits the carbon and silicon content, has eliminated this problem for operating temperatures applicable to SNAP-23A. This version of the alloy is referred to as Hastelloy-C-276. All Hastelloy-C used in the SNAP-23A program will henceforth be Hastelloy-C-276 since its basic mechanical properties are unchanged.

As noted in Section VIII-B, hydrostatic tests on the fuel capsule indicate that the assembly will not fail structurally under pressures equivalent to 20,000 feet of water over a period of 200 years (10 half-lives for Sr-90), and in addition remains structurally intact at seawater depths of 38,800 feet (the maximum recorded ocean depth) which is equivalent to 16,000 psig.

Capsule Shield Diffusion Barrier - At approximately 1370°F a eutectic exists for the nickel/uranium system. Hence a diffusion barrier between the nickel-based Hastelloy-C fuel capsule and the uranium-based shield is required when the heat source is installed in the SNAP-23A base assembly. Tantalum was selected as the barrier material on the basis of long-term diffusion-couple tests (refer to Section VIII-E). Mechanically, the diffusion barrier is similar in design to the capsule liner, but unsealed since it functions only as a separator and not a pressure vessel. The walls of this subassembly are 0.04 inch and the end caps are 0.075 inch.

Biological Shield - A nominal 3 inch thick biological shield of cast uranium-8 w/o molybdenum alloy surrounds the fuel capsule assembly. Access for loading the fuel capsule assembly into the cavity is provided by a shield plug which is "stepped" to avoid radiation streaming. Six U-8 Mo bolts are used to fasten the plug to the shield body after the fuel capsule is loaded into the cavity.

The selection of U-8 Mo is based primarily on the ability this alloy has to provide maximum shielding protection from gamma irradiation with a minimum weight and size penalty for the system. The addition of 8 w/o Mo to the uranium stabilizes the elevated temperature body-centered-cubic (gamma) phase. This reduces the dimensional severity of the uranium phase transformations and tends to increase the dimensional stability of the shield (References IV-9 and IV-10). This composition represents the best known compromise between metallurgical behavior, mechanical properties, availability, and fabricability of uranium alloys. Studies during the SNAP-23A Program at WANL have confirmed the dimensional stability and thermal conductivity of the U-8 Mo alloys. In addition, experiments have been conducted to determine the tensile ductility of U-8 Mo in the $\gamma + \alpha$ condition, which has been reported to be very low at temperatures below 1100°F. However, creep-rupture test results generated during this program indicate that strains up to 10 percent occur before U-8 Mo ruptures. This behavior indicates that the ductility of U-8 Mo at temperatures below 1100°F is a strain-rate sensitive property. Under normal operating conditions, the loadings on the U-8 Mo shield are more closely related to those imposed during the creep-rupture tests. In addition, the primary structural member for the heat source assembly is the heat source jacket. Also, the shield is in the γ condition above 1100°F during the greater part of its operating lifetime, which is a ductile phase. Therefore, the potential low ductility is not detrimental during normal system operation.

The oxidation resistance of uranium and its alloys is relatively poor in air. However, recent studies (Reference IV-11) reveal that the oxide film that is formed on a uranium-10 w/o molybdenum surface is adherent and protective at temperatures below 600°F. As noted previously, protection of the uranium-alloy shield against excessive oxidation is ensured by the hermetically sealed jacket. At the shield/accumulator interface, the metallurgical characteristics of the copper/uranium system must be taken into account. The eutectic melting point (1740°F) is sufficiently above normal and off-design temperature conditions that it does not present any compatibility problem.

The dimensions for the SNAP-23A shield were determined analytically to meet the system radiation shielding requirements for a BOL heat source of 1167 watts (174,000 curies) of SrTiO_3 . Additional shielding calculations were performed to predict the dose rates

surrounding the shielded heat source assembly loaded with the FPMS fuel capsule. Radiation tests at ORNL have verified that the shield satisfies the safety criteria. The results of the test are reported in Section VIII A while the shielding design model is described in Appendix I.

Accumulator. The primary characteristics required of the accumulator are that it have high thermal conductivity and a melting point well above its normal operating temperature.

Mechanical strength is not a significant characteristic since the accumulator is constrained by the jacket. Oxygen-free copper was selected for this application in the EMS and is retained in the current system design.

Jacket. The heat source jacket is designed to fulfill several of the basic requirements listed previously for the heat source assembly.

- (1) It provides the containment, as a hermetic seal, to protect the internals from oxidation.
- (2) It is the primary structural member, serving to support and maintain close contact between the accumulator and shield.
- (3) It provides a controlled interface with the thermoelectric converter.
- (4) It provides support points for fastening the heat source assembly to the base assembly and for fastening the converter to the stud.

The thermal expansion coefficient of the jacket is important in two ways in meeting these requirements. It is necessary to maintain the geometry of the stud-converter interface to assure good contact for heat transfer and to avoid distortions that would overstress the converter hot frame. In addition, the jacket must not expand away from the accumulator and allow gaps to occur in critical heat transfer areas. Type 304 Stainless Steel is utilized as the jacket head material. Its coefficient of thermal expansion is well matched to the OF Copper accumulator, preventing either high stress concentrations or large gaps at operating temperatures.

A high quality, structural weld joint is required to support the full weight of the fuel capsule, shield, and accumulator under static as well as dynamic conditions. It must also be a good seal weld to retain the hermetic sealing characteristics of the jacket. A butt

weld is utilized in preference to a fillet weld because of its higher resistance to thermally induced moments. The current butt weld design is shown in Figure IV-6. The local 304 SS material thickness is 0.090 inch thick as compared to the 0.060 inch thickness of the lower portion of the Inconel 718 jacket; this compensates for the weld efficiency factor.

During the final assembly of the heat source assembly, a preload of 1,000 pounds is applied to the jacket closure plate prior to welding. The shield plug is designed to bear directly on the capsule, thus assuring good axial heat conduction from the capsule to the shield. This preload also combines with the weight of the shield to compress the shield-to-accumulator and accumulator-to-jacket interfaces (the assembly is inverted during the closure weld process). Thus, good initial thermal contact is achieved.

The jacket closure plate weld design, is a modified butt weld joint design and contains a relief cavity to eliminate the "incipient crack" effect of the necessary annular assembly gap between the periphery of the end closure and the heat source jacket opening. The weld configuration was coordinated with ORNL, since this weld is made at ORNL after the fuel capsule is loaded into the heat source assembly.

D. LOWER BASE ASSEMBLY

The lower base assembly shown in Figure IV-2 supports both the heat source assembly and the thermoelectric converter and is designed to minimize thermal losses between the heat source and the thermoelectric converter.

The base assembly consists of an inner bucket supported by a tubular truss support system within a hermetically sealed container filled with thermal insulation.

The inner bucket and the hermetic seal materials are made from Inconel 600, since they do not require the higher strength of Inconel 718 and since Inconel 600 has a relatively low work hardening rate. Inconel 600 is also oxidation-resistant to 2150°F. The hermetic seal is located at the level of the heat source support flange and is a thin (approximately 0.010 inch thick) convoluted sheet that provides a gas tight closure between the inner bucket and the outer shell. The convolutions provide flexibility so that differential thermal stresses between the bucket and shell will not overstrain the material.

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Figure IV-6. Heat Source Assembly Girth Weld Scheme

The support system consists of eight struts connecting the inner bucket to a heavy ring at the periphery of the bottom of the base assembly outer shell or container. These struts are hollow tubes, providing the required tensile and compressive support with the minimum cross-sectional area-to-length ratio, and, therefore, minimizing both heat losses and weight. The material selected for the truss legs and the inner support ring in the bucket is Inconel 718, which, because of its slow response to age hardening, will retain its elevated strength properties up to 1300°F for extended time periods and requires no heat treatment after welding for corrosion protection or resistance to age hardening.

The outer shell of the base assembly is manufactured from Inconel 625 sheet. The 0.125 inch shell thickness is a stock size adequate to withstand the various imposed loads such as: 1) when the system is lifted (by the radiator protective cover); 2) the collapsing pressure due to the below-atmosphere pressure in the base assembly; and 3) the specified shock and vibration loads. This shell is welded to the top and base rings which are also made from Inconel 625. The top ring includes bolt holes for the attachment of the rain shield covering the upper insulation assembly. The ID of the ring is welded to the Inconel 600 hermetic seal. In addition to the provision for an O-ring seal, the top ring also provides a space for a degassing tube between the O-ring and the hermetic seal. This avoids any projection outside the 24 inch system diameter.

The base cover is a simple disk welded to the base ring. It is also made from 0.125 inch stock Inconel 625.

The insulation specified for use in the lower base assembly is Min-K 2002. This material was selected on the basis of satisfying the requirements of low thermal conductivity, and dimensional and chemical stability during the design lifetime.

Supplied in 2 inch thick sheets, the orientation of the Min-K slabs are arranged within the base assembly to provide the maximum thermal impedance in the primary directions of heat leakage (horizontally through the side and vertically through the bottom), taking into account the anisotropy in conductivity which is lower perpendicular to the surface of the sheet than in the parallel direction.

By hermetically sealing the base assembly container, a krypton atmosphere may be used instead of air. This results in a significant decrease in the effective Min-K conductivity.

Axial grooving has been provided at the outer periphery of the base assembly, connecting with small annular reliefs in the area of the top and bottom rings to improve pressure communication for efficient purging and backfilling after base assembly closure welds are completed. Upon proving the hermetic sealing capability of the insulated base assembly, the unit will be heated and evacuated to remove air and moisture, backfilled with the krypton, and the fill tube pinched off and seal welded.

E. UPPER INSULATION AND RAIN SHIELD ASSEMBLY

The upper insulation and rain shield assembly, shown in Figure IV-7, surrounds the upper portion of the heat source jacket with Min-K 2002 thermal insulation, thereby reducing direct heat losses from the heat source to the radiator assembly and thermoelectric converter. It is physically located between the top of the base assembly and the bottom flange of the thermoelectric converter (see Figure IV-3). Due to assembly considerations, this insulation will not be hermetically sealed like the base assembly insulation. However, a mechanically sealed insulation cover, the rain shield, is provided to seal out moisture, dust and other airborne contaminants which could otherwise be injurious to the satisfactory thermal performance of the insulation.

The rain shield consists of a 30° cone of elastomer material which connects the lower flange of the converter to the rain shield outer ring which is an Inconel 625 sheet metal transition member that attaches to the base assembly top ring. The elastomer selection for the rain shield is RTV-60, a silicone rubber which has adequate properties up to 400°F or higher. This is a room temperature vulcanizing material which can easily be molded in the desired shape and thickness.

In the cavity under the rain shield, bounded by the heat source jacket head, the base assembly convoluted hermetic seal, T/E converter bottom-closure member and the rain shield itself is filled with Min-K 2002 block insulation cut from 2 inch thick assembly. A thin deformable layer of microquartz insulation in blanket form is placed over the top of the Min-K 2002 assembly as a compliance member.

Figure IV-7. Insulation Under Heat Source Rain Shield-
Min-K 2002 Configuration

F. CONVERTER ASSEMBLY

The function of this component of the SNAP-23A system is to use the heat transferred from the radioisotope to the heat source stud for conversion to electrical power and to reject the excess heat to the environment. The electrical power generated by the converter is delivered to a voltage regulator mounted on the radiator and then to the external system load.

As shown in Figure IV-8, the converter is provided with a tapered 304 SS hot frame which fits around the stud of the heat source. A thin mica sleeve is inserted between the stud and hot frame to accommodate interface irregularities and to facilitate removal of the converter at disassembly. Boron nitride strips attached to the hot frame electrically insulate the thermoelectric couples from the hot frame structure.

Electricity is generated in 280 couples surrounding the hot frame in 40 equally space rows each containing 14 elements (7 couples). One of these rows is visible in Figure IV-8.

Each couple contains two segmented thermoelements consisting of lead telluride and bismuth telluride materials. The hot side of each element consists of PbTe material, while the cooler end contains BiTe material. Combining the best features of both thermoelectric materials in a single thermoelement results in a thermopile with high thermal efficiency.

Electrically connecting the 560 elements in series, yields an open circuit voltage of approximately 50 V.D.C. Under load, the generator is regulated by a power conditioner to 24 V.D.C., for the entire lifetime of the system.

Beryllium oxide ceramic rods isolate the thermoelectric circuit at the cold side, while conducting waste heat into the aluminum cold frame. Follower springs are used to maintain pressure (greater than 100 psi) through the contacts at the hot shoe and cold cap interfaces.

Powdered Min-K insulation is vibrated into the voids of the thermopile to thermally insulate the aluminum cold frame from the hot frame. Block insulation is also used to reduce the thermal losses in the vicinity of the upper structural supports (Figure IV-8).

Figure IV-8. SNAP-23A 60-Watt Thermoelectric Converter (PMC-2 Configuration)

The outer case of the converter is Inconel 625, and the hot frame inner housing is 304 stainless steel which has thermal expansion characteristics similar to the copper stud of the heat source assembly.

Table IV-4 lists the performance and temperature characteristics for the SNAP-23A converter at BOL and EOL. Although the efficiency changes very little, the power decreases from 80 watts to a minimum of 61 watts over the 10 years.

G. RADIATOR ASSEMBLY

Waste heat from the thermoelectric converter and power conditioner is rejected to the environment by the radiator using natural convection of ambient air. The radiator is contained within a perforated protective cover and interfaces mechanically with the converter. Thermal grease is used between the Inconel 625 converter outer shell and the aluminum radiator hub to enhance transfer of heat. The SNAP-23A radiator consists of 138, radially oriented fins, brazed to a split cylindrical inner hub and constrained externally by two outer rings. Both the hub and outer bands(shrouds) are split to facilitate mounting the radiator on the outer case of the converter. Bolted flanges on the hub and connecting plates (snubbers) on the shrouds are used to fasten the two halves of the radiator assembly together. The shroud at the uppermost portion of the finned radiator assembly functions as a mounting point for, and dissipates heat from, the power conditioning components. Figure IV-9 illustrates the assembly of a typical SNAP-23A radiator assembly.

H. PROTECTIVE COVER

The outer protective cover is a right circular cylinder (24 inches diameter by 13 inches high) made of perforated Inconel 625 sheet, closed at the top end and open at the bottom. The cover attaches to the upper-most flange of the base assembly and protects the T/E converter, radiator assembly, and power conditioning equipment from impact damage. Three equally spaced lifting points are provided in the protective cover so that the entire SNAP-23A system may be raised by sling and hoist.

The cover has approximately 60 percent void to allow adequate flow area for the normal circulation of air through the finned radiator.

TABLE IV-4
SNAP-23A CONVERTER DESIGN CHARACTERISTICS (REFERENCE B-2)

	<u>Beginning-of-Life</u>	<u>End-of-Life</u>
<u>Temperatures (°F)</u>		
Average Hot Frame	1072	911
Maximum Hot Junction	1040	---
Average Hot Junction	1022	866
Average Cold Junction	196	176
Outer Case	161	146
<u>Performance</u>		
Efficiency (percent)	7.84	7.69
Power Output (watts)	80.2	61.3
Heat Input (watts)	1022.	797.
Resistance (ohms)	8.90	7.05
Open Circuit Voltage (volts)	53.73	42.01
Load Voltage (volts)	24	24

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Figure IV-9. Dip-Brazed Aluminum Radiator Assembly

Power Conditioner - Output voltage from the SNAP-23A converter is continuously controlled at 24 ± 1 volts D. C. for the 10 year period of operation. The power conditioner system functions as a voltage limiter to maintain constant voltage regardless of decreasing power output or loss of load. At the end of life, the power conditioner is designed to accept 61 watts(e) input and deliver 60 watts(e) output.

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V. SHIPPING CONTAINERS

A. Heat Source Shipping Container

The shipping container for the SNAP-23A Heat Source Assembly is designed to satisfy the DOT/AEC requirements for normal transport and for hypothetical transport accidents. The shipping container is also intended to satisfy SNAP-23A Safety Criteria which, for certain environments, are more severe than DOT/AEC regulations. The primary features of the design are:

- Mechanically, the shipping container will survive drop, puncture and compressive loads for normal transport and hypothetical transport accidents.
- Thermally, the shipping container will maintain the heat source surface temperature at 1100°F minimum, operate at or below 180°F on the outside surface during transportation in an exclusive use vehicle and survive a 1475°F/30 minute fire without impairing the capsule's structural integrity or the biological shield's effectiveness.
- Radiation levels will be 200 mr/hr or less at the containers outside surface and 10 mr/hr or less 6 feet from the outside surface.

1. Inner Container

The entire Shipping Container Assembly for the heat source is shown in Figure V-1. The Heat Source Assembly is held by a framework of Inconel-718 support members, consisting of three radial members and three axial tension members that interface with a 304 stainless steel adapter ring. The adapter ring is secured to the heat source assembly at the jacket assembly plane by 24 steel bolts.

All six support members are fastened to the inner container. The radial members are bolted to a support ring which forms part of the inner container shell structure. The axial tension members are fastened to the ends of the inner container with clevis pins and tension bolts.

- LEGEND**
- ① SECURITY SEAL
 - ② AL WOOL COMPLIANCE
 - ③ HONEYCOMB ENERGY ABSORBER
 - ④ OUTER CONTAINER
 - ⑤ INNER CONTAINER
 - ⑥ VERTICAL SUPPORT MEMBERS
 - ⑦ ADAPTER RING
 - ⑧ ALUMINUM SHIMS
 - ⑨ THERMAL INSULATION
 - ⑩ COVER
 - ⑪ OUTER CONTAINER FLANGE
 - ⑫ TIE DOWN LUGS
 - ⑬ HEAT SOURCE ASSY
 - ⑭ LATERAL SUPPORT MEMBERS
 - ⑮ SAND
 - ⑯ THERMOBESTOS INSULATION

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Figure V-1. Heat Source Shipping Container Assembly

The volume between the heat source assembly and the inner container is filled with Ottawa sand packed to approximately 110 lbs/ft³. The sand provides load distribution properties which protect the heat source assembly in the event of a transport accident. The support members, described in the previous paragraph, are designed to yield under a 15g load. This feature is ensured by the machined slots in each member. Once the supporting tension members have buckled or yielded, the heat source assembly rests in the load-distributing sand and high local stresses on the heat source assembly are avoided.

The inner container is a right circular cylinder, nominally 33.75 inches high and 21.0 inches in diameter. The cylindrical shell end caps are CORTEN A steel, 0.375 inch thick. The cylindrical section is fabricated in two halves and joined by full penetration, circumferential welds at the support ring. Each half is rolled from sheet stock and closed by a full penetration, seam weld. The top plate and the bottom plate are cut directly from sheet stock.

Mounting flanges are welded to the top and bottom of the cylindrical section. Both inside and outside fillet welds are employed for added strength. The top and bottom plates are bolted to the flanges with 18 steel bolts each.

The quality of all welds in the inner container is assured by a weld qualification test and a visual inspection.

2. Thermal Insulation

A one inch thick, Thermobestos insulation pad is located immediately below the inner container to reduce axial heat losses. Temperature control of the heat source is achieved by varying the amount of thermal insulation in the 0.75 inch annulus provided between the inner container and the surrounding honeycomb. Sufficient thermal insulation is placed in the annulus to stabilize the heat source shield temperature at approximately 1100°F. The balance of the annulus is filled with aluminum shims. The actual thickness of insulation and shim material is adjustable to compensate for ambient temperature extremes.

3. Honeycomb

An eleven inch thick, honeycomb cylinder surrounds the inner container. Additional honeycomb is provided above and below the inner container. The cylindrical honeycomb is fabricated in quadrants so that the core direction is oriented to radiate from the shipping container's centerline. The inner and outer honeycomb surfaces are epoxy bonded to 0.030 inch thick aluminum sheet. Each quadrant is formed from two concentric quadrants bonded to 0.030 inch thick, aluminum separators. The honeycomb cylinder extends about five inches above and below the inner container. The advantage of this configuration is an inherently greater resistance to corner impact. The recesses at the top and bottom of the quartered honeycomb cylinder are filled with axially oriented honeycomb discs, about five inches thick. The remaining end spaces are fitted with 12 inch thick axially oriented honeycomb slabs. The honeycomb must be kept at a reasonably low temperature to retain its crushable property. To maintain this temperature, aluminum wool is employed as a filler material in the concentric and diametral gaps between the honeycomb and outer container. Its purpose is to ensure good heat transfer to the outer container by providing conductive paths across the gaps. The wool is not intended to retain a preload or provide other restraining forces.

4. Outer Container

The outer container is a cylinder, about 70 inches high and 42 inches in diameter. It is rolled from 0.125 inch carbon steel sheet stock and welded with a full penetration seam weld. The cylinder edges are reinforced with box flanges for extra rigidity at the cover interfaces. The covers are 0.125 inch thick, CORTEN-A steel sheet. These covers are attached to the cylindrical section with 135 steel bolts each.

Three security seals, positioned 120 degrees apart, are employed on each end of the shipping container. The seals are wired between pairs of adjacent bolts. Vandalism or unauthorized access into the container will be obvious by damage to the seals.

The box flange on the top cover is provided with four, one half inch thick, tie down lugs. Each lug is fillet-welded in place. The welds extend eight inches along the top and two-and-one-half inches along each side of each lug. The lugs are arranged to facilitate the application of tie down bars tangent to the box flange.

5. Tie Down Device for Truck Transport

The tie down device is designed to meet the requirements of 10CFR 71.31, viz.; 2g's vertical, 10g's foreward, and 5g's transverse, without generating stress in any material of the package in excess of its yield strength. Stress analyses (Reference V-1) have been performed to verify conformance with these requirements.

The tie down device is shown in Figure V-2. A 3/8 inch thick, steel base plate is bolted to the truck platform. Lateral jacking screws are provided on the plate to initially center the heat source shipping container. Four tie down fittings are anchored to the base plate. A solid steel tie rod, 1.25 inch diameter is threaded into each clevis which, in turn, is pinned to a tie down fitting. A second clevis is threaded on the opposite end of each tie rod and pinned to one of the tie down lugs on the shipping container. As shown in Figure V-2, the tie rods cross each other to provide additional support. Each tie rod linkage is placed in tension at the completion of the assembly procedure.

V-6

Figure V-2. Heat Source Shipping Container Tie Down System

B. SNAP-23A System

A preliminary shipping container design concept has been formulated for the SNAP-23A System. A cubic configuration was selected for the overall container to minimize terminal velocity in the event the container is subjected to a free fall drop following an airborne transport accident. Initial sizing has been conservative. Material thicknesses and strengths do not consider the impact resistance of the heat source or the energy absorption of the generator components. A secondary design goal has been to make the shipping container adaptable to multiple shipping methods.

It must be emphasized that the design has not been optimized. Materials selection, cube size, and energy transference must be further investigated before the concept can be translated to final design and hardware.

1. Design Description

The shipping container concept is shown on Figures V-3 and V-4. The SNAP-23A System is housed within a four foot cube. It is planned that the four foot cube, or inner container, will be the complete shipping container for all transport modes except aircraft. The inner container is made with a commercially available aluminum shipping pallet and aluminum box beams. High damping shock and vibration isolators separate the generator mounting plate from the shipping pallet. The generator is bolted directly to the mounting plate. Additional ties are provided for generator side mounts if they should prove necessary. The SNAP-23A System is encased in 5052 aluminum honeycomb. The honeycomb selection is preliminary and will provide a crush strength of 540 psi. The box beam frame is covered with aluminum sheet for added rigidity and better energy distribution properties. Cooling air passages are provided in the container so that generator operating temperatures will remain within allowable limits.

The honeycomb within the inner container is oriented radially along the sides of the SNAP-23A System. A vertical orientation is used at top and bottom. The component parts will be designed for ease of assembly and disassembly. The inner container is reusable barring major accidents.

Figure V-3. SNAP-23A System Shipping Container Conceptual Design

Figure V-4. SNAP-23A System Shipping Container Conceptual Design Layout

The inner container assembly is entirely surrounded by the outer container. The outer container is necessary only if air transport is a requirement but can be employed for any and all shipments if so desired. The outer container is a faced honeycomb structure. Facing is utilized to make the honeycomb rigid in the shear plane and in bending and it also aids in distributing impact forces. The honeycomb is fabricated as six, flat faced sections and eight pyramid sections. Twelve, edge sections may be required if the detail analysis so determines. The pyramid sections, placed on each corner of the cube, are made from high density, high crush strength honeycomb. The flat faced sections which are the sides of the outer container, are made from a lighter, lower crush strength material. The inner container is connected to the outer container by bolts which fasten into the inner facing on the bottom honeycomb section. The facing is reinforced with pads in this area to ensure a secure anchorage. Both the inner facing and the outer facing are provided with sufficient air passages to cool the generator.

2. Preliminary Impact Analysis

An impact analysis has been performed to justify the design concept. The shipping container has been examined semi-parametrically for size, weight, terminal velocity, and impact energy. The analysis has assumed that all impact energy will be absorbed in crushing the honeycomb and does not consider deformation of the other structure or the SNAP-23A System.

Honeycomb is a uniaxial material and its energy absorption capability falls off rapidly as the axis of impact moves away from the core axis. Part of the analysis, then, has been devoted to minimizing the off-axis effects of impact. The most vulnerable part of a cube during impact is the corner. The ratio of volumetric compression to deformation distance is smaller on a corner impact than on a side or edge impact. Therefore, the corner has been assumed to be the worst case and has been analyzed in depth.

Cubes of six, seven, and eight foot sides have been considered. Average honeycomb densities used in the cube, range from four to ten pounds per cubic foot. Weight and terminal velocity have been calculated for each cube utilizing an assumed generator and

structure weight of 1500 lbs and an average honeycomb density for the balance of the volume. Impact energy has been computed as the total kinetic energy of the falling body from the expressions:

$$E = 1/2 \cdot \frac{W}{g} \cdot V_t^2$$

$$\text{and } V_t = \left(\frac{2W}{\rho A C_D} \right)^{1/2}$$

where:	ρ	=	Air density at sea level
	A	=	Side area of cube
	C_D	=	Subsonic drag coefficient of a free falling cube
	W	=	Total weight
	g	=	Gravitational constant

Figure V-5 shows the results of the energy and terminal velocity calculations for six, seven, and eight foot cubes. The average honeycomb density has been varied from four to ten lb/ft³. Figure V-6 was prepared using the data from Figure V-5 and the energy absorption characteristics of 5052 aluminum honeycomb. The impact energy developed by the body is a function of mass and terminal velocity. Terminal velocity increases directly with the mass and inversely with the projected area. Since the mass of the honeycomb grows with the cube of the side dimension and the projected area increases with the square of the side dimension, there will be an optimum combination of honeycomb density and cube size for the system shipping container.

In the preliminary analysis of the shipping container, no distinction has been made regarding whether the outer honeycomb crushes from an outside force or if the generator crushes the outside honeycomb from the inside. Figures V-5 and V-6 can be employed together to obtain the crush characteristics of the various configurations.

Since the corner has been selected as the most critical impact area, special design precautions have been taken to protect it. The honeycomb in the corners is aligned

Figure V-5. Weight Versus Impact Energy as a Function of Cube Size and Average Honeycomb Density

Figure V-6. Crush Volume Versus Honeycomb
Density for Various Cube Sizes

with the most severe impact angle, i.e; the impact angle at which the crush depth is greatest. The crush depth is a function of the total volume crushed. The relationship between the corner crush depth and the corner crush volume is $Vol = 0.855 (\text{crush depth})^3$. The expected corner crush depth as a function of total impact energy available and honeycomb density is shown in Figure V-7. Adequate protection of the corners requires the use of high density honeycomb in the corner volumes. By using Figures V-5, V-6 and V-7, the overall cube size and required honeycomb densities can be selected as a viable configuration. Optimization of the design will require trials with different cube sizes and honeycomb densities. The design will be considered optimized when the cube size has been reduced to a minimum and the g-loads transmitted to the system are not sufficient to damage the heat source.

3. Effects of Thermal Environments

The SNAP-23A System shipping container is designed to permit adequate convective cooling of the SNAP-23A System during normal handling and transportation operations. Cooling air passages of the inner and outer container are sized to permit the converter to operate within design temperature limitations. Fire is the most significant abnormal thermal environment which must be evaluated. The shipping container design considers the possible effects of an 1850°F, one half hour fire.

Ease of handling, and low impact velocity following free fall, require that the shipping container be constructed of relatively low density material. The major part of shipping container weight is 5052 aluminum honeycomb material. Structural supports and facing sheets are also made from aluminum. Exposure of the system shipping container to an 1850°F fire will eventually result in shipping container exterior temperatures which are in excess of aluminum melting temperature. The low melting point of aluminum is not anticipated to be a disadvantage since the shipping container does not have to survive a severe fire environment. Rather, the container must protect the enclosed SNAP-23A system and this will be accomplished through several aspects of the container design.

(a) The high thermal conductivity of the aluminum material will allow the entire mass of the container to be utilized as a heat sink.

Figure V-7. Energy Available Versus Corner Crush Distance

(b) The mode of heat transfer from a large fire to the generator system will normally be a combination of convection and radiation. Radiation is of the most concern if an appreciable area of the radiator fins see the fire environment. The shipping container will be designed to eliminate most of the view between the radiator surfaces and the external environment. Computer studies made for the SNAP-23A system (without container) have indicated that this approach will be adequate. That is, the aluminum radiator will not be destroyed if the radiation input is inhibited. The preliminary design of the shipping container utilizes aluminum facing sheets which act as radiation barriers. A higher temperature (but still light weight) material such as titanium can be substituted if necessary.

In summary, the aluminum container will not provide indefinite thermal protection for the SNAP-23A System, but it will give the desired fire retardation effects to completely satisfy safety criteria.

VI ACCIDENT ANALYSIS

A. INTRODUCTION

The analyses in the following sections provide the background required for assessing the overall nuclear safety of the SNAP-23A system. The scope of the safety analysis is consistent with the operations analysis study of Reference VI-1 and covers those areas which are associated with a potential nuclear safety hazard or problem. In general, there have been substantial revisions and updating of the data contained in Reference VI-2, the PSAR. It is now possible to assess certain safety problems in terms of available test data and more refined accident models than were heretofore available.

Where appropriate, bulky analyses and data compilations have been placed in appendices of this report. Detailed test results used to support the accident analysis are contained in Section VIII of the report.

All analyses use the FPMS as the reference configuration. Where applicable, shipping containers form part of the configuration evaluated. For the heat source, the shipping container is well defined. The shipping container for the SNAP-23A system is described and justified in Section V-B. However, the configuration for the system container is preliminary and one of the purposes of this report, is to better define the properties of the container.

Accidents have been categorized by constructing simplified accident networks for the heat source and the SNAP-23A system. The networks identify the probability associated with accidents occurring at a specific phase of the SNAP-23A lifetime. The extent of the probabalistic approach is sufficient to justify which type of accident is most likely.

Considerable emphasis in this report has been placed upon air transportation. The combination of response analyses and probability of occurrence provide the background for a judgment to be made as to whether the SNAP-23A can be safely deployed with unrestricted air transportation.

B. HEAT SOURCE EVALUATION

1. Accident Network

Simplified accident networks have been prepared for the heat source to identify

and categorize credible malfunctions and/ or accidents. The networks roughly cover the time period from heat source assembly at Oak Ridge to SNAP-23A system assembly at Westinghouse.

Post-mission (recovery) handling, storage, test and transportation have not been specifically considered in the accident networks. The results presented here and in Section VI-C1 can, however, be easily extended to post-mission probabilities for the appropriate sequence of events. Figure VI-1 demonstrates the general network division utilized. The total probability of one accident for a particular heat source is estimated to be 3.14×10^{-2} . This probability value is obtained from the sum of the two branches and includes all types of accidents regardless of severity. The way in which the probability

Figure VI-1. Accident Occurrence Probability for Heat Source Handling, Storage, Test and Transport

values were obtained is discussed below. The further breakdown of the two categories of accidents is shown by Figures VI-2 and VI-3.

a) Handling, Storage, and Test. Handling, storage, and test of the heat source involves two possible configurations depending on whether or not the shipping container is involved.

The events which lead to an accident in this category are a fire, explosion, earthquake, equipment mishandling or failure, operator error, and a test mishap. The first three environments are time dependent (the longer the cask or heat source remains at a location, the higher the probability of a fire, natural disaster, or explosion occurrence) whereas an equipment, test mishap or operator error depends on the amount of manipulation and testing required. An estimate of the absolute probabilities appropriate to this

Figure VI-2. Heat Source Handling Storage and Test Accident Network

Figure VI-3. SNAP-23A Heat Source Transportation Accident Network

phase is summarized in Table VI-1 and is based on detailed analyses conducted for the TRANSIT program (Reference VI-3). While the TRANSIT heat source and test environment are significantly different from those of the SNAP-23A, these probabilities can nevertheless be considered indicative of the accident risk associated with handling, testing, and storage of the SNAP-23A heat source.

TABLE VI-1
HANDLING, TEST AND STORAGE
ABSOLUTE ACCIDENT PROBABILITIES

<u>Accident Environment</u>	<u>Heat source in shipping container</u>	<u>Heat Source alone</u>
Fire	12.0×10^{-4}	4.00×10^{-4}
Explosion	0.3×10^{-4}	0.10×10^{-4}
Earthquake	1.0×10^{-4}	0.33×10^{-4}
Handling (equipment failure and operator error)	200.0×10^{-4}	30.00×10^{-4}
Testing (equipment failure and operator error)	----	50.00×10^{-4}
Totals	213.0×10^{-4}	84.00×10^{-4}

Total handling, storage, and test accident probability = 3×10^{-2} .

During storage, the heat source was assumed to be in the shipping container. Since the greatest amount of time is spent in storage, the time dependent accidents prevail for the heat source in the shipping container configuration. In the case of fueling, assembly, and testing of the heat source alone, the failure or misuse of equipment and test mishaps are the prevailing accident causes. Adding both columns results in a total probability of

2.97×10^{-2} for the heat source handling storage and test accident.

From the total of the first column, the probability that the heat source in its container will be involved in the handling, storage and test accident is 2.13×10^{-2} . From the total of the second column, the probability of the heat source alone in the accident is 8.4×10^{-3} .

b) Transportation. One of the guidelines for the SNAP-23A is that the heat source be transported from ORNL to WANL by truck. The shipping distance per heat source is therefore, fixed at 480 miles and thus all probabilities in Figure VI-3, the transport accident network, are based upon this distance.

Extensive studies of all trucking accidents (Reference VI-4) indicate accident rates ranging from 3.64×10^{-6} to 3.83×10^{-6} per mile of travel. Statistics have also shown that the accident rate for trucks carrying dangerous articles, such as the truck which will carry the heat source, is somewhat lower, varying between 3.35×10^{-6} and 3.55×10^{-6} per mile of travel. The value 3.5×10^{-6} was selected as a reasonable and conservative rate for the shipments of interest since they are most likely to be in a specific van. The total number of miles which any heat source in its container is likely to be transported is 480. Therefore, the transportation accident probability is $3.5 \times 10^{-6} (480) = 1.67 \times 10^{-3}$.

Reference VI-4 subdivides truck accident types and shows that of all such accidents, 14 percent will be head-on collisions with autos or other trucks. Thus, the probability of a head-on collision with another truck is approximately 2.4×10^{-4} . Collisions involving dangerous cargo are a special case of the preceding one. A total of 16 percent of all truck accidents involve collisions with other trucks (three percent are head-on and 13 percent are rear end and angle collisions). Of these truck versus truck accidents, it is estimated that a maximum of 10 percent of the second trucks will be carrying dangerous cargoes (flammables, explosives, etc.) Therefore, $(0.16) \times (0.10) \times (1.67 \times 10^{-3}) = 2.7 \times 10^{-5}$ is the probability of the collision involving a dangerous cargo. Finally, the probability of other truck transportation accidents is:

$$(1.67 \times 10^{-3}) - (2.4 \times 10^{-4}) - (2.7 \times 10^{-5}) = 1.4 \times 10^{-3}$$

It is possible to obtain probabilities for each potential environment which results from the three initiating events. For example, it is shown in Part E of Appendix F that the probability of an accident producing a secondary explosion is $\sim 4.4 \times 10^{-3}$. Thus, the probability of secondary explosion following collision between the truck carrying the heat source and another with a dangerous cargo, is $(4.4 \times 10^{-3})(2.7 \times 10^{-5}) \approx 1.2 \times 10^{-7}$.

2. Response to Handling Storage and Test Accidents

As noted on the accident network (Figure VI-2) for the heat source, two configurations must be considered for the handling, storage, and test phase: the bare heat source and the heat source in its shipping container.

a) Impact. The credible accident during handling of the bare heat source has been postulated as free fall to impact from a height of six feet. The accident could occur, for example, when the heat source is removed from its shipping container and placed into the generator base assembly. The impact attitudes of the heat source in the six foot drop are assumed to be top, bottom, side, and corner. A detailed description of the analytical results appears in Appendix G. The method of analysis utilized is based upon the energy capacity of the material and the dynamic stress at impact. Conservatism is obtained by purposely ignoring the resistance and strength of the outer Inconel/steel shell and the attenuation of the copper heat accumulator. In the top impact case, the resulting stress on the U-8 Mo shield is only 20 percent of the yield strength at 1100°F . The dynamic stress at the fuel capsule-to-shield interface is 16,500 psi for a bottom impact: these stress levels will not compromise shield integrity. Corner impact at the bottom edge of the heat source represents the most severe impact orientation. As noted in Appendix G, this case is not amenable to rigorous analytical solution, but was qualitatively evaluated from the results of the first two cases. Severe damage to the shield was judged to be unlikely if corner impact occurs.

b) Crushing and Debris. Crushing environments on the heat source with and without its shipping container can occur if plant lifting and transport equipment fail or are mishandled. The most severe case would be heavy equipment such as a chain hoist falling

due to mechanical failure or building collapse. Crushing environments can also result from earthquakes or other natural events which could cause structural failure of heavy equipment or building members. At this time, the impact environments and responses described above are felt to be equivalent in severity to the crushing and debris effects.

c) Explosion. The response of the heat source without container to a severe blast overpressure environment has been evaluated. A two percent TNT yield explosion at a distance of 30 feet has been defined in Appendix F to be the SNAP-23A accident model for an explosive environment.

(It is not strictly correct to use this environment to represent the handling, storage, and test phase of the bare heat source. The environment was derived for a transportation accident and is more severe than will be encountered by the unprotected heat source.) The integrity of the Inconel 718 jacket without the internal support of the shield/capsule assembly has been determined using a failure criterion for blast-loaded cylindrical shells. The criterion is the same as that used in Appendix F for the SNAP-23A system. Based on the dimensions of the jacket, the energy release from the reference explosion and the mechanical properties of Inconel 718, the critical pressure, p_{cr} , from the blast required to fail the shell has been computed to be 2120 psi. Since the incident overpressure, $p = 73.5$ psi, from the explosion is much lower than p_{cr} , the jacket alone could survive the blast with negligible radial deformation. In the actual case, the jacket shell can withstand even higher external pressures than p_{cr} since it is internally stiffened by the heat accumulator and the biological shield. It is concluded that the heat source will remain intact after experiencing the defined blast overpressure.

d) Fire. The occurrence of a serious fire has been assumed to be credible for handling, storage, and test of the heat source, as well as for most other phases of the SNAP-23A system assembly and lifetime.

Initial temperatures of the bare heat source have been determined by assuming that the heat source is in an upright condition and rejects heat to a laboratory or assembly area environment at 70°F. The results of the steady state evaluation are shown in Figure VI-4

Figure VI-4. Steady State Temperature of Upright Heat Source in Air

and assume that the heat source rejects heat by natural convection and radiation with the Inconel jacket having an emissivity of 0.5.

The transient temperature response of the heat source has been evaluated for exposure to the two specific fire environments. In Figure VI-5, the transient temperatures are presented for the heat source in a 1475°F minute fire. (The fire is similar to the hypothetical fire specified by AEC and DOT transport regulations.) Exposure of the bare heat source is highly unlikely since the fire would have to occur in a controlled laboratory test cell or assembly area environment. However, the results reveal the inherent resistance of the heat source to thermal damage. As shown in Figure VI-5, none of the heat source components reach damaging temperatures. Cooldown after the one-half hour fire is by radiation and convection. The bottom or resting surface is assumed to remain adiabatic. Emissivity of the heat source exterior has been taken as 0.8 during the fire and during cooldown. This assumption is quite conservative during the fire or until the surface oxidizes.

Figure VI-6 indicates the response of the heat source to a so-called standard fire. The fire was characterized by the time-temperature relation described in Appendix B. Cessation of fire and heat source cooldown in air was assumed to occur after 1.0 hours. As might be expected, the temperatures caused by the standard fire are somewhat higher than for the 1475°F fire but do not cause heat source damage. Temperatures at all locations in the heat source remain well below local melting point or eutectic formation limits.

e) Thermal Stress in the Radiation Shield. A thermal stress evaluation of the U-8 percent Mo radiation shield was performed. The thermal cycle for the heat source was chosen as a hypothetical accident case yielding the worst thermal gradients in the U-8 percent Mo shield. As an initial condition, the heat source was assumed to be at the temperatures corresponding to the one-half hour point of an 1850°F fire. During the fire, the heat source is inside an open circuited generator. At time zero, the generator assembly is separated and the bare heat source plunged into water. A rapid cool-down of the heat surface occurs as a result of boiling heat transfer. The thermal analysis assumes the convective heat transfer coefficient to vary with surface temperature, thus changing rapidly as

Figure VI-5. Heat Source Temperature Profile During and After a 1475°F Fire

Figure VI-6. Heat Source Temperature Profile During and After a Standard Fire

the boiling goes from partial film to nucleate. Temperatures for a period of several hours after immersion were determined by using the heat source analytical model described in Appendix B. The temperatures at one radial plane through the heat source are noted as a function of time and radius in Figure VI-7.

The temperature distribution as a function of time which yields the largest temperature gradient through the U-8 percent Mo biological shield has been analyzed by the FEATS computer code to obtain thermal stresses within the assembly. (The FEATS code is a Westinghouse developed program which treats finite elements as shown in Appendix C.)

The analytical model used for this thermal stress calculation is shown in Figure VI-8. Table VI-2 lists the material properties as a function of temperature, and Table VI-4 illustrates the stress distribution determined by the FEATS code at the worse case, 0.05 hour point. The effective stress tabulated in Table VI-3 is the stress intensity derived by the yield criteria of Von Mises as determined by the computer code. Reference VI-5 defines the code and application of the yield criterion. Minus signs indicate compressive stresses and those values with no sign are positive or tensile stresses. The effective stress is always positive due to the mathematical relationship of the yield criterion.

During the cooling process, the outer Inconel shell and intermediate copper layer, which have higher coefficients of thermal expansion, contract more than the U-8 Mo biological shield. The compressive forces exerted by the shrink fit of the Inconel shell and copper layer onto the biological shield lower the thermal tensile stresses in the U-8 Mo shield.

The thermal stresses developed in the Inconel shell during the quench are below the yield strength of the material. The stress in the intermediate copper layer exceeded the copper yield strength in some regions. At the locations where copper yield stress is exceeded, copper will flow plastically and redistribute or equalize the contact forces at the copper-Inconel and copper-U-8 Mo interfaces. Therefore, plastic flow of the copper layer does not constitute a failure or jeopardize the integrity of the assembly. The thermal stresses in the biological shield which result from contraction of the outer layers are below the yield

NOTE: Heat Source Inside Generator During Fire

These data are initial conditions for thermal stress analyses

Figure VI-7. Temperature Distributions in Heat Source for Fire Followed by Water Quench

613208-10B

Figure VI-8. Heat Source Analytical Stress Model for FEATS Computer Code

TABLE VI-2
HEAT SOURCE MATERIALS PROPERTIES

<u>Material</u>	<u>Temp °F</u>	<u>Elastic Modulus lb/in² × 10⁻⁶</u>	<u>Poisson's Ratio</u>	<u>Thermal Expansion in/in °F × 10⁶</u>	<u>Yield Strength³ lb/in² × 10³</u>
U-8 Mo	70	12	0.2	6.5	133
	500	12	0.2	6.5	74
	1000	15.2	0.2	8.0	52.6
	1500	1	0.5	13.0	16.7
Inconel 718	70	29	0.294	8.96	135
	500	27.1	0.275	8.96	125
	1000	24.8	0.271	8.96	118
	1500	21.3	0.321	8.96	90
Copper	70	17	0.3	9.8	9.3
	500	17.3	0.3	9.8	8.6
	1000	5.6	0.3	12.5	5.6
	1500	1.1	0.3	13.3	1.1

TABLE VI-3
THERMAL STRESSES IN HEAT SOURCE
0.05 HOURS AFTER WATER QUENCH

<u>Element Number</u>	<u>Temp °F</u>	<u>Radial Stress (psi)</u>	<u>Axial Stress (psi)</u>	<u>Tangential Stress (psi)</u>	<u>Effective Stress (psi)</u>
1	97	2865	904	3071	2437
2	97	2705	-258	2797	3024
3	129	916	1533	5499	6387
4	129	1256	-24	932	1675
5	186	-2254	6263	20963	32435
6	217	10449	-980	11212	13856
7	212	47212	30070	66319	65427
8	297	20181	-9309	18182	30670
9	455	-4849	-17859	-4848	16008
10	778	-18308	-12436	-17766	8687
11	1141	-33916	-8356	-34891	27333
12	1298	-695	430	-221	5426
13	219	72766	23077	80396	78688
14	280	25394	-4335	27343	31873
15	768	-23799	-11961	-19509	11058
16	1239	-24074	-4413	-19433	20862
17	1300	-21006	-9594	-14293	17780
18	240	63478	10509	72898	72505
19	246	31859	898	36899	40085
20	701	-10409	-7428	-8491	4278
21	1213	-25506	-12068	-25991	15587
22	1278	-10270	-6562	-12139	13939
23	274	47409	21165	68772	71823
24	274	20413	5978	33050	43866
25	689	-5998	-8745	-4436	14916
26	1188	-23458	-16040	-28665	13718
27	1275	-6904	-6593	-10087	14960
28	308	15439	38049	62825	62853
29	308	-1183	21273	29296	52623
30	695	-14066	-1902	-1651	29397
31	1144	-20569	-22893	-32810	11259
32	1212	-12140	-18583	-21662	13435
33	322	-2531	5007	45531	44765
34	322	-8467	26665	55099	64846
35	322	-2826	31985	33256	38143
36	575	-3426	17929	20247	29500
37	829	-46943	-12986	-22579	31037
38	1149	-12458	-18245	-26453	13192

TABLE VI-3 (Con't.)

<u>Element Number</u>	<u>Temp °F</u>	<u>Radial Stress (psi)</u>	<u>Axial Stress (psi)</u>	<u>Tangential Stress (psi)</u>	<u>Effective Stress (psi)</u>
39	347	1041	101285	86251	93803
40	347	-7185	42744	37393	47546
41	667	-9996	19044	16321	28022
42	1087	-23647	-32567	-36624	12375
43	1122	-4356	-29848	-27518	24951
44	1292	-1754	629	-1577	2299
45	382	-454	109640	94533	103711
46	382	-1198	44415	40703	44178
47	638	-4375	24836	27831	31017
48	1043	-7794	-37770	-32182	27611
49	1307	730	4453	5649	4648
50	404	-2620	105379	92796	102380
51	404	-3519	41384	38238	43456
52	658	-3986	24874	28227	30994
53	1055	-4754	-35351	-26556	27421
54	1324	3897	6481	10647	5937
55	396	-1404	106702	93530	102681
56	396	-2132	43632	39029	43760
57	708	-4004	18785	18333	23020
58	1107	-5399	-25834	-22228	19104
59	1288	955	1448	2654	2053
60	346	8556	120612	96914	115214
61	346	-4006	48975	39988	57422
62	767	-10427	5382	-1409	18752
63	1185	750	-14822	-14093	18469
64	1253	-8489	-13629	-12263	6328
65	266	36705	36733	77096	83815
66	266	14991	22506	39774	46252
67	765	-10596	-8966	-13563	4172
68	1280	-2000	-11748	-7655	9054
69	195	64905	7023	81584	75280
70	195	36933	-5463	42780	50768
71	735	-12015	-8114	-16015	7259
72	1288	-7896	-13903	-11883	6315
73	134	111015	1269	111448	110066
74	134	61684	-615	61859	62396
75	728	-16853	-3257	-18061	14264
76	1337	651	-826	599	2035

TABLE VI-4
RESULTS OF HEAT SOURCE TRANSPORTATION ACCIDENT
IMPACT ANALYSES

Impact Attitude	Drop Height	Honeycomb				Maximum Stress (psi)		
		Nominal Thickness in.		Impact Deformation in.		Stainless Steel	Shield	Copper
		Inner	Outer	Inner	Outer			
End - On	30 ft	5	12	2.2	.5	3760	1730	690
Side	30 ft	.11		5		--	1156	536
Corner	30 ft	.22		11.6		1253	979	417
Side-Puncture by Mandrel	40 in	End-17		End-6.4		--	--	--
		Side-11		Side-9.7		--	--	--

strength of the U-8 Mo material. Based upon the above analysis, it may be concluded that the biological shield of the heat source assembly can be expected to survive the postulated water quench accident situation.

The most reassuring information that thermal shock does not present a safety problem can be found in the heat treatment process used by Union Carbide/Paducah for uranium/molybdenum alloys. In this process, uranium-8 w/o molybdenum derbies (18" dia x 6-1/2" thick) are routinely heated to a temperature of 1250°F and quenched in water without damage.

3. Response to Transportation Accidents

All postulated transportation accidents occur with the heat source enclosed in its shipping container. The container has been described in some detail in Section V-A of this report.

a) Impact. In transport of the SNAP-23A heat source, it is essential to avoid severe damage to the U-8 Mo biological shield. Severe damage is defined as any damage such that radiation streaming through cracks (fractures), structural joints, or shield closures results in radiation levels greater than 1000 mr/hr at a distance of one meter from the surface of the unit. The integrity of the heat source following impact, is enhanced by the shipping container. The design of the container depends upon the environments resulting from truck transportation, particularly impact.

The response of the heat source within its shipping container has been analyzed for impact at various attitudes following a 30 ft. drop to a hard surface. This is an accident environment which is considered equivalent to an impact accident during transportation by a carrier.

The design of the shipping container for the SNAP-23A unit incorporates features which reduce both impact and fire hazards associated with an accident environment. By insulating the SNAP-23A heat source unit within its container package, the temperature of the shield can be maintained at approximately 1100°F and the surface temperature of the container at 180°F. At 1100°F the U-8 Mo shield exhibits maximum ductility. The

insulating materials selected in the design also afford acoustical impedance, thereby attenuating the stresses at impact. The impact attitudes of the container package are end-on, side, and corner. The puncture resistance of the unit was also determined after free fall from a height of 40 inches onto a mandrel. The detailed results of these analyses are presented in Reference VI-6 and summarized in Table VI-4. The shipping container assembly utilizes layers of aluminum honeycomb for absorption of impact energy. Table VI-4 presents for each case the nominal honeycomb dimensions before impact and honeycomb deformation after impact. For the 30 foot drop cases, none of the magnitudes of deformation reach 70 percent of the original thickness which is the limit beyond which the change in crush force will rapidly increase above a constant and tolerable value. The analyses demonstrate that the SNAP-23A/FPMS heat source shipping container can survive a 30 foot drop in any orientation without affecting the integrity of the heat source.

b) Explosion and Shrapnel. The reference explosion defined in Appendix F is based on the environment resulting from a truck transportation accident. Such an environment is not considered a credible transport condition by the AEC and DOT, and need not be evaluated for a shipping license. However, it has been shown in section VI-B, paragraph 2C that the bare heat source will remain intact after being subjected to the reference overpressure environment. The outer container carbon steel shell will withstand an external explosive overpressure up to a critical value of 1090 psi. This computation does not consider the internal stiffening of the layers of honeycomb energy absorber. Since the incident overpressure from the blast is 73.5 psi, the outer container will probably undergo negligible deformation. Therefore, the heat source in its shipping container would experience no damage from the over-pressure environment. The response of the heat source within its shipping container, to shrapnel from the referenced explosion has also been estimated. The source of the shrapnel fragments is assumed to be the exploding tank of liquid propane referred to in Appendix F. Considering steel shrapnel fragments from the tank as being of the dimensions 2 in. x 2 in. and 1 ft. x 1 ft. square, the corresponding ballistic coefficient will be on the order of 10 psf. (for each). The velocity attained by the

shrapnel is determined by using the parametric curves in Reference VI-3 presenting shrapnel speed versus TNT equivalent yield, shrapnel initial distance from center of the blast and ballistic coefficient. The speed attained for the shrapnel is approximately 1100 fps for a TNT yield of 860 lbs (2%) and a tank radius of 5 ft. It is conservatively assumed that no blockage takes place as the result of other structures between the shrapnel source and the shipping container. Consequently, potential penetration by shrapnel at 1100 fps. of the 0.125 in. thick, steel, outer container of the shipping container can be analyzed using the results of Reference VI-7. Thompson and Hayduk presented the following expression for the critical thickness of a plate at which perforation takes place by a spherical projectile traveling at a corresponding critical velocity:

$$t_{cr} \propto \frac{m V_{cr}}{a^2 \sqrt{\rho_t}}$$

where:

- t_{cr} = critical sheet thickness
- m = projectile mass
- V_{cr} = projectile velocity causing penetration
- a = projectile radius
- ρ_t = sheet or target density

A constant of proportionality was determined from data considering hail perforating aluminum. Application of the equation for the case of the 2" x 2" tank fragment at 1100 fps. impacting the 0.125 in. thick outer container yields a critical thickness, $t_{cr} = 0.37$ in. Therefore, since a thickness equal to 0.37 in. is required, the shrapnel may penetrate the outer container shell. However, it is anticipated that the 11 in. thickness of aluminum honeycomb would prevent the shrapnel from reaching the inner container and the heat source. Other penetration correlations are available as will be discussed for the case of the SNAP-23A System.

c) Fire. The thermal environment associated with the hypothetical transportation

accident is the exposure of the package for a 30 minute period to a 1475°F fire which has an emissivity coefficient of 0.9, assuming that the surface of the package has an absorption coefficient of 0.8. It is further assumed that artificial cooling is not employed until internal temperatures have begun to fall.

Even a cursory survey of the design features of the SNAP-23A/FPMS shipping container is sufficient to recognize that the heat source (containment vessel) is well protected against thermal damage from a short-term fire. Serving as thermal barriers are the sand and insulation barriers as well as the air gaps that are expected to develop as the outer regions of the honeycomb and aluminum wool melt in the early stages of the fire.

Furthermore, the materials used for its component parts give the heat source an inherent resistance against fire damage where the effectiveness of the package in protecting its contents has no practical significance. Using a Westinghouse thermal transient code and the thermal property data presented in Appendix A, the response of the bare heat source to a 30 minute fire at 1475°F was evaluated. The view factor between the fire and the heat source was determined as follows:

$$F_{12} = \frac{1}{\frac{1}{\epsilon_1} + \frac{A_1}{A_2} \left(\frac{1}{\epsilon_2} - 1 \right)} = 0.8$$

where ϵ_1 = heat source surface emissivity assumed to be 0.8 for oxidized stainless steel and Inconel

ϵ_2 = fire emissivity (conservatively assumed to be 1.0)

The results of this analysis are summarized in Figure VI-9 which reveals that all heat source components remain well below their thermal/metallurgical limits at all times. It is, therefore, evident that containment of the radioactive material and effectiveness of the shielding are assured in the course of the fire, whether or not the heat source is protected by the shipping container.

Although the above analysis indicates that component temperatures for the bare

Figure VI-9. Heat Source Temperature Profile During and After a 1475°F Fire in Air

heat source fall rapidly once the fire has been extinguished, a realistic evaluation of the postfire phase cannot ignore the effect of the damaged container on long-term heat rejection.

Partial or complete meltdown of the aluminum components in the outer container during the fire would change the heat transfer mechanism from conduction to the less effective radiation and convection, resulting in higher equilibrium temperatures.

To evaluate the significance of this change in heat rejection capability, the inner container with its surrounding insulation was assumed to be resting on the base of the outer container; the outer container was also assumed to have remained essentially intact.

This analysis was primarily involved with determining the temperature drop from the outer container to the inner container. Heat transfer from the outer container to the ambient surroundings was based on the results from nominal steady-state conditions prior to the fire. Steady-state calculations indicated a peak temperature of 170°F on the outer container in a 100°F ambient based on heat transfer by natural convection plus radiation from the surface with an emissivity of 0.3. It should be noted that this emissivity is more conservative than the recommended value of 0.8 when heat rejection is involved. For this analysis, therefore, a uniform temperature of 170°F was assumed for the outer container.

The temperature drop through the insulation and sand to the heat source assembly was also based on the results of nominal steady-state conditions. In a 100°F ambient with the insulation adjusted to provide a minimum heat source temperature of 1100°F , the temperature drop from the insulation surface to the copper accumulator was approximately 850°F . Using this temperature drop to represent the shipping container thermal resistance after the fire, the remaining thermal resistance to evaluate is the temperature drop from the inner container to the outer container.

The outer container is unsealed and the space between the inner and outer containers is sufficiently large to expect that convective air currents will develop to provide a convective heat transfer path in addition to a radiative path. Since the area is enclosed, the convective heat transfer rate will be impeded and cannot be calculated by conventional natural convection correlations. The temperature drop was, therefore, calculated for

three modes of heat transfer to provide conservative, optimistic, and an intermediate estimate of the temperature drop between the containers. The conservative mode of heat transfer is by radiation only.

The second mode is by radiation plus convection, using a conventional equation for natural convection from a vertical cylinder represented by the equation.

$$h_{wi} = 0.19 (T_{wi} - T_{amb})^{1/3}$$

The net convection term is determined by

$$Q/A_{conv} = h_{ec} (T_{wi} - T_{wo})$$

where:

$$\frac{1}{h_{ec}} = \frac{1}{h_{wi}} + \frac{1}{h_{wo} \frac{A_{wo}}{A_{wi}}}$$

The subscripts i and o refer to the inner and outer containers, respectively. This mode of heat transfer is optimistic because the two cylinders will impede the convective currents. Based on a correlation presented in Reference VI-8 for enclosed spaces, the convection term is expected to be suppressed by approximately 50 percent. The third mode, therefore, is represented by radiation plus convection with the overall convective heat transfer coefficient reduced by 50 percent.

Since convection and radiation are in parallel, the total heat transfer rate is

$$Q/A_{tot} = Q/A_{rad} + 0.5 Q/A_{conv}$$

where

$$Q/A_{rad} = \sigma \epsilon (T_{wi}^4 - T_{wo}^4)$$

$$\epsilon = \frac{1}{\frac{1}{\epsilon_{wi}} + \frac{A_{wi}}{A_{wo}} \left(\frac{1}{\epsilon_{wo}} - 1 \right)}$$

Assumptions included in the analysis were:

- Surfaces of inner and outer containers are coated with aluminum

$$(\epsilon_{wi} = \epsilon_{wo} = 0.1)$$
- Heat transfer occurs from the top and side of the shipping container insulation only.
- For radiation the equation for the view factor is based on concentric cylinders.
- Natural convection correction is for heat transfer from a vertical cylinder.

Figure VI-10 presents the heat source accumulator temperature as a function of the heat flux at the inner container surface based on a 850°F temperature drop through the insulation and sand for the three heat transfer modes. The temperature of the copper accumulator was selected for display since it has the lowest melting point of all the heat source components. Based on an FPMS thermal power of 1100 watts, the heat flux at the surface of the inner container is estimated to be 230 Btu/hr-ft². The figure reveals that the copper accumulator reaches a maximum temperature of about 1500°F for the most pessimistic heat transfer condition (radiation only). For a more realistic assumption of radiation plus suppressed convection from the top and side of the insulation, the accumulator temperature is 1300°F. The temperature at which the copper and U-8 Mo forms a eutectic is 1740°F and represents the lowest point where heat source damage can occur. Heat rejection in the post-fire phase is, therefore, considered adequate to maintain heat source integrity. Hence, it is concluded that the reference 1475°F, 30 minute transportation fire will not affect containment of the radioactive material or effectiveness of the shielding.

In addition to satisfying the requirements specified in the AEC regulations for transport of radioactive materials (10 CFR 71), the SNAP-23A safety criteria requires that the heat source, within its shipping container, will survive a 30 minute 1850°F fire without loss of fuel containment. The high temperature capability of the heat source will be similarly effective in preventing extensive damage from this more severe thermal environment. This is illustrated in Figure VI-11 where it is evident that capsule integrity is maintained (Hastelloy-C melting point~2300°F) as required by the safety criteria. Furthermore,

Figure VI-10. Peak Heat Source Accumulator Temperature After Loss of Honeycomb

Figure VI-11. Unprotected Heat Source in 30 Minute 1850°F Fire (Run M3)

shielding effectiveness is also preserved since the temperature of the U-8 Mo shield remains well below its melting point (2190°F). Minor damage could result at the shield/accumulator interface where the peak temperature approaches the 1742°F copper/ uranium eutectic.

d) Oxidation of Biological Radiation Shield. The construction of the heat source and the protection provided by the shipping container make it very unlikely that shield oxidation could result from a transportation accident. Nevertheless, the oxidation characteristics of U-8 Mo have been investigated through a number of data sources. These references are noted on Figure VI-12 where the expected shield recession rate has been plotted as a function of shield temperature. Note that the data are all for the case where the oxygen has free and unlimited access to the U-8 Mo surface. This, in fact, will not be the case since exposure of the U-8 Mo to air is impeded by multiple barriers (outer jacket and accumulator) which would considerably retard the oxidation process. Extensive exposure of the shield surface to air implies an enhancement in heat rejection and a corresponding reduction in shield temperature to a level where oxidation rates are quite moderate.

The majority of the test data shown were derived under the assumption that the oxide film observed during the test was U_3O_8 . The amount of uranium reacted will be somewhat greater if the assumption had been made that the oxide was UO_2 . The two rates can be described as follows:

$$\begin{aligned} \text{Reaction rate of U for } UO_2 \text{ oxide} &\propto \left(\frac{238}{32}\right) \quad (\text{weight gain before spalling}) \\ \text{Reaction rate of U for } U_3O_8 &\propto \left(\frac{714}{128}\right) \quad (\text{weight gain before spalling}) \end{aligned}$$

Therefore, the uranium reacted for formation and spalling of UO_2 is 1.34 times the uranium reacted if U_3O_8 oxide formation is assumed. This possible error is not considered significant in comparison to the conservative assumption made here regarding chemical reaction kinetics and specifically unlimited oxygen availability.

The data fit shown on Figure VI-12 has been used in conjunction with a safety criterion for radiation to arrive at the time versus temperature estimate shown in Figure VI-13. It has been a long-standing SNAP-23A safety criterion that accidents

Figure VI-12. Biological Shield Recession Rate Due to Oxidation in Air

Figure VI-13. Time Required for Shield Oxidation to Exceed 1000 mr/hr Radiation Criterion

will not lead to a radiation level greater than 1000 mr/hr at a distance three feet from an exposed surface. This criterion has been converted to an appropriate shield thickness requirement from which a limiting total recession may be calculated.

During shipment of the heat source, the nominal shield temperature is about 1100°F. Under the conservative assumption that oxidation proceeds at this temperature, about 10 days must elapse before the radiation criterion will be exceeded. Even though this time is conservative by perhaps orders of magnitude, it does not represent a problem since shipment of the heat source is closely controlled.

e) Sequential Accidents. The previous sections have identified the response of the heat source/shipping container assembly to a number of transportation accident environments. It may be postulated that these environments occur in the sequence: impact, explosion, fire, water immersion, and oxidation. The free drop analysis (Appendix G) revealed that not only would fuel containment and shield effectiveness be maintained, but also relatively little damage would be incurred by the packaging. In the end-on orientation, neglecting the energy absorbed by the outer container, the inner 5 inch layer of honeycomb crushed 2.22 inches while the outer 12 inch layer crushed only 0.48 inch. For the case of side-on impact, again assuming that no energy absorption takes place in the outer container, a 5.0 inch deflation in the 11 inch thick honeycomb structure was predicted. The most damage to the packaging results from a corner impact which produces an 11.6 inch deflection of an available 22 inch thick honeycomb structure for this orientation. In none of these cases is the outer container envelope expected to fail. The localized damage that would result from the 30 foot drop will not affect the general capability of the packaging to survive puncture on a mandrel. As discussed in Appendix G an end-on impact would produce a maximum mandrel penetration into the honeycomb of 6.4 inches, neglecting the energy absorbed in shearing the cover. Impact on the side of the package is expected to produce a penetration into the honeycomb of approximately 9.7 inches.

Explosion will not materially change the post impact configuration.

The insulation surrounding the inner container and the sand inside are expected to

remain virtually undisturbed¹ as a result of the mechanical damage incurred by the 30 foot drop and the subsequent mandrel puncture. Hence, the resistance of the package to thermal damage from a fire will not be compromised by the above accidents. The inherent capability of the heat source to survive the 1475°F/30 minute fire without any thermal protection, as discussed in Section VI B-2 provides a high degree of confidence that the integrity of the heat source will be maintained, regardless of the condition of the packaging.

Immersion of the SNAP-23A/FPMS shipping package in water subsequent to the fire drop, puncture, and thermal accidents discussed above should result in a gradual cooldown of the heat source as the water penetrates the insulation, inner container shell, and the surrounding sand. The demonstrated resistance of the U-8 Mo shield material to thermal shock reveals that shielding effectiveness will not be affected even if damage to the packaging is sufficient to allow a rapid quench of the heat source.

Oxidation following all other accidents will depend upon shield temperature. Any damage to the container will have the effect of reducing the shield temperature and oxidation rate. In any event, the oxidation rate is not critical even at maximum transport steady state temperature of the heat source radiation shield.

C. SNAP-23A SYSTEM

1. Accident Network

The accident networks for the SNAP-23A system span the time period from fueling of the generator to permanent deployment at the site. The network results are also applicable to the post mission phase providing that all assumptions made here are consistent with post mission conditions. Two basic configurations are of importance with the SNAP-23A system. These are the system alone and the system in a shipping container. The shipping container can be further subdivided into two configurations, depending upon the transport mode involved.

The probability of an accident of any type or severity involving the SNAP-23A system is divided into two main categories as shown in Figure VI-14. The accident probability during transportation of the SNAP-23A system between WANL and an arbitrary site 3000 miles away is 1.12×10^{-2} . The sum of these two branches yields the probability

Figure VI-14. Accident Occurrence Probability for SNAP-23A System

that the SNAP-23A system undergoes an accident prior to deployment. The accident can range from minor to severe.

a) Handling, Storage, Test, and Siting Accidents. The handling, storage, test, and site accident network is presented in Figure VI-15. The probability that the SNAP-23A system is involved in a handling, storage, or test, accident is approximately 3.14×10^{-2} . This magnitude is assumed to be the same as that for the heat source determined in Section VI B-2.

The probabilities that the SNAP-23A system alone and the SNAP-23A system in its container are the configurations involved in the accident are approximately 8.4×10^{-3} and 2.2×10^{-2} respectively. These are the same magnitudes as used for the heat source in Table VI-1. The SNAP-23A system accident probability on site depends strongly upon the site selected and is undetermined at this time. Therefore, all probability numbers are exclusive of the site, as noted on Figures VI-14 and VI-15.

b) Transportation. The transportation accident network for the SNAP-23A system is presented in Figure VI-16. The configuration initially involved in any accident will be the system inside the appropriate shipping container; all usual transport modes are permissible. It has been arbitrarily assumed that transport is over a distance of 3000 miles. A combination of transport modes has not been evaluated, although this obviously can be done in any sequence desired. Certainly the transport phase will involve a combination of transport modes with possibly a final deployment by helicopter. The probability of an aircraft accident in a 3000 mile span is determined to be 1×10^{-4} with the aid of Appendix H. The probability of a ship accident over the same distance is not known, but has been qualitatively assumed to be less than that for the aircraft. This relation between ship and air transport is suggested in Reference VI-12. The truck accident probability for the SNAP-23A system is the same as that derived for the heat source, 3.5×10^{-6} accidents per mile. Multiplication of this value by 3000 miles yields the probability of 1.1×10^{-2} .

Multiplication of the accident/mile statistic by distances up to 3000 miles (and probably somewhat beyond) is equivalent to the probability resulting from the binomial

SNAP-23A System in Shipping Container 2.13×10^{-2}	SNAP-23A System Alone During Handling Storage, Test 8.4×10^{-3}	SNAP-23A System Alone and Operating on Site *
<ul style="list-style-type: none"> ● Drop & Impact ● Impact From Debris or Equipment ● Fire ● Explosion 	<ul style="list-style-type: none"> ● Drop causing Impact ● Impact From Debris or Equipment ● Fire ● Explosion 	<ul style="list-style-type: none"> ● Fire ● Flood ● Vandalism ● Sabotage ● Earth Burial ● Deep or Shallow Water Burial ● Impact

* Site Probability Depends Upon Site Selection and Cannot Be Estimated at This Time.

Figure VI-15. SNAP-23A System Handling, Storage, Test, and Site Accident Network

Figure VI-16. SNAP-23A Generator Transportation Accident Network

distribution. Assuming the binomial distribution applies to this discrete process, the probability of an accident occurring in 3000 miles is determined as follows:

$$nPr = \frac{n!}{r!(n-r)!} (p)^r (q)^{n-r}$$

where for no occurrences in 3000 tries:

$$r = 0$$

$$n = 3000$$

$$p = 3.5 \times 10^{-6} \text{ accid. / mile}$$

$$q = 1-p$$

$$\text{then } {}_{3000}P_0 = \frac{3000!}{0!(3000)!} (3.5 \times 10^{-6})^0 (.9999965)^{3000}$$

$${}_{3000}P_0 = (.9999965)^{3000}$$

$$\ln ({}_{3000}P_0) = 3000 \ln (.9999965) \approx 3000 (.9999965-1)$$

$$\approx -.0105$$

$$\text{now } {}_{3000}P_0 \approx e^{-.0105} \approx (1-.0105) \approx .9895.$$

finally, the probability of one accident in 3000 tries is:

$${}_{3000}P_1 = 1 - {}_{3000}P_0$$

$$= 1 - .9895$$

$$= .0105 \approx 3000 \times 3.5 \times 10^{-6}$$

In Appendix H, the data are shown for subdividing the aircraft accident into three types: takeoff or landing, mid-air collision, and other freight accidents. These are shown on Figure VI-16 for a 3000 mile shipment. Division of the truck accidents into three specific categories is also shown on Figure VI-16. The probabilities use the same data described in Section VI-B Paragraph 1.

2. Handling, Storage, and Test Accidents

a) Impact. Two configurations are involved for this phase of SNAP-23A operation:

The bare system and the system in its shipping container.

The bare SNAP-23A system is considered in Case 1 (See Table G-1 of Appendix G) for impact against a rigid medium. The impact impedances of the base assembly are neglected in the analysis. It has been estimated that the biological shield will remain intact at an impact velocity of 180 fps, which is equivalent to a free fall from 500 feet. The capsule will remain intact for considerably higher system impact velocities. This accident environment is much more severe than any condition that could be encountered during handling, storage, and test.

The shipping container will significantly enhance the impact resistance of the system when it is being handled as a shipping package.

b) Crushing and Debris. An accident which results in debris falling on the SNAP-23A system should not affect the integrity of the heat source. However, mechanical damage to the fins could reduce the heat rejection capability of the system. A thermal analysis was performed in which it was assumed that the effectiveness of the fins was reduced by 40 percent, which represents rather extensive damage to this component. The results of this analysis are summarized in Figure VI-17 which reveals that steady state temperatures for this damaged system are well below the damage level of any heat source component.

c) Fire. The SNAP-23A system has been evaluated for its response to a 1475°F one-half hour fire. This fire is considered to be an appropriate design safety environment for handling, storage, and test accidents. Since this time/temperature profile is the design basis for transportation accidents, the detailed analysis of system response to this environment is discussed below in the evaluation of transportation accidents (Section VI-C Paragraph 3). It is concluded therein that such an environment will not compromise heat source integrity.

d) Open Circuit. Another potential accident that might occur during this phase is a failure in the power conversion system such that an open circuit is created. Such an event would increase the thermal impedance through the converter from loss of Peltier cooling and cause heat source temperatures to rise above the operating point. The steady

Figure VI-17. SNAP-23A System Temperatures ($^{\circ}\text{F}$), Steady State, 40% Loss of Fins. BOL, Open Circuit

state temperatures in the heat source following such a failure are shown in Figure VI-18 for beginning-of-life (BOL) and end-of-life (EOL) conditions. It is again evident that heat source temperature limits (see Table IV-3) are not exceeded.

e) Converter Assembly. As the insulation is added on top of the heat source in the course of final system assembly, heat rejection to the environment will be reduced and the temperature of the heat source assembly will increase. When the thermal insulation has been completely installed around the heat source assembly, prior to mounting the converter/radiator assembly on the stud, the steady-state shield temperature will have a nominal value of 1700°F , while the corresponding temperature of the fuel capsule will be approximately 1825°F . As shown in Table IV-3, the shield - accumulator interface imposes the greatest constraint on heat source temperature and must be maintained below 1742°F to avoid the formation of a eutectic which would result in localized melting. Since the anticipated margin of safety on temperature is relatively small, procedural controls will be employed to avoid the possibility of heat source damage. This approach is considered quite feasible, since the time to reach steady state from the initial configuration (exposed top surface) will require an elapsed time of at least 12 hours. Pre-fitting and pre-assembly of generator components are expected to reduce the assembly time to a period of a few hours, at which time the shield temperature is expected to approach 1100°F .

3. Transportation Accidents

a) Impact. The response of the SNAP-23A system to impact resulting from transportation accidents is presented in this section. Transportation modes considered are aircraft, truck, and ship. The most severe impacts on the system result from the aircraft accident cases. Detailed computations and derivations are presented in Appendix G, Impact Analyses.

In order to determine the response of the SNAP-23A fuel capsule for a given impact case, it is necessary to determine the resistance or impedance of each of the components involved. Impact impedances, ρC , or the product of density and speed of sound through the medium, are computed for the Hastelloy-C capsule, U-8 Mo shield, Inconel 718 stainless-

Figure VI-18. SNAP-23A System Temperatures (°F), Steady State, Open Circuit

steel shell, and shipping container. An impact impedance was also derived for the aircraft fuselage. In a specific impact case, several of these comprise the receiver medium onto which the capsule impacts. The composite impedance, $\rho_e C_e$, of this receiver medium is:

$$\frac{1}{\rho_e C_e} = \sum_{i=1}^n \frac{1}{\rho_i C_i} \quad (\text{for } n \text{ receiver components})$$

This expression is analogous to the composite spring constant for a vibrating system composed of n springs in series. The capsule response during impact depends upon the composite impedance of the receiver medium. Low magnitude provides protection through high energy absorption. However, if the component impedances are too low or soft, failure stress is also low. Thus there is an optimum impedance, where high strength and good impact energy absorption are both achieved. An example is the aircraft fuselage.

During impact, the stress at the interface between the capsule and the receiver medium is:

$$\sigma = \rho_i C_i V_f \left[1 - \exp. \left(\frac{-2 \rho_c C_c}{\rho_i C_i} \right) \right] \quad \text{psi}$$

where:

$$\begin{aligned} \rho_i C_i &= \text{capsule impedance, lb sec/in}^3 \text{ at its normal operating temperature} \\ \rho_c C_c &= \text{receiver medium impedance, lb sec/in}^3 \\ V_f &= \text{Impact velocity, ips} \end{aligned}$$

The stress and the capsule mechanical properties at elevated temperatures are used to determine the parameters of an impact failure criterion developed for thin and thick walled cylindrical shells. The responses of the fuel capsule and the U-8 Mo shield were determined by employing this semi-empirical criterion. The results are presented in Table VI-5 and discussed in the following paragraphs. The cases are considered in an order of increasing relative impact velocity.

TABLE VI-5
IMPACT CASES INVOLVING THE SNAP-23A SYSTEM

Configuration	Accident Cases	Relative Impact Velocity (FPS)	Capsule	Shield	Impedances Comprising Receiver Medium (lb sec /in ³)			
					U-Mo Shield	Steel Shell	Container	Aircraft
SNAP-23A System	1-Truck head-on collision - Velocity of Each is 10 MPH	180	Pass	Pass	140	133 133		
SNAP-23A System & container & aircraft	2-Aircraft crash during takeoff or landing	220	Pass	Pass	140	133	40*	.085
SNAP-23A System & cubic container	3-Typical Terminal velocity impact after midair release	237	Pass		140	133	40*	
SNAP-23A System & container	4-Limiting Terminal velocity impact after midair release	≤ 550	Pass		140	133	40*	
SNAP-23A System alone	5-Terminal velocity impact after midair release	591	Fail		140	133		
SNAP-23A System & container & aircraft	6-Aircraft crash en route	843	Pass		140	133	40*	.085
SNAP-23A System & container & 2 aircraft	7-Head-on midair collision between aircraft	1690	Pass		140	133	40*	.085 (2)

* - estimated upper bound

Case 1 - The first accident case considers the SNAP-23A only and corresponds approximately to a head-on collision of two trucks, each travelling at 60 mph. The analysis is conservative in that the SNAP-23A system is considered without its shipping container, and the impact impedances for the trucks are not included. The shield is estimated to remain intact at an impact velocity of 180 fps assuming it hits a steel shell receiver medium (see Appendix G). The fuel capsule will survive the 180 ft /sec velocity, (Estimates indicate it will survive at least 268 ft/sec for this configuration with the U-8 Mo shield as the receiver medium.)

The system should survive at impact velocities beyond those achieved in any postulated credible truck accident.

Case 2 - This case considers the SNAP-23A system and its shipping container in an aircraft that crashes head-on into granite during takeoff or landing. The aircraft accident during takeoff or landing has the highest occurrence probability (see the accident network, Figure VI-16, and Appendix H). The relative impact velocity of 220 fps corresponds to the maximum landing approach speed of 130 knots for a Boeing 707. The composite component impedance considered in Table VI-5 was determined to be $\rho_c C_c = 0.0847 \text{ lb sec/in}^3$. The low impedance of the aircraft fuselage (derived in Appendix G), $0.085 \text{ lb sec/in}^3$, completely dominates the composite magnitude. Thus, the high energy absorption capability of the aircraft fuselage results in low stresses on the capsule and shield due to impact.

Case 3 - The SNAP-23A system is considered to be in a high drag shipping container of the concept described in detail in Section V-B. The assembly impacts at terminal velocity into granite.

The container configuration used for this analysis consists of an aluminum I-beam and channel structural elements. A five-foot cubic container weighing 1620 lbs yields a terminal velocity impact of 237 fps, which is assumed for this case.

The composite impedance of the non-rigid receiver medium has been determined for the shield, steel shell and container components. The estimated upper bound of impedance for the container was synthesized to be 40 lb sec/in³ (see Appendix G). This impact case results in survival of the capsule. It demonstrates that a typical high drag container concept can be evolved that protects the capsule during terminal velocity impact. The preliminary container design described in Section V-B should have an impedance which is considerably better than the 40 lb sec/in³ used in the impact analyses. Consequently, it is expected that the incorporation of real container properties will result in impact resistance better than preliminary calculations suggest.

Case 4 - This case determines the limit terminal velocity impact which the capsule will survive. The SNAP-23A system is assumed to be in a container whose impedance is 40 lb sec/in³. The threshold stress value which divides the pass and fail regions of the impact criterion was employed to back calculate a limit impact velocity of 550 fps. (The composite impedance is the same as in Case 3, $\rho_c C_c = 25 \text{ lb sec/in}^3$.) Therefore, in an accident resulting in midair release of a container, the capsule will survive an impact terminal velocity of 550 fps.

Case 5 - This case assumes a mid-air release of the SNAP-23A System (no container) with a subsequent impact at terminal velocity onto granite. The terminal velocity of the 1220 lb system is determined to be 590 fps as follows:

$$V_t = \sqrt{\frac{2W}{C_D \rho A}} = \sqrt{\frac{2 \times 1220}{0.65 \times 0.002378 \times 4.5}} = 590 \text{ fps}$$

where:

$$\begin{aligned}
 W &= \text{SNAP-23A System weight} = 1220 \text{ lbs} \\
 C_D &= \text{Subsonic drag coefficient for a cylinder in an end-on attitude} \approx 0.65 \\
 A &= \text{Aerodynamic reference area of generator} \\
 &= \text{Length} \times \text{Diameter} = \frac{27'' \times 24''}{144} = 4.5 \text{ ft}^2 \\
 \rho &= \text{Atmospheric density at sea level} \\
 &= 0.002378 \text{ lb/sec}^2/\text{ft}^4
 \end{aligned}$$

Due to the aerodynamic asymmetry of the SNAP-23A system, induced by the protective cover holes, the most likely attitude at subsonic impact is assumed to be end-on.

The composite impedance of the receiver medium consists of the shield and steel shell and is computed to be 68 lb sec/in^3 . The impact criterion employed in Cases 3 and 4 is required for two reasons as follows:

- a) Impact at a lower terminal velocity
- b) Energy absorption capability if its impedance is low, $\rho_c C_{c \leq} 40 \text{ lb sec/in}^3$

Case 6 - This considers the SNAP -23A system in its shipping container in an aircraft that crashes en route. The probability of occurrence for this accident is less than that for the takeoff and landing accident category. A severe case is assumed whereby the aircraft crashes head-on into a granite mountain at cruising speed. The cruising velocity of a Boeing 707, 575 mph or 843 fps, is taken as the impact velocity.

As in Case 2 the aircraft impedance dominates the composite impedance of the receiver medium which was determined to be $0.0847 \text{ lb sec/in}^3$. The impact stress at the capsule-shield interface is only 1690 psi as a result of the high energy absorption of the aircraft fuselage. Consequently, the impact criterion indicates survival of the capsule for this case.

Case 7 - An aircraft accident postulated in the transportation accident network is a head-on mid-air collision with another airplane. The SNAP-23A system with its container is assumed to be in the airplane. The impact velocity is assumed to be the

relative velocity, 1690 fps, between two 707's at their cruising speeds. The composite impedance of the receiver medium, $\rho_c C_c = 0.0424 \text{ lb sec/in}^3$, is completely dominated by the presence of two aircraft fuselage impedences. It consists of the non-rigid shield, steel shell, and aircraft fuselage impacting another fuselage. The fuselage absorbs all of the energy of impact, yielding a negligible impact stress on the capsule of 1725 psi. Therefore, the impact criterion indicates that the capsule safely survives the mid-air collision impact.

It was also determined that the stress at the shipping container - fuselage interface was approximately 1720 psi. This loading against a side panel of the fuselage would probably result in the panel tearing out and subsequent mid-air release. The 1720 psi is well below the yield strength of 6061 aluminum at $\leq 200^\circ\text{F}$ (a maximum temperature at which the container would be surrounding the generator under normal operation). Therefore, we revert back to Case 3 or 4 whereby the intact generator in its undamaged container could undergo free fall to impact. The calculation indicates that the part of the fuselage structure to which the container is bolted may be part of the configuration at terminal velocity impact.

If one considers the mid-air collision whereby the aircraft is hit in the side by another aircraft, the case would be similar to Case 6 except the axial impedance of a complete fuselage would be neglected. In most side impact cases, the accident would probably result in impact at the terminal velocity of a configuration consisting of the SNAP-23A system, shipping container, and the attached fuselage structure.

Impact accidents due to transport via an ocean going ship have been given only cursory attention since the postulated accident is no more severe than a 30-foot drop (Section VI C Paragraph, 3). The impact response analyses made for aircraft and truck transport are adequate to cover ship transport. (The aircraft accident may become more applicable as ship velocities increase although the large masses involved will predominate the response evaluations).

More modest impact loads on the system during a transport accident could cause damage to the aluminum radiator and reduce heat rejection to the environment. However, damage could be more severe than the 40-percent loss assumed for a handling, storage, or

test accident. Accordingly, a thermal analysis was performed to assess system component temperatures for a transport accident of such magnitude that the entire radiator was destroyed while the insulation capability of the base assembly remained intact. The results of this analysis are summarized in Figure VI-19 which indicates that some melting will occur at the shield/accumulator interface for steady state conditions due to the 1742°F eutectic of the copper/uranium system. Fuel containment will not be affected and only minor degradation in shielding effectiveness would result. It should be borne in mind that transport of a SNAP-23A system will be accomplished under controlled conditions such that corrective action would be taken before the resulting thermal transient could cause damage.

b) Fire. The fire environments to which the system could be exposed during transport are defined in Appendix B. Briefly, these are a 1475°F 30-minute hydrocarbon fire which represents a nominal condition specified by AEC/DOT for transport licensing purposes and an 1850°F 30-minute fire which is an extreme condition imposed by the SNAP-23A Safety Design Criteria.

The inherent capability of the SNAP-23A heat source to survive direct exposure to high temperature fires was discussed in Section VI-B Paragraph 3. A major factor in this survivability was the favorable post-fire condition for heat rejection to the environment.

In the case of the SNAP-23A system, however, heat rejection in the post-fire phase depends primarily on the thermal path through the converter/radiator assembly. In fact, the thermal design of the system is based on 80 percent of the radioisotope heat passing through this assembly. Hence, the ability of the system to survive a fire without compromising safety depends on the extent to which thermal damage to the converter/radiator assembly has affected its thermal impedance.

The influence of the radiator in controlling system temperatures was revealed in earlier discussions where 40-percent and 100 percent reductions in fin effectiveness were evaluated. A detailed review of the converter assembly was also conducted to determine its vulnerability to thermal damage and associated effect on thermal impedance.

The basic heat path through the converter is illustrated in Figure VI-20. Approximate

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Figure VI-19. SNAP-23A System Temperatures (°F)
Steady State, No Fins, BOL Open Circuit

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Figure VI-20. SNAP-23A Converter T/E Element Heat Path Schematic

imately 1048 watts of heat (BOL) are transferred radially from the heat source stud to the converter's stainless-steel hot frame. Heat from the hot frame is conducted through the boron nitride electrical insulators to the copper hot junction electrodes and iron buttons of each T/E element. A small amount of heat is also transferred from the hot frame into the insulation and outer support structure surrounding the T/E elements and separating the hot frame from the aluminum cold frame. Approximately 960 watts of heat are transferred through the T/E elements where a large temperature drop ($\sim 817^{\circ}\text{F}$ at BOL) occurs. All of the heat transferred through the elements and the Min-K insulation, less the amount converted to electricity, is transferred through the aluminum cold frame and rejected by the radiator fins.

An examination of the heat path through the converter and the melting temperatures of converter components (Table VI-6) indicated that a change in the heat path and a corresponding effect on heat transfer was possible in a fire environment. For example, damage to the low melting point aluminum cold frame and the P-leg cold segment might alter the normal heat path through the thermoelectric elements. However, the complexity of the converter structure severely hampered an attempt to develop an analytical model that would permit a realistic assessment of thermal damage from a fire and its effect on thermal impedance.

Consequently, this information was obtained experimentally by means of simulated fire tests with thermal/mechanical mockups of the system converter. The results of these tests, which are reported in Section VIII-F, indicate that the gross thermal conductance of the converter increases slightly following the 1475°F fire while an appreciable increase in conductance results from the 1850°F fire. From a safety standpoint, these results are favorable since heat rejection is actually enhanced.

These test results were used to upgrade prior predictions on the transient behavior of the system to a fire, which had been based on a constant converter conductance. The differences between these predictions were found to be minor. The temperature response of key system components during the fire and in the immediate post-fire phase is shown in

TABLE VI-6
SNAP-23A T/E CONVERTER COMPONENTS
AND MELTING TEMPERATURES

<u>Component</u>	<u>Melting Point, °F</u>
Hot Frame - Stainless Steel 304	2550-2650
Electric Insulator - Boron Nitride	3000
Hot Junction Electrode - Copper	1980
Button - Iron	2100
P-leg Hot Segment - PbSnTe	1575
P-leg Cold Segment - BiSbTe	1200
N-leg Hot Segment - PbTe	1691
N-leg Cold Segment - PbTe	1691
Follower - BeO	4000
Cold Frame - Aluminum 2024	935-1180
Cold Frame Outer Case - Inconel 625	2350-2460
Upper Cover Assembly - Stainless Steel 304	2550-2650
Lower Cover Assembly - Stainless Steel 304	2550-2650

Figure VI-21 and VI-22 for the 1475°F fire and the 1850°F fire respectively. Radiator effectiveness after a 1475°F fire was a major concern in determining the post-fire steady-state temperature levels. The results of the fire analysis indicated that over 80 percent of the fin area remains unmelted after the one-half hour fire. Assuming no other fin damage, this prior fin loss would result in negligible change to the thermal performance (that is, less than 15°F rise of fin root temperature). This, however, would be an optimistic assumption. In all probability, significant fin degradation can be assumed over and above this actual fin melting. Such items as fin warpage and fire residue build-up along the fin surface causing air flow blockage are potential areas for degradation. The actual reduction in fin effectiveness from the above effects is unknown. Therefore, for the safety evaluation, the worst case was used. Convection from the fin surface area was assumed to be zero. This assumption reduces the radiator thermal model to a solid cylinder having a diameter equal to the stud diameter plus twice the remaining fin height. Since analyses of the 1850°F fire indicates that the fins will be completely destroyed, the post fire results for this accident assume no radiator is present. Only radiation and natural convection in an ambient environment were assumed for the cooling mechanism in the post-fire phase. It is evident that heat source components are virtually unaffected. The most significant change revealed in these figures is the increase in converter cold frame temperature over the pre-fire levels, which is the result of a loss of heat rejection capability from the outer case of the converter due to the damage and loss of fins of the radiator. Fin emissivity has a strong effect on the extent of melting during the fire, as shown in Table VI-6, where aluminum fin emissivities of 0.5 and 0.8 are compared. The radiator is anodized aluminum and the emissivity of the fin surface will increase as a result of fire exposure, so the value of 0.8 assumed in these analyses are considered realistic and probably close to what can be expected for the fire duration.

Steady state temperatures of heat source components following exposure of the system to a fire accident have considerable impact on the urgency and nature of the corrective action that must follow such an accident. As shown in Figure VI-23 and Table VI-7 the

Figure VI-21. Response of SNAP-23A System Components During and Following a 1475°F Fire

TABLE VI-6
SYSTEM TEMPERATURES RESULTING FROM
RADIATOR FIN EMISSIVITIES OF 0.5 AND 0.8

<u>Emissivity</u>	<u>Fire Temp (°F)</u>	<u>Time During Fire (hrs)</u>	<u>Temperature * (°F)</u>	
			<u>Fin</u>	<u>Cold Junction</u>
0.5	1475	0.2	605	575
0.5	1475	0.4	990	940
0.8	1475	0.2	725	675
0.8	1475	0.4	1125	1090
0.5	1850	0.2	960	890
0.5	1850	0.4	Melted	1180
0.8	1850	0.2	1100	1000
0.8	1850	0.4	Melted	1500

*Analytical thermal model data

Figure VI-22. Response of SNAP-23A System Components During and Following an 1850° Fire

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Figure VI-23. SNAP-23A System Steady Temperatures Following a 1475° Fire

TABLE VI-7
COMPARISON OF POST-FIRE STEADY STATE
SYSTEM TEMPERATURES WITH SAFETY LIMITS

<u>Component</u>	<u>Critical Temperature ~°F</u>	<u>Calculated Temperature</u>	
		<u>Post 1475°F Fire</u>	<u>Post 1850°F Fire</u>
Fuel	3470 melt	2280	2140
Fuel/Fuel Capsule I/F	< 2000 desirable	1870	1720
Hastelloy-C Fuel Capsule	< 2310 melt	1860	1710
U-8 Mo Biological Shield	2100 melt	1790	1630
Shield/Copper Accumulator I/F	1742 eutectic melt formation	1730	1580
Heat Source Outer Shell	< 2000 for minimal strength	1730	1570

long-term effects of a 1475°F have not compromised either fuel containment or shielding effectiveness. Minor damage is possible at the shield/accumulator interface since the temperature at this point is close to the 1742°F eutectic. The temperature of the converter hot frame (~1400°F) is sufficiently low that no significant change in converter conductance is anticipated from long term heating effects. By virtue of the detailed manner in which the converter components respond to a fire environment, system temperatures following an 1850°F fire shown in Figure VI-24 are substantially lower than the less severe 1475°F fire, despite the fact that the radiator is completely destroyed. The analyses and tests described above indicate that the SNAP-23A system has an inherent capability to survive the postulated fire accidents.

c) Explosion and Shrapnel. Insofar as SNAP-23A system concerned, the damage from an explosion can fall into two parts:

- (1) Damage to the radiator from direct overpressure impingement
- (2) Damage to the radiator, converter, and shielding due to shrapnel generated from an explosion

The magnitude of shrapnel damage which might occur can be estimated by approximate methods which have been developed in other SNAP Programs. The worst environment to which the SNAP-23A may be subjected is considered to be a low yield explosion with the center of the explosion occurring at least 30 feet away from the generator.

Considering the two percent yield propane explosion described in Appendix F, it is possible to make some rough estimates of radiator response to the resulting overpressure. The dynamic response of the radiator fins can be evaluated if the net pressure versus time profile for each individual fin can be determined. In Appendix F, the deflection of a fin is estimated based on its static strength characteristics, its natural frequency, and the blast pressure versus time environment.

The SNAP-23A radiator fins are approximately 2.6° apart and extend around the periphery of the hub. The forward fins, with respect to the shock front, will experience a minimum pressure differential across them due to nearly equalized pressures in the areas

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Figure VI-24. SNAP-23A System Steady State Temperatures
Following an 1850°F Fire

between them. The side fin could be a critical case and lends itself readily to an approximate analysis since its pressure load can be estimated. The side fin sees an overpressure pulse, exclusive of any reflected overpressure contribution consisting of two parts:

(a) Side overpressure magnitude for the time span equal to the fin thickness divided by the shock front velocity.

(b) Drag pressure over the time span until pressure relief is obtained.

These two overpressures are roughly equal in magnitude. The overpressure time is extremely short compared to the natural period of the fin. Due to this short loading time and the close proximity of neighboring fins, the side fin would not experience the overpressure load uniformly over its entire leading surface area.

The dynamic response curve, applied to the case of the side fin, indicates that the yield strain of the fin is not exceeded, and no permanent deformation takes place. While the analysis made is approximate, it indicates that the SNAP-23A overpressure accident environment will not lead to failure of the SNAP-23A system through the loss of heat rejection capability.

The response of the SNAP-23A system within its shipping container to the shrapnel environment can be estimated, based on the analysis performed in Section VI-B Paragraph C. The container employed for transportation by truck and by ship would consist of faced aluminum honeycomb components. In the explosion of a tank truck, shrapnel fragments are estimated to attain speeds of 1100 fps. Small fragments at these speeds will penetrate the outer sheeting if it is less than approximately $3/8$ of an inch thick. This assumes that the SNAP-23A shipping container is directly exposed to the blast with no intermediate structures present.

d) Earth Burial. As is shown later, the extent of earth burial is directly proportional to the impact velocity. Since the potential and magnitude for earth burial increases with impact velocity, it becomes apparent from the previous results on impact analyses that this problem is of primary concern for air transport accidents. Examination of the various air transport accident cases indicates that the worst situation is a complete ejection of a

bare system from the airframe following a mid-air collision, leading to ground impact of the system at approximately 590 feet per second. However, the impact analyses (Appendix G) indicate that a properly designed shipping container is necessary to insure fuel capsule integrity under these terminal velocity conditions. Thus it is mandatory that the system first be in an approved shipping container before it can be placed on board the aircraft and, second, be firmly restrained within the aircraft in accordance with an accepted procedure.

To have terminal velocity impact of the bare system following a mid-air collision would thus require that the shipping container be entirely stripped from the system either during the impact or shortly thereafter from aerodynamic forces acting on a structurally weakened container. It is believed that this is not a credible situation because:

(1) The probability (0.6×10^{-5} per Figure VI-17) of a mid-air collision is low.

(2) The air frame itself forms a crushable, energy-absorbing structure whose properties can be taken to advantage in reducing collision impact loads to the shipping container by specifying container placement and retention within the air frame.

(3) The low probability of the impact occurring in a preferred attitude leading to worst loadings on the shipping container.

Hence, burial of a bare system as the result of mid-air collision considered to be academic although it serves a useful purpose as an upper theoretical limit in determining earth burial depths.

Several analytical techniques have been developed to predict the depth of earth penetration of a projectile. The depth of penetration is a function of several parameters including nose performance coefficient and soil properties. Reference VI-15 summarizes the development of empirical equations which predict penetration. In Reference VI-16, attitude and terminal velocity data were used to modify the equations developed in Reference VI-15. The equation arrived at is:

$$D = KSN \left(\frac{W}{A} \right) V$$

where:

$$\begin{aligned}
 K &= 0.000354 \text{ in/lb}^{1/2} - \text{constant from test results} \\
 S &= \text{Soil constant} = 15 \text{ in sec/lb}^{1/2} \text{ for soft soil} \\
 &= 7.5 \text{ for hard soil} \\
 N &= 0.56 \text{ for flat end} \\
 &= 1.386 \text{ for side on cylinder} \\
 W &= \text{weight in pounds} \\
 A &= \text{frontal area in}^2
 \end{aligned}$$

The above equation has been used to predict maximum burial depth of the SNAP-23A system in the shipping container upon impact at terminal velocity. The shipping container configuration is considered to be a cube having seven-foot wide faces and having the design characteristics as given in Section V-B. For impact directly on one face in soft soil the following values for the various parameters in the penetration equation apply:

$$\begin{aligned}
 W &= 3500 \text{ lb} \\
 A &= (7 \times 7) (144) \text{ in}^2 \\
 V &= 237 \text{ ft/sec (terminal velocity of system in shipping container)} \\
 S &= 15 \\
 N &= 0.56
 \end{aligned}$$

The results show that the maximum penetration of the soil is:

$$D_{\text{max}} = 0.561 \text{ ft}$$

It is interesting to note that this depth of penetration precludes burial of the system, which is located no closer than 26 inches from the nearest face of the shipping container. Even with the crushing of the honeycomb structure to reduce impact loads to the system, earth burial should not occur. Also, the above earth penetration depth is conservative, in the sense that no allowance was made for energy absorption by the shipping container. Although energy absorption in the shipping container would produce crushing of the honeycomb structure, thereby putting the heat source closer to the ground surface, the depth of earth penetration would be less than that noted above.

Now if the bare system is considered, with impact occurring in the worst configuration, viz side-on, the following values for the variables in the penetration equation apply:

$$\begin{aligned} W &= 1288 \text{ lb} \\ A &= 625 \text{ in}^2 \\ V &= 590 \text{ ft/sec} \\ S &= 15 \\ N &= 1.4 \end{aligned}$$

It is found that the penetration depth is:

$$D_{\max} = 8.7 \text{ ft}$$

Obviously, the addition of the shipping container has a profound effect on the magnitude of a potential earth burial depth following a terminal velocity drop. It appears that a shipping container designed to reduce the impact loads to the heat source/fuel capsule under terminal velocity conditions will also provide sufficient means to prevent significant earth penetration. Note that the protective functions of the system shipping container include maintaining the required heat rejection paths within the system and container. Because the system shipping container configuration cannot be buried and post-impact heat rejection is possible, a safety problem is not evident.

e) Water Burial. Water burial can be postulated as the result of a transportation accident over water or a siting accident in the case of a marine application. The effect of water burial is to impose a hydrostatic pressure load on the heat source and to subject the assembly and its components to salt water corrosion.

As previously noted, the bare fuel capsule has been designed to survive burial in ocean depths up to 20,000 feet. Detailed stress analyses of the fuel capsule are reported in Appendix C where it is shown that the critical capsule welds will meet this safety criterion. To verify the capsule design, a series of hydrostatic pressure tests were performed; the results of these tests are reported in Section VII-B and demonstrate that the capsule will meet the safety criterion. In fact the test results indicate the capsule will survive hydrostatic pressure even at the deepest known ocean depth.

Corrosion must be minimal if the hydrostatic pressure test results are to be applicable for a long time period. Galvanic corrosion of the Hastelloy-C fuel capsule is the severe environment which has been examined in detail through corrosion testing. The corrosion tests were made for the specific material arrangement which appears in the SNAP-23A heat source and are reported in Section VII-C. Two basic assumptions were made for the test:

(1) The surface area in contact with seawater for each material equals its geometric surface area

(2) All materials are in electrical and electrolytic contact with each other .

The effect of accumulation of corrosion products on the surface of the materials was not accounted for, and in most cases, tends to retard the rate of corrosion. The test results indicate that the fuel will remain contained for approximately 100,000 years, equivalent to 3470 half lives of strontium-90.

f) Oxidation of Biological Shield. Oxidation of the U-8 Mo shield can be postulated to occur as the result of an impact accident. The extent of shield oxidation depends upon the temperature of the shield and can be approximated by using the data and approach presented in Section VI-B3. At a maximum and constant shield surface temperature of 1300°F, about 50 hours must elapse before the radiation criterion of 1000 mrem/hr at three feet is exceeded. This time estimate is extremely conservative because real oxidation rate will probably be a couple of orders of magnitude less than that of the data of Figure VI-13.

g) Sequential Accidents. The previous sections on the SNAP-23A System have identified the response of the system to a number of transportation accident environments. Preliminary estimates of the shipping container effects on system response have also been made. It may be postulated that the accident environments occur in the sequence impact, explosion, and fire, due to an airplane accident on takeoff or landing.

The impact analysis (Appendix G) has indicated that the system in the shipping container will remain intact following the impact at 237 ft/sec. Both fuel containment and shielding effectiveness can be maintained. Quantitative estimates of specific damage

to the radiator and converter have not been made. However, the recommended shipping container design offers considerable protection to the system on all surfaces and for all impact attitudes including hitting against the radiator cover.

Other accident sequences can be cited which result in a somewhat different environment. For example, the truck accident involving another truck with an explosive, flammable cargo can be postulated to consist of impact, explosion, fire, and water quench. The impact will be equivalent to the 30-foot drop for which it has been shown that containment and shielding effectiveness can be maintained. The explosion will not destroy the effectiveness of the system heat rejection nor materially damage the system. The blast wave analysis in Appendix F does not account for a shipping container, the effect of which will be to provide substantial blast wave blockage and further reduce the explosion effects. Detailed fire analyses have shown that the system alone will withstand the fire environment. The changes in the shipping container as a result of the impact and explosion will not affect the outcome of the fire environment. The shipping container is designed to yield and crush so that it will still retard the direct thermal radiation exchange between the fire and the system. Water quenching the system following a fire will not compromise shielding integrity. Results reported in Section VI-B 2-e indicate the shield will not crack when the system is exposed to a one-half 1850°F fire and followed by a water immersion of the bare heat source.

4. Siting

On site, the SNAP-23A system alone can be subjected to accidents and/or potential environments resulting from an accident. These environments are: fire, flood, vandalism, earth burial, deep or shallow water burial, and impact by ship.

In most cases, the probability and/or the severity of these environments will be known beforehand as the result of a site evaluation. Engineered safeguards can be used where the advantage of using the system outweighs a potential environmental hazard.

With two exceptions, the severity of the potential site environments should be no worse than those already examined in this Accident Analysis Section. For example, fire, impact, overpressure, oxidation, and water burial site environments can all be directly

related to the corresponding transportation accident evaluation. The exceptions, which appear to result in somewhat different responses than those documented, are earth burial and vandalism. In the latter category, penetration of the system by a high velocity projectile is of particular interest.

a) Earth Burial. Earth burial analyses were performed for two burial depths and two effective soil conductivities using the thermal model described in Appendix B. Recent studies (Reference VI-17) have verified that the soil conductivities assumed are still representative of earth burial in a low conductivity medium. Table VI-8 summarizes some typical values of soil conductivity.

It was assumed that the system was operating under an open circuit condition and that the radiator fins were removed. Prior studies (Reference VI-2) established that the presence of the fins had no major impact on the temperature of a buried system. The temperature profile in a soil of "average" thermal conductivity ($0.5 \text{ Btu/hr-ft-}^{\circ}\text{F}$) surrounding a system buried at a moderate depth (4 feet) is shown in Figure VI-25. The transient response of heat source components to this environment is presented in Figure VI-26 which reveals that shield melting will occur at the shield/accumulator interface in approximately 35 hours. Steady state temperatures at several locations within the system are summarized in Table VI-9 for a combination of burial conditions. A conductivity of $0.1 \text{ Btu/hr-ft-}^{\circ}\text{F}$ was chosen for some runs because it represents a lower bound in the conductivity range and hence yields the highest system burial temperatures.

The trends observed are as expected. Temperatures rise significantly when burial depth is increased or soil conductivity is lowered. Some combinations of soil conductivity and burial depth (e.g., $K = 0.5$, $h = 1$, and $K = 0.5$, $h = 4$) lead to results which indicate that fuel integrity can be maintained. Soil conductivity is the dominant factor. In all cases, some shield melting occurs (shield melts at about 2190°F). Fuel capsule integrity will be compromised if complete burial occurs in a very low conductivity soil. Engineered safeguards should, therefore, be employed to prevent earth burial of the system if the characteristics of a site make the system vulnerable to such an event.

b) Projectile Penetration. The literature can be used to define effects resulting from impact of a high velocity bullet into a target. Goldsmith (Reference VI-18) cites the following perforation results for a soft conical nose bullet travelling at 2500 to 2600 ft/sec:

TABLE VI-8
TYPICAL SOIL CONDUCTIVITIES

Material	Average Temperature (°F)	Thermal Conductivity (Btu/hr-ft-°F)
Sand		
Dry	68	0.2
10% water	68	0.6
Soil		
Dry	70	0.22
Wet	70	1.5

Figure VI-25. System Ground Burial; Conductivity = 0.5, Depth = 4 Feet

Figure VI-26. System Transient Temperatures with Ground Burial

TABLE VI-9
SUMMARY OF STEADY-STATE GROUND BURIAL

Complete SNAP-23A System buried with long axis vertical

Burial Depth (ft)	Soil Conductivity (Btu/ft-hr-°F)	Temperature (°F)			
		Fuel	Fuel Capsule Wall (1)	Midpoint of Shield Sidewall (2)	Radiator Root
1	0.1	3045	3825	2620	1531
1	0.5	2439	2218	2013	868
4	0.1	3400	3190	2980	1827
4	0.5	2510	2290	2080	889

(1) Melt temperature = 2310°F

(2) Melt temperature = 2190°F

- (1) The thickness, h_o , just penetrated for mild steel is 0.53"
- (2) For alloy steel plate the thickness is 0.43"
- (3) $h_o = 0.42"$ for annealed Ni-Cr steel
forged Mn-Cr steel
water quenched Mn steel
- (4) $h_o = 0.18"$ for Ni-Cr steel quenched

Reference VI-19 bases penetration on projectile momentum per unit area.

Considering just the heat source, the projectile must penetrate the Inconel shell, copper accumulator, U-8 Mo shield, and Hastelloy-C shells before hitting the fuel.

Reference VI-20 presents the following equation for the penetration of a copper, steel, lead, or aluminum projectile into a target of copper, steel, or aluminum:

$$P(\text{inches}) = K_t K_p (\rho_p V L - C)$$

where:

ρ_p = projectile density, slugs/in³

V = projectile velocity, ft/sec

L = projectile length

K_t, K_p, C = constants

Consider a steel projectile with a velocity of 2700 ft/sec impacting the heat source target.

Then

$$P = (0.029) \left[(0.0090) (2700) (0.5) - 0.6 \right]$$

$$P = 0.336 \text{ inches}$$

The case of the bullet impacting the SNAP-23A system is more difficult to evaluate because of the many layers involved. Consider the layers to be penetrated before the shielding is exposed.

Base assembly outer shell	0.125 Inconel 625
Base assembly inner bucket	0.125 Inconel 625
Heat source jacket body	0.060 Inconel 625
Copper accumulator	<u>0.300</u>
	0.610

The total thickness of these layers is 0.610 inch. In addition, the MIN-K insulation is 5.5 inches thick. The conclusion which can be drawn from these data is that the typical 30-06 rifle bullet will not reach the shield layer.

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VII. RADIOLOGICAL CONSEQUENCES OF A FUEL RELEASE

The accident analysis presented in the previous section indicates that the SNAP-23A fuel capsule will survive off-design and accident environments without compromising fuel containment. Nevertheless, an evaluation of the radiological consequences of a fuel release is included herein to provide some basis for estimating the potential hazard of very-low probability accidents.

The analyses presented below assume that a fuel release does occur without having specifically defined the mechanism by which fuel containment is lost. Because a fuel release is in fact a very remote possibility, the magnitudes of the source terms, and hence the radiological sequences, are somewhat arbitrary and open to conjecture.

As a result of the variety of accident environments which the SNAP-23A system might encounter during a typical mission, fuel releases have been assumed to fall into one of three general categories: underwater, underground, and airborne.

A. UNDERWATER RELEASE

It has been assumed that the entire fuel inventory of the SNAP-23A capsule dissolves over a period of about 10 years, based on measured dissolution rates in seawater. The resulting activity concentration in the water surrounding the generator has been calculated for release in both deep and shallow ocean water. In addition, a release after 280 years was considered.

Appropriate dispersion models for both deep and shallow ocean releases, which are described in Appendix E, have been chosen from the literature and from previous SNAP safety analyses (Reference VII-1). Briefly, the equation for each condition gives the activity concentration as a function of time and distance from the source. For the deep ocean case, the model represents diffusion with vertical mixing limited to a 1-meter layer near the ocean floor. The shallow ocean model has been derived from studies of dye patches in the Cape Kennedy area, and represents the more active mixing in shallow water due to shears and eddies. A current of 0.1 m/sec was assumed for shallow ocean. It was found

that the concentrations are generally higher in deep ocean since vertical mixing is very limited and horizontal diffusion is slow.

The index used to gauge the hazard for each release condition is the maximum permissible continuous concentration, MPCC, proposed in Reference VII-2. It is based on the allowable body burden received by man through ingestion of contaminated seafood. The MPCC is evaluated in Appendix E as $3.5 \times 10^{-5} \frac{\mu\text{c}}{\text{cc}}$.

The results from Appendix E are summarized in Table VII-1. The values shown for a fuel release after 280 years have been estimated by using the fact that the activity release rate will be reduced by a factor of about 1,000. Furthermore, a capsule breach is much less likely in shallow water due to the low hydrostatic pressure and demonstrated corrosion resistance of the fuel containment.

TABLE VII-1
RESULTS OF UNDERWATER RELEASE ANALYSIS

		Max. Permissible Continuous Concentration	Max. Distance from Source at Which MPCC Is Exceeded	Total Time That MPCC Is Exceeded
Deep Ocean	BOL	$3.5 \times 10^{-5} \frac{\mu\text{c}}{\text{cc}}$	< 1000 meters	10 years
	After 280 yr		~ 1 meters	10 years
Shallow Ocean	BOL		< 100 meters	2 hours
	After 280 yr		~ 1 meter, if at all	2 hours

It can be seen that an underwater release would be hazardous only within a small area close to the source. For shallow ocean release the MPCC would be exceeded not more than 100 meters from the source for about 2 hours or less. For a continuous release in deep water, the MPCC is exceeded in a small area enclosing a radius of approximately 190 meters for about 10 years. A release at 20,000 feet would not directly affect marine

animals which are ingested by man. The exact degree of hazard in this case cannot be determined, since little is known about uptake by deep sea marine life. At less than 9000 feet edible marine life could be contaminated. However, the degree of uptake would be very limited because of the small area in which the MPCC would be exceeded.

It may be concluded that an underwater fuel release will not present a severe safety hazard. As mentioned previously, the actual degree of hazard to man is dependent upon site factors such as the types of local marine life, existence of a fishing industry, and the magnitude and direction of local ocean currents. For a site in the vicinity of known fishing areas, it may be assumed that authorities will be aware of a possible fuel release and will take steps to prevent a hazard to the population.

B. UNDERGROUND RELEASE

Exposure of the strontium titanate fuel to the soil could present a hazard to humans through ground water contamination or vegetation uptake. The problem is considered briefly in this section, and a qualitative estimate of the hazard is made. Earth burial is the most obvious accident environment which could lead to an underground fuel release.

The thermal effects of burying the SNAP-23A System in earth have been presented in Section VI C. These analyses indicate that complete burial of the generator in low conductivity soil will result in melting of the shield and capsule walls. The relatively high melting point of the fuel makes it unlikely that fuel meltdown will ever occur. Some of the results of earth burial analyses are summarized in Figure VII-1.

A 4-foot burial of the generator in dry sand or soil allows the possibility that the biological shield and fuel capsule can exceed their respective melting points (Figure VII-1). Whether or not soil will melt depends upon the condition of the external structure of the SNAP-23A System as a result of the burial accident. A break in the rain cover or base assembly outermost shell will admit soil to the insulation cavities and perhaps to the heat source exterior. Contact of soil with the very high temperature heat source surfaces depends upon attitude, the amount of insulation degradation, and of course, the soil

Figure VII-1. Fuel, Capsule, and Shield Temperatures for Steady-State Ground Burial of the SNAP-23A System.

consistency. Assuming that a silica sand penetrates the insulation, some sand meltdown can occur (see Section VI C for sand properties).

It has been assumed that any molten materials would spread into the surrounding soil until thermal conditions allowed them to resolidify. Three distinct possibilities have therefore been considered for the final form:

1) The fuel, surrounding soil, and possibly some of the other generator materials solidify into an irregular glassy lump.

2) The fuel pellets remain nearly unchanged, since their melting point is much higher than the rest of the generator; however, the pellets may become enclosed in a crust of somewhat "sintered" soil.

3) The contact between the fuel and various other substances at high temperature causes the Sr-90 to change chemically into a more soluble form.

In reality, the final fuel form would probably be some combination of the last two forms. The hazard has been estimated qualitatively by considering each form separately.

1. Glassy Solid

A number of tests at Chalk River (References VII-3 and 4) indicated that burial of high-level fission products incorporated in glass is a very safe and permanent disposal method. The glasses used at Chalk River were chosen for their high resistance to leaching by water; however, any glassy mixture of fuel and soil should be much safer than forms (2) and (3) above. This view is supported by measurements made after recent underground nuclear explosions. According to Reference VII-5, much of the radioactive debris following an explosion is permanently contained within the glass-like material which lines the cavity walls and solidifies on the floor of the cavity. Thus, any formation of glassy solids is a desirable result of earth burial.

2. Unchanged or Encrusted Fuel Form

If the fuel pellets are essentially unchanged but are exposed to groundwater,

leaching will occur. The leach rate for the SrTiO_3 fuel form is estimated to be 0.01 mg/day/cm^2 (Reference VII-6) after an exposure of approximately 100 days. This corresponds to an initial rate of 0.41 curies/day if each fuel pellet has its total area exposed. This rate is conservative because:

- a) The pellets may be partially protected by encrusted soil and/or generator materials.
- b) For the shallow burials considered here, the water table would normally be well below the fuel. Significant leaching and transport would occur only when precipitation soaked into the soil.

It is not known to what extent SrTiO_3 is absorbed by the soil after leaching. However, experience with underground disposal of mixed fission products indicates that movement is slow. At Chalk River, where the soil is very shallow and permeable, kilocuries of fission products have been dissolved in nitric acid and pumped into soil pits (Reference VII-7). Movement through the soil was reported as extremely limited.

Considering the slow leach rate of SrTiO_3 , burial of the original fuel form is not expected to cause significant contamination of ground water or vegetation.

3. Sr-90 in Soluble Form

One would expect a more severe burial hazard if some of the SNAP-23A fuel changed to a soluble form. Transport could occur by diffusion through moist soil, by dissolution in new water soaking downward, and eventually by diffusion throughout the water table. Extensive data are available to show that movement through soil is very limited for nuclides in general and for soluble Sr-90 in particular. The result for mixed fission products at Chalk River was cited above; also, studies of underground nuclear explosions indicated that, due to absorption, the dissolved radionuclides move more slowly than the groundwater and do not constitute a danger to water supplies more than a few miles from the test site (Reference VII-5). Studies made specifically on soluble strontium have shown that it is very strongly fixed in soil by means of absorption and ion exchange. Experiments have been performed (Reference VII-8) in which Sr-90 was placed at various

levels in plots of soil to study uptake in different crops. It was found that the downward movement of Sr-90 was only 10-15 cm over a 4-6 year period. In fact, natural soil was tested as a means of removing radioactivity from water. According to Reference VII-9, from 87.4 to 95.9 percent of the Sr-90 was removed from water by natural soil filtration.

Consequently, it appears that even in soluble form, the movement of Sr-90 would be limited to distances from several feet to a few miles from the burial site, depending on the type of soil.

Thus far, several physical considerations have been ignored which further reduce the probability of a hazard due to earth burial. These are as follows:

- a) The buried system would probably be located quickly either visually or by infra-red or radiation detectors.
- b) The shield and capsule wall would melt only in very low conductivity soil i.e., very dry soil. In such dry soil, little or no ground water would be present to cause leaching.
- c) Dry soil areas are generally not used for crops, and the low water tables cannot support more than a few wells. Thus, the hazard to the population would be limited.

C. AIRBORNE RELEASE

Since the SNAP-23A system is intended for ground-based applications and employs a solid fuel form, the formation of inhalable fuel is very unlikely. The environments for which a release could be postulated are either (1) a high yield explosion causing partial vaporization of all components, or (2) a large, high temperature fire lasting about 20 hours. Neither of these environments is credible for the SNAP-23A system. Therefore, the analyses presented in Appendix D and discussed here are intended only to give a complete documentation of SNAP-23A nuclear safety.

The source term was postulated by consideration of the vapor pressure of the fuel at high temperature. It was found (see Appendix D) that the vapor pressure of SrO as a

function of temperature provides a conservative estimate of the SrTiO_3 vapor pressure curve. Then, the maximum mass loss rate per unit area is given by:

$$R = \frac{P}{\sqrt{2 \pi m k T}} \quad (\text{VII-1})$$

where

R is the rate of mass loss

P is the vapor pressure

m is the mass of molecules

k is the Boltzmann's constant

T is the absolute temperature

This expression applies for the solid in a vacuum. If any partial pressure of SrTiO_3 exists above the solid, the mass loss rate would be reduced. It is thought that Equation VII-1 may be conservative by as much as a factor of 100 for actual fire conditions. It has been assumed that one-half of the total fuel pellet area is available for vaporization, considering the initial stacking of the fuel pellets and the design strength of the containment. The release rate used in Appendix D is then $Q = 3.78 \times 10^{-8}$ ci/sec for a fire temperature of 1850°F .

It has been assumed that the released fuel is in the form of an inhalable aerosol with aerodynamic diameter less than 10μ . Due to the lack in information on the form of vaporized SrTiO_3 , the analyses in Appendix D are performed for both soluble and insoluble Sr-90.

Dispersion of the vaporized fuel is analyzed according to the Pasquill-Gifford model for a ground-level source. The largest isopleths for a given concentration occur in a stable atmosphere, the Pasquill Class "F" condition. This condition constitutes the worst case for airborne release and is considered in Appendix D. Values of the dispersion parameter as a function of downwind distance were obtained from Reference VII-10. Appendix D presents the activity concentration in air at breathing height as a function of

downwind distance for two typical wind speeds. The inhaled activity is obtained as a function of distance for a person exposed to the cloud for 30 minutes.

The radiological consequences are evaluated by means of a simple body model. The retention of soluble and insoluble particles in the lung is assumed be that given by the ICRP (see Table D-2). The subsequent distribution of activity is analyzed according to the model defined in Appendix D. Basically, the transfer fractions from the lung to the gastro-intestinal tract, blood, and bone are those given in Reference VII-11 for soluble and insoluble Sr-90. Equations for the rate of burden accumulation are integrated to obtain organ burden as a function of time, and inhaled activity for the upper and lower lung, the GI tract, and the bone. The dose rate is related to the burden by

$$\dot{D}_i(I, t) = \frac{E_i}{M_i} q_i(I, t) \quad (\text{VII-2})$$

where

$\dot{D}_i(I, t)$ is the dose rate to organ (i) as a
function of inhaled activity and time

E_i is the effective energy delivered to organ (i)
per curie of burden (see Appendix D)

M_i is the mass of organ i

$q_i(I, t)$ is the burden in organ (i) as a function
of inhaled activity and time

Combining the results of the dispersion analysis to obtain organ burden as a function of time after inhalation and total activity inhaled, i.e., $q(I, t)$, and then carrying out the indicated integration of Equation VII-2 for a time period of 50 years yields total doses to an organ as a function of the receptor's downwind distance.

It is found that the bone is the critical organ for strontium. Therefore, bone doses were determined for both a soluble and insoluble form of inhaled strontium as a function of the receptor's downwind distance. The results are given in Figure VII-2. It is seen that the doses are well below the allowable critical bone doses for the general public, which

Figure VII-2. 50-Year Dose for 30 Minute Exposure to Soluble and Insoluble Sr-90 as a Function of Distance Downwind from the Source

is 25 rem for a 50 year period (References VII-11 and 12). Thus, an airborne fuel release does not constitute a hazard to a person standing 100 meters or more downwind from the source. The analysis was not extended to distances less than 100 meters, since extrapolation of the dispersion data is somewhat questionable and the Pasquill-Gifford equation is not valid at close distances from a violent fire. It is reasoned that a fire or other event capable of releasing fuel would also raise the contaminants several hundred feet before they could disperse downwind. Therefore, airborne contamination less than 100 meters from the source would be minimal.

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VIII. APPLICABLE SAFETY TEST SUPPORT

A. RADIATION MEASUREMENTS ON FPMS HEAT SOURCE

1. Introduction

A radiation test was performed by ORNL with the SNAP-23A biological shield assembly on November 18, 1968 to verify the adequacy of the shield configuration and the analytical model used in the shield design. The SNAP-23A safety criteria specifies that the radiation level at the surface of the system will not exceed 200 mr/hr while the value at 1 meter from the system will be limited to 10 mr/hr. The radiation source used in this test was the SNAP-23A/FPMS fuel capsule, whose design characteristics are summarized in Table VIII-1.

2. Bare Fuel Capsule

Radiation measurement data were obtained with TLD's (thermoluminescent detectors) at various locations around the bare SNAP-23A capsule as shown in Figure VIII-1. Each TLD was analyzed and interpreted by ORNL personnel. In Table VIII-2, final experimental data have been tabulated for comparison with calculated dose rates.

3. Fuel Capsule Inside Biological Radiation Shield

After installing the SNAP-23A/FPMS fuel capsule in the U-8Mo biological shield, a series of radiation measurements were taken on the assembly shown in Figure VIII-2.

Cutie Pie dose rate instruments and x-ray film were used to make a comprehensive survey of radiation levels. Large x-ray films were placed over the ends of the shield in order to detect non-uniformities that might lead to radiation streaming. Qualitative interpretation of the film did not reveal any evidence of streaming. Smaller films were used at selected locations with similar results.

The two Cutie Pie meters used for the shield dose rate measurements were calibrated on the morning of the test. This calibration consisted of making a series of dose rate readings over the range of 50 mr/hr to 10 r/hr using a standard Ra^{226}Br source. These measurements were then compared to readings from the same standard source using a Victoreen Condenser "R"

TABLE VIII-1

SNAP-23A FPMS CAPSULE DATA

Fuel Form	- Sintered $^{90}\text{SrTiO}_3$ Pellets
Total Curies (1 November 1968)	- 165,700
Total Power (1 November 1968)	- 1127 watts
Weight Fuel	- 6,018 grams
Number of Fuel Pellets*	- 4
Height of 4 Pellet Stack	- 6.845 inches
Pellet Diameter	- 3.998 inches
Height of Single Pellet	- 1.3 to 1.9 inches
Average Density of 4 Pellet Stack	- 4.27 grams/cc
Average Power Density of 4 Pellet Stack	- 0.80 watts/cc
Outer Capsule Material	- Hastelloy-C 0.415 inch thick
Liner Material	- Hastelloy-C 0.035 inch thick
Capsule Size	- 8.026 inch x 4.990 inch diameter

* Pellets contain nickel mesh disks for increased strength. One pellet contains one percent SiO_2 by weight to promote compaction.

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Figure VIII-1. Dose Rate Measurement Positions for Bare Capsule

TABLE VIII-2

MEASURED VERSUS CALCULATED DOSE RATE FROM BARE SNAP-23A SOURCE*

Dose Point	Distance from Surface (cm)	TLD, Dose (R/hr)	Calculated Dose R/hr
2	50	223	298
3	100	70	84
5	50	395	360
6	100	114	120
8	50	626	833
9	100	185	240
11	50	808	872
12	100	208	243
20	100	185	240

* Capsule surface readings at dose points 1, 4, 7, 10, and 19 could not be taken because of the high temperature of the capsule.

Figure VIII-2. SNAP-23A Shield Test Assembly

meter which is periodically sent to the National Bureau of Standards for calibration against a primary standard. The maximum allowable deviation between the Cutie Pie instruments and the Victoreen is ± 10 percent over the entire range. It should also be noted that readings taken at "contact" with the Cutie Pie are effectively reading at about 1.5 inch from the surface being measured.

Figure VIII-3 describes the locations where radiation measurements were obtained whereas Table VIII-3 lists the coordinate, distance from surface, and dose rate measured by Cutie Pie and film for each of these locations. The last column contains the dose rate at each position calculated by the analytical model described in Appendix I and provides a comparison of predicted values versus measured data. Reasonably good agreement between theoretical and measured dose rates is indicated. Interpolation of the test data to radiation levels at contact with the surface of the system (approximately 150 mr/hr) and at 1 meter from the surface (approximately 9 mr/hr) reveals FPMS compliance with the system safety criteria without credit for attenuation by intervening heat source and base assembly components. It also verifies that the variation in molybdenum content (6.5 - 8.0%) and the presence of fine surface cracks from casting defects in the shield have had no effect on shielding effectiveness.

Figure VIII-3. SNAP-23A Shield-Dose Measurement Position

TABLE VIII-3
DOSE RATES FROM SNAP-23A SHIELD

Dose Point	Coordinate		Distance from Surface (cm)	Cutie Pie No. 1 mr/hr	Cutie Pie No. 2 mr/hr	Film Dose mr/hr	Calculated Dose mr/hr
	r (cm)	Z (cm)					
1	0.0	17.55	on surface	580	600	*	1127
2	0.0	67.55	50	42	44	---	63
3	0.0	117.55	100	18	18		22
5	6.55	17.55	on surface	240	220	*	473
6	6.55	117.55	100	15	16	---	21
7	---	---	on surface	180	200	*	315
8	---	---	50	20	20	---	15
9	---	---	150	6	7	---	5
10	14.55	10.5	on surface	200	240	403	249
11	64.55	10.5	50	38	38	---	34
12	14.55	5.25	on surface	385	390	533*	431
13	114.55	5.25	100	12	12	---	12
14	14.55	0.0	on surface	400	400	*	475
15	64.55	0.0	50	38	40	40	39
16	114.55	0.0	100	12	12	13	12
17	164.55	0.0	150	6	6	---	6
24	14.55	-5.25	on surface	320	320	*	341
25	114.55	-5.25	100	12	12	---	35
26	14.55	-10.5	on surface	220	220	---	249
27	64.55	-10.5	50	37	36	35	35
28	---	---	on surface	240	250	*	304
29	---	---	50	24	23	---	14
30	---	---	100	8	10	4.5	4
31	---	-18.8	on surface	300	310	*	176
32	---	-38.8	20	78	70	---	54
33	---	-118.8	100	8	8	---	7
37	---	-18.8	on surface	220	240	*	217
38	---	-18.8	on surface	250	260	*	209
39	---	-38.8	20	73	70	60	67
40	0.0	-18.8	on surface	260	240	*	356
41	0.0	-68.8	50	26	26	---	24
42	0.0	-118.8	100	10	9	0	8
43	0.0	-168.8	150	4	5	---	4
Bolt Holes	---	-18.3	on surface	230	230	---	---

* Film damaged by heat
 --- Dose not measured by this method

B. HYDROSTATIC PRESSURE TEST

1. Background

One of the nuclear safety design criteria specified for the SNAP-23A system stipulates that 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. Structural analysis of the fuel capsule design (Appendix C) indicates that the present design will meet this requirement. A test program was conducted at ORNL to verify that this design criterion was satisfied. The specific objectives of the hydrostatic test program were:

- The verification of the structural integrity of the SNAP-23A fuel capsule at a hydrostatic pressure equivalent to 20,000 feet of seawater. The corresponding hydrostatic pressure is 9,040 psig.
- Provide data for comparison with theoretical stress predictions.
- Obtain a quantitative estimate of the margin of safety in the design by subjecting the capsule to a hydrostatic pressure of 16,000 psig (which is equivalent to a seawater depth of 35,800 feet or the maximum ocean depth recorded). This test objective is not a safety requirement and is only intended to show where the failure point may occur.

Three SNAP-23A fuel capsules with the physical characteristics shown in Figure VIII-4 were prepared for the hydrostatic test series. Hastelloy-C was the encapsulation material and two fuel simulants were utilized. The first capsule (no. 23-6) was loaded with three non-radioactive SrTiO_3 pellets and one 1 9/16" high LAVA-A ceramic (aluminum silicate) pellet at the center of the capsule to shim the fuel load to the proper length. The second capsule (no. 23-4) was loaded with one LAVA-A pellet. The third capsule, which has not been tested as yet, was loaded with three non-radioactive SrTiO_3 and shimmed to length with a brass disk. The LAVA-A has a density of 0.085 lb/in³, is fired to 1000°F, and has a measured compressive strength of about 14,000 to 15,000 psi. Each capsule was pressurized to one atmosphere with helium gas.

Figure VIII-4. Geometric Details of a SNAP-23A Test Specimen

TABLE VIII-4
SNAP-23A HYDROSTATIC PRESSURE TEST PLAN

<u>Run</u>	<u>Capsule</u>	<u>Capsule Inventory Simulant</u>	<u>Hydrostatic Pressure (psig)</u>	<u>Simulated Ocean Depth (feet)</u>	<u>Water Temperature (°F)</u>	<u>Measurements</u>
<u>1</u>	<u>A</u>		0 1000 ± 100 2000 3000 4000 5000 6000 7000 8000 9040 ± 100	0 2250 4460 6720 8900 11160 13340 15540 17720 20000	<u>Ambient</u>	<u>Continuous Strain Readings</u>
<u>2</u>	<u>B</u>	<u>Simulant and Natural SrTiO₃</u>	0 1000 ± 100 2000 3000 4000 5000 6000 7000 8000 9040	0 2250 4460 6720 8900 11160 13340 15540 17720 20000	<u>Ambient</u>	<u>Continuous Strain Readings</u>
<u>3</u>	<u>C</u>	<u>Natural SrTiO₃ and Metal Shims</u>	0 11300 16000	0 25400 35800		<u>Scribe Lines</u>

Each test capsule was instrumented with strain gages and provided with scribed lines on the top and bottom covers and side wall to permit a visual measurement of any permanent strain.

Prior to instrumenting, each capsule received all standard inspections, such as leak check, dye penetrant, and ultrasonic inspection of all welds. All significant capsule dimensions shown in Figure VIII-4 were measured and recorded.

2. Facility Description

A high pressure test chamber located at the Oak Ridge National Laboratory (ORNL), Oak Ridge, Tennessee was selected for this test program. The test section is six inches in diameter by 23 inches long. Changes in hydrostatic pressure were made in 1000 psig increments, and measurements were taken at each 1000 psig level. Strain gage measurements were taken up to a pressure of 10,000 psig. Beyond this pressure, all testing was done without instrumentation.

3. Testing Program

Table VIII-4 lists the test conditions and measurements requested in the program plan.

Capsule 23-4 was actually subjected to hydrostatic pressures up to 10,000 psig during the first run. At each pressure setting, readings of strain were taken until equilibrium was achieved. The pressure was increased slowly to the next level. Continuous strain readings were taken between pressure increments. Following the test period, the hydrostatic pressure was returned to 0 psig. The capsule was then recycled to 9,000 psig in four steps and returned to 0 psig for a second time.

Capsule 23-6 was similarly subjected to hydrostatic pressures up to 9,2000 psig. After measurements were obtained at 1000 psig increments, the hydrostatic pressure was returned to 0 psig. The capsule was again subjected to hydrostatic pressure in five steps to 9,000 psig and returned to 0 psig for a second time.

Anomalies in the strain gauge and scribe mark data have made interpretation of these measurements inconclusive. However, a full weld inspection of each capsule after the

test, using a dye penetrant test, and an ultrasonic inspection established fuel capsule integrity and hence, verified that the present SNAP-23A capsule design will meet the hydrostatic pressure requirement Reference VIII-B1.

As previously noted, the third test, which subjected the capsule to a pressure of 16,000 psig pressure, is not a safety requirement and is only intended to estimate the design safety margin. Comparison of pre- and post-test nondestructive inspection, including dimensional, dye penetrant, and ultrasonic inspection, indicated no significant changes occurred. The capsule ends were found to be dished in a maximum of 0.030 in. Subsequent sectioning of the end cap welds showed that two small apparent discontinuities that were observed in both the pre- and post-test ultrasonic inspections were weld flaws. However, no evidence was found of any propagation of the flaws during the test. (Reference VIII-B2.)

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C. CORROSION TESTS AT U. S. NAVAL RADIOLOGICAL DEFENSE LABORATORY

1. Introduction

Electrochemical corrosion-rate measurements on materials used in the SNAP-23A System were made in seawater at room temperature at NRDL (Reference VIII-1). The materials examined included:

Copper	304 Stainless Steel
Hastelloy-C	Tantalum
Inconel-600	Titanium
Inconel-718	Uranium/8 percent molybdenum alloy

Experiments were conducted to determine the normal (i. e., uncoupled) corrosion rate of each material in seawater, a galvanic series of the materials in seawater, and the effect on the corrosion rate of galvanic coupling of the materials in seawater. From an interpretation of these results, an estimate was made as to how long the radioisotopic fuel would remain encapsulated after the influx of seawater due to an accident.

Unless otherwise noted, the data and discussion which follow have been extracted from Reference VIII-C1.

2. Measurement of Normal Corrosion Rates

Test specimens (electrodes) of the SNAP-23A materials were supplied to NRDL by Westinghouse Astronuclear Laboratory in the shape of right circular cylinders 0.5 inch in height and 0.25 inch in diameter. Each electrode had a 4-40 threaded stud extending from one end (integral with the electrode itself) for mounting on the electrode holder assembly of the electrochemical polarization apparatus. Before use, each electrode was polished and then ultrasonically cleaned in benzene followed by ethanol. Immediately before immersion in the test seawater, the electrodes were rinsed in running distilled water. Table VIII-5 lists the composition of the various test electrodes.

The seawater used for the corrosion tests was collected about a mile from the Farallon Islands, located about 20 miles off the northern California coast at San Francisco. Before use, the seawater was filtered through a Millipore filter (0.8 micron).

TABLE VIII-5
COMPOSITION OF TEST ELECTRODES*

Element	Cu	Hastelloy C	Inconel-600	Inconel-718	304SS	Tantalum	Titanium	U/8%-Mo
Al				0.51		5 ppm		
B				0.0035				
C		0.06	0.01	0.05	0.05	46 ppm	0.025	300 ppm
Ca						5 ppm		
Co		0.82		0.11	0.09	5 ppm		10 ppm
Cr		15.76	14.68	18.25	18.28	5 ppm		
Cu	99.96		0.03	0.05	0.17	5 ppm		
Fe		5.03	7.12	17.77	69.55	30 ppm	0.16	45 ppm
H						5 ppm		
Mg						5 ppm		
Mn		0.50	0.19	0.07	1.55	5 ppm		
Mo		15.91		3.07	0.40	5 ppm		7.5
N						21 ppm	0.021	
Nb				5.35		100 ppm		
Ni		57.25	77.76	53.70	9.33	5 ppm		125 ppm
O ₂						40 ppm		
P		0.016		0.004	0.023			
S		0.003	0.007	0.01	0.027			
Si		0.81	0.18	0.08	0.53	5 ppm		
Sn						5 ppm		
Ta						99.96		
Ti				0.97		5 ppm	99.79	
U								92.45
V		0.24				5 ppm		
W		3.61				75 ppm		
Zr						5 ppm		

* All entries are in percent except as noted.

Conditions of aeration and deaeration of the seawater were achieved by continuous purging with air or nitrogen. Measurements of the anodic and cathodic polarization of a metal electrode by an externally applied current or potential, provided the basic electrochemical data for the study of the corrosion behavior of the metal in a corrosive medium. (Reference VIII-C2) Polarization data can be obtained by two methods depending on whether the current or the potential is allowed to be the independent variable. When current is the independent variable the term galvanostatic polarization applies; when potential is the independent variable the term potentiostatic polarization applies. Galvanostatic polarization techniques were used for the determination of corrosion rates at NRDL. The theory behind the application of the technique has been adequately described in the literature (Reference VIII-C2.)

In essence the galvanostatic technique is to (in turn) cathodically and anodically polarize the test electrode by applying an external current. The degree of polarization, measured as an over-voltage E , is plotted against the logarithm of the applied current density, i . For values of E greater than about 50 mv such a plot should yield a straight line, with slope B_c for cathodic polarization and B_a for anodic polarization. Values of the corrosion current density, i_{corr} , can be obtained from such measurements in two ways. One method consists of solving for i_{corr} in the following equation:

$$\Delta E / \Delta i = \frac{B_a B_c}{(2.3) (i_{corr}) (B_a + B_c)} \quad (1)$$

where $\Delta E / \Delta i$ is the slope (on a linear - coordinate plot) near the region of the open circuit potential (i. e., E less than about 0.05 v) for which the change of E with i is linear. Figure VIII-5 illustrates the linear plot of cathodic overvoltage versus current density for Hastelloy-C in aerated seawater.

The second method of obtaining i_{corr} is the extrapolation of the line of slope B_c to the open-circuit potential. The current density at the intersection represents the dissolution rate of the electrode and hence its corrosion rate in terms of current density. Figure VIII-6 illustrates the data points for galvanostatic cathodic polarization of Hastelloy-C in aerated

Figure VIII-5. Linear Plot of Cathodic Overvoltage vs. Current Density for Hastelloy C in Aerated Seawater

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Figure VIII-6. Galvanostatic Cathodic Polarization of Hastelloy C in Aerated Seawater

seawater plotted on semi-log paper. The slope, B_c , is measured from the best fit line between the current density data points between 0.5 and 0.3 V of overvoltage. By extrapolating the line of slope B_c to zero overvoltage, the resultant point of intersection defines the current density which represents the dissolution rate of the electrode and hence its corrosion rate.

Faraday's Law was then used to translate the current density ($\mu\text{amp}/\text{cm}^2$) into a more immediately useful quantity such as mil per year (mpy). In application of Faraday's Law, the following valences were assumed for the corrosion products in seawater: +2 for copper, 304 stainless steel, Hastelloy-C, Inconel-600, and Inconel-718; +4 for titanium and uranium/8 percent-molybdenum; +5 for tantalum. The relationship between current density in $\mu\text{amp}/\text{cm}^2$ and corrosion rate in mpy is:

$$\text{corrosion rate (mpy)} = 0.129 \frac{\text{equivalent weight, g/g}}{\text{density, g/cm}^3} \times \text{current density (}\mu\text{amp/cm}^2\text{)} \quad (2)$$

where the equivalent weight and density refer to the electrode material.

Table VIII-6 lists the electrochemical data for the normal (uncoupled) corrosion rate of each of the SNAP-23A materials as determined by NRDL.

3. Determination of Galvanic Series

The open-circuit potential of each test electrode was measured, as a function of time, with respect to a saturated calomel reference electrode (SCE) in aerated and in deaerated seawater. All of the test electrodes were immersed in a common container while the measurements were made. These potentials were the basis for determining a galvanic series of the SNAP-23A materials.

4. Determination of Currents Due to Galvanic Coupling

The galvanic currents between each of the SNAP-23A materials in turn and all the others, as a group, when short-circuited (i.e., galvanically coupled) to each other, were

TABLE VIII-6

SUMMARY OF ELECTROCHEMICAL DATA FROM GALVANOSTATIC
POLARIZATION OF SNAP-23 MATERIALS

Material	Seawater Condition	β_c (V)	β_a (V)	$\Delta E/\Delta i$ (V/ $\mu\text{amp}/\text{cm}^2$)	i_{corr} ($\mu\text{amp}/\text{cm}^2$)		Normal Corrosion Rate (mpy)	
Cu	Aerated	0.214	0.227	7.5×10^{-3}	6.4*	(7.5)**	2.9*	(3.5)**
Cu	Deaerated	0.411	0.046	3.6×10^{-1}	5.0×10^{-2}	(1.3×10^{-1})	2.3×10^{-2}	(6.0×10^{-2})
HC	A	0.290	0.147	5.7	7.4×10^{-3}	(1.2×10^{-2})	3.5×10^{-3}	(5.7×10^{-3})
HC	D	0.270	0.250	4.8	1.2×10^{-2}	(1.0×10^{-2})	5.7×10^{-3}	(4.7×10^{-3})
I-600	A	0.116	0.281	3.3	1.1×10^{-2}	(1.1×10^{-2})	5.0×10^{-3}	(5.0×10^{-3})
I-600	D	0.214	0.290	3.9	1.4×10^{-2}	(1.2×10^{-2})	6.3×10^{-3}	(5.4×10^{-3})
I-718	A	0.300	0.157	7.8	5.7×10^{-3}	(7.0×10^{-3})	2.2×10^{-3}	(2.7×10^{-3})
I-718	D	0.205	0.124	4.2	8.0×10^{-3}	(1.1×10^{-2})	3.1×10^{-3}	(4.3×10^{-3})
304SS	A	0.177	0.178	1.1×10^1	3.4×10^{-3}	(3.0×10^{-3})	1.5×10^{-3}	(1.3×10^{-3})
304SS	D	0.229	0.243	8.7	5.9×10^{-3}	(5.3×10^{-3})	2.6×10^{-3}	(2.3×10^{-3})
Ta	A	0.145	0.290	2.0×10^1	2.1×10^{-3}	(3.0×10^{-3})	5.9×10^{-4}	(8.5×10^{-4})
Ta	D	0.420	0.212	3.1	2.0×10^{-2}	(3.2×10^{-2})	5.6×10^{-3}	(9.0×10^{-3})
Ti	A	0.118	0.241	1.0×10^1	3.3×10^{-3}	(1.6×10^{-3})	1.1×10^{-3}	(5.4×10^{-4})
Ti	D	0.235	0.190	6.9	6.6×10^{-3}	(6.8×10^{-3})	2.3×10^{-3}	(2.3×10^{-3})
U/8%-Mo	A	0.165	0.070	1.8×10^{-2}	1.2	(1.6)	5.1×10^{-1}	(6.8×10^{-1})
U/8%-Mo	D	0.184	0.125	8.4×10^{-1}	3.9×10^{-2}	(4.2×10^{-2})	1.7×10^{-2}	(1.8×10^{-2})

* Computed by corrosion equation (1).

** Data from B_C line extrapolation.

measured in aerated and deaerated seawater at NRDL. Results of these experiments are listed in Tables VIII- 7 and VIII- 8 for both aerated and deaerated seawater. It is assumed that the materials would corrode in the order shown by the galvanic series in Tables VIII- 7 and VIII- 8. Table VIII- 9 contains a summary listing of normal corrosion rates and galvanic corrosion rates by materials of construction.

5. Application of Results to the SNAP-23A System

Once seawater enters the SNAP-23A system, prediction of fuel containment from NRDL results requires two basic assumptions. These are:

- That the effective surface area (i. e., the surface area in contact with seawater) of each material is equal to its geometric surface area.
- That all materials are in electrical and electrolytic contact with each other.

Table VIII-10 shows the geometric surface area used for each of the materials. These surface areas were approximated from drawings of the SNAP-23A System which were furnished by WANL.

It was also assumed that the materials would corrode in the order shown by the galvanic series in Tables VIII- 7 and VIII- 8. Since the fuel capsule material, Hastelloy-C, is the material of most concern, the materials more cathodic in the galvanic series will not be considered except for their surface-area effect on the corrosion rates of the materials below them in the series. The case of aerated seawater will be considered first

The first material to corrode will be the uranium-8% molybdenum biological shield material.* Its surface-area ratio to the rest of the materials is about 1:9. This anode-to-cathode ratio would not significantly change the corrosion rate of this alloy, i. e., 3.4 mpy measured with an anode-to-cathode ratio 1:7). At this rate the biological shield, being three inches thick, would last about 8.8×10^2 years.

* Actually, the aluminum radiator fins would corrode before the U/8% Mo alloy. Based upon results obtained in a similar study on the SNAP-21 System, the aluminum fins would last only a few years.

TABLE VIII-7

GALVANIC CORROSION CURRENTS ($\mu\text{amp}/\text{cm}^2$) OF SNAP-23 MATERIALS IN AERATED SEAWATER

U-8%-Mo	Cu	I-600	Ta	Ti	304SS	HC	I-718	Σi_{cath}	Σi_{anod}
<u>6.3</u>	5.0	5.0×10^{-1}	1.8×10^{-1}	2.4×10^{-1}	4.3×10^{-1}	2.7×10^{-1}	2.8×10^{-1}	6.9	6.3
	<u>6.7×10^{-1}</u>	8.7×10^{-2}	2.7×10^{-2}	1.0×10^{-1}	2.0×10^{-1}	1.1×10^{-1}	1.8×10^{-1}	7.0×10^{-1}	6.7×10^{-1}
		<u>2.2×10^{-1}</u>	4.0×10^{-3}	1.7×10^{-2}	1.1×10^{-1}	3.1×10^{-2}	8.0×10^{-2}	2.4×10^{-1}	2.2×10^{-1}
			<u>9.7×10^{-3}</u>	<u>1.2×10^{-3}</u>	3.7×10^{-3}	1.9×10^{-3}	5.0×10^{-3}	1.1×10^{-2}	1.1×10^{-2}
				<u>2.0×10^{-3}</u>	2.7×10^{-4}	2.3×10^{-4}	1.6×10^{-3}	2.1×10^{-3}	2.0×10^{-3}
					<u>3.0×10^{-4}</u>	<u>2.2×10^{-4}</u>	5.3×10^{-4}	5.3×10^{-4}	5.2×10^{-4}
						<u>3.7×10^{-3}</u>	3.7×10^{-3}	3.7×10^{-3}	3.7×10^{-3}

TABLE VIII-8

GALVANIC CORROSION CURRENTS ($\mu\text{amp}/\text{cm}^2$) OF SNAP-23 MATERIALS IN DEAERATED SEAWATER

U/8%-Mo	Ta	Cu	HC	I-600	304SS	Ti	I-718	Σi_{cath}	Σi_{anod}
<u>2.1×10^{-2}</u>	<u>5.0×10^{-3}</u>	<u>1.7×10^{-3}</u>	1.0×10^{-3}	3.7×10^{-3}	6.0×10^{-3}	2.0×10^{-3}	6.3×10^{-3}	1.9×10^{-2}	2.8×10^{-2}
c_i	<u>7.0×10^{-3}</u>	<u>5.3×10^{-3}</u>	5.0×10^{-4}	1.7×10^{-3}	3.7×10^{-3}	1.4×10^{-3}	4.7×10^{-3}	1.2×10^{-2}	1.2×10^{-2}
		<u>8.3×10^{-3}</u>	8.3×10^{-4}	1.3×10^{-3}	2.5×10^{-3}	1.2×10^{-3}	4.3×10^{-3}	9.3×10^{-3}	8.3×10^{-3}
			<u>7.7×10^{-4}</u>	<u>5.7×10^{-4}</u>	<u>3.1×10^{-4}</u>	1.2×10^{-5}	1.6×10^{-3}	1.6×10^{-3}	1.7×10^{-3}
				<u>9.0×10^{-4}</u>	<u>3.3×10^{-4}</u>	<u>1.2×10^{-4}</u>	1.3×10^{-3}	1.3×10^{-3}	1.4×10^{-3}
					<u>6.7×10^{-4}</u>	<u>3.0×10^{-4}</u>	1.0×10^{-3}	1.0×10^{-3}	9.7×10^{-4}
						<u>5.7×10^{-4}</u>	5.7×10^{-4}	5.7×10^{-4}	5.7×10^{-4}

NOTE: Underlined currents denote anodic material in series.

TABLE VIII- 9
NORMAL AND GALVANIC CORROSION RATE OF SNAP-23A
MATERIALS IN SEAWATER

<u>Material</u>	<u>Seawater Condition</u>	<u>Normal Corrosion Rate (mpy)</u>	<u>Galvanic Corrosion Rate* (mpy)</u>
Cu	A	3.5	3.1×10^{-1}
Cu	D	6.0×10^{-2}	3.8×10^{-3}
HC	A	5.7×10^{-3}	1.8×10^{-3}
HC	D	5.7×10^{-3}	3.6×10^{-4}
I-600	A	5.0×10^{-3}	1.0×10^{-1}
I-600	D	6.3×10^{-3}	4.1×10^{-4}
I-718	A	2.7×10^{-3}	----
I-718	D	4.3×10^{-3}	----
304SS	A	1.5×10^{-3}	1.3×10^{-4}
304SS	D	2.6×10^{-3}	2.9×10^{-4}
Ta	A	8.5×10^{-4}	2.7×10^{-3}
Ta	D	9.0×10^{-3}	2.0×10^{-3}
Ti	A	1.1×10^{-3}	6.8×10^{-4}
Ti	D	2.3×10^{-3}	1.9×10^{-4}
U/8% Mo	A	6.8×10^{-1}	2.7
U/8% Mo	D	1.8×10^{-1}	8.9×10^{-3}

* Measured with unequal surface areas of anode and cathode.

TABLE VIII-10

APPROXIMATE SURFACE AREAS OF SNAP-23A MATERIALS

<u>Material</u>	<u>Surface Area (cm²)</u>
Cu	7.7×10^3
Hastelloy-C	1.1×10^3
Inconel 600	2.8×10^4
304 SS	1.8×10^4
Ta	1.3×10^3
Ti	1.1×10^3
U/8% Mo	7.4×10^3

The next material to corrode will be copper (accumulator). The anode-to-cathode surface-area ratio is about 1:8. This ratio will increase the corrosion rate of copper from 3.8 mpy (measured at an area ratio of 1:6) to about 5.0 mpy (measured at an area ratio of 1:6) to about 5.0 mpy since it is corroding under cathodic control. At this rate the thinnest portion of the copper accumulator, being 1.8 inch thick, would deteriorate in about 25 years. The thickest portion of the accumulator, being about 3.5 inch thick, would last proportionately longer.

Analogous calculations for the materials higher (more noble) in the galvanic series predict the following life times: Inconel-600, 6.3×10^3 years; tantalum, 5.6×10^4 years; titanium, 2.2×10^4 years; 304 SS, 7.8×10^4 years; Hastelloy-C, 5.3×10^4 years.

Similar calculations can be made for the case of deaerated seawater. However, it can be seen from the comparison of the corrosion rates in aerated and deaerated seawater (Table VIII-9) that the fuel would remain encapsulated for a comparable length of time in deaerated seawater.

Several comments can be made about the estimates of how long the fuel would remain contained after seawater entered the interior of the SNAP-23A system. The first concerns the two basic assumptions: effective surface area and all material being in electrical and

electrolytic contact with each other. In an actual situation, these assumptions are not very realistic since there are many interfaces into which the seawater could probably not enter. This means that the anode-to-cathode surface area ratios would not be the assumed geometric area ratios and/or some materials may be eliminated from the galvanic series. Probable changes in the anode-to-cathode surface area ratios would not significantly affect the corrosion rate of materials corroding under anodic control. The elimination of a material from the galvanic series would, of course, alter the order in which the other materials would corrode. Elimination of materials below Hastelloy-C in the galvanic series would not affect its corrosion rate; elimination of materials above it would decrease its corrosion rate.

Another fact which was not taken into account in the analysis, is the effect of accumulation of corrosion products on the surface of the corroding materials. In most cases, this would tend to slow down the corrosion rate.

The effect of other forms of corrosion (e.g., pitting, crevice, stress) was not considered in the above discussion and it is realized that for some of the materials these other forms of corrosion will play an important part. For example, local corrosion of the 304 SS and the Inconel alloys is expected to be relatively severe, particularly if a stagnant seawater environment exists. The effect of local corrosion on the performance of Hastelloy-C is expected to be very small.

6. General Conclusions

The NRDL experiment results indicate that, if seawater enters the interior of the SNAP-23A system, the fuel will remain contained for 3,470 half lives of strontium-90 or approximately 100,000 years.

REFERENCES

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- VIII-C-2 "Corrosion of Fuel Capsules By and Release of Radionuclide Fuels to Seawater, " S. Z. Mikhail and D. A. Kubose, Nuclear Metallurgy, Volume 14 Symposium on Materials for Radioisotope Heat Sources, Gatlinburg, Tennessee, October 1968.
- VIII-C-3 M. G. Fontana and N. D. Greene, Corrosion Engineering, New York, McGraw-Hill Book Company, 1967, Chapters 9 and 10.

D. ELECTRICALLY HEATED SYSTEM TESTS AT WANL

The objectives of the Experimental Mockup System (EMS) performance tests at WANL were to evaluate the SNAP-23A system performance at design operating conditions and to correlate the data with analytical models used during mechanical, thermal, and electrical design. In addition, tests were conducted at off design conditions to determine the effect on overall system performance. An electrical heat source was used during EMS testing to simulate the isotope heat source to be used in the FPMS and in future SNAP-23A systems.

Both chromel-alumel and iron constantan thermocouples were used to measure temperature conditions surrounding the thermoelectric converter and electrical heat source. Table VIII-11 and Figure VIII-7 obtained from Reference VIII D 1 indicate the number of thermocouples and their location within the EMS generator. All thermocouples were periodically monitored with back-up instrumentation shown in parentheses on Figure VIII-7. A total of 42 thermocouples were monitored continuously with 30 spare couples being available for verification measurements or as backup instrumentation. In addition to the temperature measurements, input heater power, output voltage, and current data were also continuously monitored. Internal pressure measurements from the heat source assembly and the base assembly were recorded for each period of data acquisition.

Experimental tests were conducted with the EMS to verify analytical predictions obtained with the WANL thermal model. The thermal performance of the EMS was predicted by two-dimensional thermal analyses utilizing the Westinghouse TOSS computer code. Another method utilizing the thermal analyzer program (TAP-4) at Hittman Associates, Inc. predicted temperatures at beginning and end of life for safety evaluations. The comparison of predicted temperatures with measured values provides a measure of the accuracy with which the TAP-4 and TOSS computed values compare with actual temperature measurements.

The experimental mockup system has been operating for over one year, and data have been obtained for many different generator operating conditions including:

TABLE VIII-11

DATA MONITORED CONTINUOUSLY DURING EMS TESTING

<u>14 Iron Constantan Thermocouples</u>	<u>T/C Number</u>
Converter	
1 Hot Frame Web	5 (6)
1 Hot Frame Mid-Plane	7
2 Cold Electrode, Upper	24, 23 (25)
2 Cold Electrode, Lower	32, 31 (33)
1 Hot Electrode, Upper	12 (11, 13)
2 Hot Electrode, Mid-Plane	16, 15 (17, 18)
3 Hot Electrode, Lower	19, 20, 21
2 Outer Case	47, 49 (50, 48)
28 Chromel Alumel Thermocouples	
<u>T/C Number</u>	
Radiator	
3 Fin Base, Vertical Traverse	206, 211, 216 (231, 236, 241)
*1 Power Conditioning Fin	246 (247)
**2 Fin, Lower Radial Traverse	217, 220 (242, 245)
1 Power Transistor	254 (255)
Electrical Heat Source	
1 Top	308
1 Bottom	307
**2 Wall	301, 306 (302, 303, 304, 305)
Base Assembly (External)	
*1 Bottom, Center	100
*1 Side, Center	112 (111, 113)
*1 Side, Air Inlet	114 (115, 116)
*1 Upper Rain Cover	123 (124, 125)
Base Assembly (Internal)	
*1 Inner Can, Bottom Center	151
1 Truss Support	157 (158)
1 Inner Can, Bottom Edge	153 (154)
Heat Source Assembly	
2 Stud, Surface	323, 329
*1 Stud, Center, Top	332
1 Stud, External, Top	333
1 Stud, External, Center	334
1 Stud, External, Lower	335
1 Stud, External, Converter T/C Groove	336, 337
1 Outer Jacket	342 (343)
1 Power Leak Feed Through	338 (339)
4 Power Measurements	
<u>Measurement No.</u>	
1 Heater Input Power	400
1 Converter Output Current	71
1 Converter Output Voltage	70
1 Load Current	75
2 Pressure Measurements	
1 Base Assembly Pressure	420
1 Heat Source Assembly Pressure	421

Alternates to the specified thermocouples are shown in parentheses

- * Omit during shock and vibration testing.
- ** Hottest T/C to be used is controller input.

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Figure VIII-7. General Layout of the Instrumentation for the EMS

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- 900 watt steady state (Kr-N₂ Fill Gas)
- End of life test point (Kr-N₂ Fill Gas)
- Beginning of life test point (Kr-N₂ Fill Gas)
- Beginning of life reduced by 30 watts (Kr-N₂ Fill Gas)
- Intermediate Test Point (Kr-N₂ Fill Gas)
- Intermediate Test Point (Kr-N₂ Fill Gas)
- BOL-30 watts (Insulation Evacuated)
- BOL-30 watts (N₂ Fill Gas)
- BOL-30 watts (Kr Fill Gas)
- BOL-30 watts before and after lateral shock and vibration
- BOL-30 watts before and after vertical shock and vibration plus vibration sweep.
- BOL Life Tests (1778 hours to > 10,000 hours)

Complete data were obtained during the electrical testing of the EMS system for a comparison of theoretical temperature points versus measured values.

Table VIII-12 has been obtained from Reference VIII D 2 and lists the temperature and electrical data for the EMS generator during the initial 440 hours of testing. Many of the temperatures listed are $\bar{T}$ values and are calculated by averaging two or more temperature readings. For more precise comparisons of theoretical versus measured temperature readings, it is necessary to identify the thermocouple location and compare on a point for point basis. Some deviations become obvious because the thermocouples and nodal points may not be at the same precise location. Engineering judgment is necessary to compare the analytical values with the thermocouple readings.

TABLE VIII- 12
SNAP-23A EMS - THERMAL TEST DATA

	Test Point 900 Watt Steady State	Test Point EOL	Test Point Intermediate	Test Point BOL	Test Point BOL Reduced by 30 Watts	Test Point at Base Assembly Evacuated (Vacuum Insulated)
System Input Power	898 watts	1112 watts	1286 watts	1439 watts	1404 watts	1410 watts
Heat Source	1007.5°F	1182.3°F	1294.°F	1397.°F	1384.3°F	1395.8°F
Maximum Heat Source	1015.	1191.	1303.	1407.	1394.	1406.
Bucket Bottom	957.3	1127.7	1236.7	1338.3	1326.3	1339.3
Bucket Mounting Ring	926.5	1092.5	1197.5	1295.5	1284.5	1291.5
Base Center	147.	164.	172.	186.	187.	170.
Base 6.5" Rad	120.5	131.	135.5	146.	147.	134.5
Side Base Assembly	90.	95.	98.	102.	104.	96.5
Stud	843.2	994.	1083.8	1168.4	1159.	1167.6
Hot Frame	792.3	933.	1017.7	1096.7	1088.	1096.3
Hot Cap	765.6	903.6	986.	1062.9	1054.8	1062.8
Cold Cap	160.2	174.	182.4	191.8	193.9	193.2
Cold Frame	136.	147.5	152.2	159.2	161.2	161.2
Cuter Case	134.5	145.8	149.7	156.2	157.2	157.7
Radiator Base	127.3	137.	142.7	149.	151.	151.
Converter Voltage	20.29	24.07	24.78	24.94	24.92	24.96
Converter Current	2.12	2.69	3.16	3.54	3.48	3.53
Converter Inst. O-C Voltage	31.40 volts	40.565	46.30	51.09	50.83	51.65
Converter Power Out	43.03	64.786	78.30	88.34	86.72	88.11
D.C. Voltage	19.77	23.41	23.94	23.94	23.95	23.95
Load Resistance Current	2.09	2.65	2.72	2.73	2.73	2.74
D.C. Power Out	41.34	62.172	65.16	65.47	65.38	65.62
Power Dissipated in P.C.	0.59	0.866	10.49	19.44	17.96	18.92
Power Loss in Leads	1.10	1.748	2.65	3.51	3.38	3.57
Time After Start of Test	122 Hours	196 Hours	216 Hours	265 Hours	343 Hours	437 Hours

The analytical thermal model for safety analysis was developed to handle both steady state and transient thermal conditions during postulated accident environments and is described in Appendix B. Figure VIII-8 compares the temperature predictions of the TOSS and TAP-4 computer programs with the thermocouple readings from the EMS test data. These readings were obtained early in the program at a test point equivalent to end of life (EOL) conditions for the SNAP-23A generator system. The results from both analytical thermal models show good agreement with actual measured temperature data and lends confidence to the thermal predictions made for the various accident conditions presented in this report.

Figure VIII-8. Comparison of Analytical Thermal Model Predictions with EMS Test Results (EOL)

REFERENCES

- VIII-D 1 WANL-3800-21, "SNAP-23A Quarterly Progress Report," June-August 1968.
- VIII-D 2 WANL-3800-25, "SNAP-23A Quarterly Progress Report," Sept-Nov. 1968.

E. COMPATIBILITY AND SHIELD DUCTILITY TEST AT WANL

Compatibility tests were performed at WANL to evaluate the long-term reactions between heat source components at normal and off-design temperatures. The program also included a study on the ductility of the U-8 Mo shield in the aged condition.

1. Compatibility of Heat Source Materials

Samples for compatibility testing (Reference VIII-E1) were prepared by bolting washers of the candidate heat source materials together in the order in which they appear in the heat source assembly. A titanium washer was included in the early tests since it was the prime candidate material for the diffusion barrier can. The photographs in part A and B of Figure VIII-9 show the separate heat source material compatibility washers and an assembled compatibility sample. Washers were produced 1/4 inch thick for all materials with the exception of the U-8 Mo* where a 3/8 inch thickness was used. Samples were also assembled from washers that were heat treated in air at 570°F for 21 hours to increase the oxide film on the materials. This was done to evaluate the effect of surface condition on the interaction between heat source materials. All samples were encapsulated in quartz capsules evacuated and backfilled with high purity argon before sealing. A photograph of an encapsulated sample is shown in Figure VIII-9, part C.

Samples of clean Hastelloy-C/Ti/U-Mo/Cu/Inconel-718 were placed in storage furnaces at regulated temperatures of 1000, 1200, and 1400°F to be withdrawn at test times of 100, 500, 1000, and 10,000 hours. In addition, one sample was heated to 1820°F for 3 hours to accelerate the reaction rates and ascertain the compatibility of materials at the higher temperatures.

* U-10 Mo alloy on hand at Westinghouse was incorporated in these tests as a substitute for U-8 Mo because of a long lead time on this latter alloy. It is believed that the two percent higher Mo content in this alloy will not affect the results of the compatibility tests.

(A)

COMPATIBILITY WASHERS

L TO R: HASTELLOY-C,
TITANIUM, U-10MO, CU,
INCONEL 718.

(B)

ASSEMBLED COMPATIBILITY
SAMPLE

L TO R: AS ABOVE

(C)

ENCAPSULATED
COMPATIBILITY SAMPLE

Figure VIII-9. Compatibility Materials and Samples

Photomicrographs of the test specimens showed the interaction between Hastelloy-C and titanium.

During the 3 hours of testing samples at 1820°F , liquifaction occurred between the titanium and Hastelloy-C, and U-10 Mo and copper couples. Extensive erosion of both titanium and copper was apparent; this result is predictable from the Ti-Ni and Cu-U binary phase systems in which eutectics occur in the vicinity of 1800°F . The evidence of so extensive a titanium-Hastelloy-C interaction at 1200, 1480, and 1800°F disqualified titanium for long-time service as a barrier material between the isotope capsule and radiation shield. Two alternate barrier materials (tantalum and tungsten) were selected and test specimens were prepared similar to the initial compatibility samples. Each sample consisted of compatibility washers of Hastelloy-C/Ta/U-Mo/W/Hastelloy-C bolted together with monel bolts and sealed in quartz. Furnace tests were conducted at 1000°F , 1200°F , and at 1400°F with sample withdrawals at 162, 2096, 5000, and 10,000 hours. Another sample was exposed to a high temperature transient condition of 1820°F for 3 hours. The results indicated that a liquid phase did not occur between the compatibility washers. In addition, microscopic examination in the vicinity of the interfaces between these materials revealed that little, if any, solid state interaction had occurred.

An assumed straight-line relationship between the diffusion zone thickness and the square root of time at 1400°F and 1200°F gave a diffusion thickness at 5000 and 10,000 hours of 29.7 mils and 42 mils, respectively. The zone thickness at 1200°F was estimated to be 12.7 mils at 10,000 hours. Measurements on the compatibility samples revealed that the zone thickness is smaller than the estimated value as shown in Table VIII-13.

The thickness of the diffusion measured for 10,000 hours at 1400°F is lower than the value measured for the 5012 hour test. Examination of the entire interface area showed that the diffusion zone was not uniform in thickness. Apparently, the interface was not in good initial contact or the interface was separated upon cooling when the furnace failed after about 5000 hours of exposure. Based on extrapolating the existing data, this zone should be about 23 mils thick at 10,000 hours.

TABLE VIII-13
COMPARISON OF CALCULATED AND ACTUAL
DIFFUSION ZONE THICKNESS BETWEEN
COPPER AND THE URANIUM-MOLY ALLOY

Time (Hrs.)	Temp. (°F)	Zone (Measured) Thickness (mils)	Calculation (mils)
5012	1400	19.3	29.7
10,000	1400	15.3*	42.0
		23.0**	
10,000	1200	6.4	12.7

*This value appears low

**Expected thickness based on extrapolation of all previous data.

The other objective of these tests was to demonstrate the compatibility between copper/Inconel 718, U-10Mo/tungsten, U-10Mo/tantalum, Hastelloy C/tungsten, and Hastelloy C/tantalum. All of these samples demonstrated compatibility in the 10,000 hour compatibility test program. At the Inconel 718/copper interface, no interdiffusion zone was present although the interfaces did show diffusion bonding.

The effectiveness of tantalum and tungsten as a diffusion barrier between Hastelloy-C and U-10Mo is illustrated in Figure VIII-10. No reaction zones were formed and no diffusion bonding occurred after 10,000 hours at 1400°F. These tests demonstrate that the selection of tantalum as a diffusion barrier was sound and that interaction between the fuel capsule and U-8 Mo shield should not occur.

Tests were also performed with oxidized compatibility samples at 1400°F to compare surface effects on the interaction of heat source materials with results from "clean" samples. In general, the interactions were at a much slower rate and were not as aggressive as for "clean" samples.

Figure VIII-10. Lack of Reaction, Showing the Effectiveness of Tantalum and Tungsten as a Diffusion Barrier Between U-10Mo and Hastelloy C.

Prior to completion of the long-term compatibility tests, an investigation was initiated to find a barrier material between copper and U-8 Mo. Thin layers of tantalum, tungsten, and aluminum oxide were tested and found unattractive because of thermal expansion, flaking, and cracking problems. Since the 10,000-hour compatibility tests indicated no apparent problems at 1400°F, the investigation of barrier material between U-8 Mo and copper was discontinued.

2. U-8Mo Impact Test Results

Tensile test results indicated that the tensile ductility of U-8Mo in the γ' + α condition in the temperature range of 500 to 900°F is very low (Reference VIII-E2) and, hence, would not be able to withstand any significant impact stresses without fracturing. Because of the stringent safety requirements of radioisotope power systems, and the associated precautions which must be taken to ensure safe transportation, it is necessary to know if the U-8Mo shield can withstand impact stresses in this temperature range without failure.

In order to evaluate the behavior of U-8Mo under high strain rate loading, notched and unnotched Charpy impact tests were made over the temperature range of 500 to 900°F. A series of test specimens were machined from as-cast U-8Mo rods, 3/4 inch in diameter and 12 inches long. Five notched specimens and five unnotched specimens were machined to the dimensions described in ASTM Standard Methods for "Notched Bar Impact Testing of Metallic Material," Designation E23-66, for a Type A Charpy (simple-beam) impact test specimen. The impact test specimens were cleaned with ethyl alcohol after machining and individually encapsulated in Vycor tubes filled with 1/3 atmosphere of pure argon. These encapsulated specimens were then heat treated by holding them for 96 hours at 900°F to ensure transformation to the γ' + α condition.

The impact test temperatures were attained by heating the test piece for 10 to 20 seconds in an infrared radiant heating chamber to a temperature somewhat above the test temperature and then soaking for 30 seconds at this temperature. At this time the specimens were transferred to the Charpy testing apparatus and the striker was released as the specimen temperature reached the desired test temperature. Table VIII-14 presents the results of the

impact tests in tabular form. A plot of energy absorbed versus temperature is presented in Figure VIII-11.

Analysis of the results of the impact tests indicated the following:

- 1) U-8Mo is notch sensitive.
- 2) Above 600°F the material in the $\gamma' + \alpha$ condition can be considered to be fairly ductile.

TABLE VIII-14
IMPACT TEST RESULTS FOR U-8Mo

Temperature (°F)	Energy Absorbed (ft-lbs)	
	Notched	Unnotched
500	7.5	40.0
600	16.0	120.0*
700	18.0	71.0
800	18.5	96.5
900	19.8	104.0

*Sample did not fracture.

3) The U-Mo in the $\gamma' + \alpha$ condition is not as brittle as indicated by the tensile tests previously reported.

4) Although impact test results are not directly comparable to tensile test results, it appears that the brittle behavior is strain-rate sensitive.

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Figure VIII-11. Charpy-Impact Test Results on U-8Mo

REFERENCES

- VIII-E1 WANL-3800-4, "SNAP-23A Quarterly Progress Report, August 1967.
VIII-E2 WANL-3800-9, SNAP-23A Quarterly Progress Report, December 1967.

F. CONVERTER FIRE TEST AT 3M

1. Objective

Safety analyses have revealed that heat rejection to the environment via the converter/radiator assembly must not be significantly compromised if shielding effectiveness and fuel containment are to be maintained. The objective of the converter fire test at 3M was to determine the effect of a fire on the overall thermal conductance of the SNAP-23A thermoelectric converter. To perform the thermal transient tests, special six-couple thermoelectric test modules were designed and fabricated as accurate mechanical and thermal mockups of the SNAP-23A converter. In addition, a test stand was constructed to supply heat for simulation of the temperature/time profile corresponding to a fire and to allow measurement of the steady-state thermal conductance of the module (Reference VIII-F1.)

2. Test Model and Facilities

The design characteristics of the fire test module are summarized in Table VIII-15 and illustrated in Figure VIII-12. The converter segment consisted of six thermoelectric couples which were representative of the SNAP-23A converter. Spacings between the thermoelectric elements and the distance between hot and cold frames were similar to the full size converter. Components such as thermoelectric elements, followers, springs, and cold frame were similar to SNAP-23A hardware. The test module was contained in a stainless steel tube with a wall as thin as practical from mechanical strength and fabricability considerations to minimize the total heat conducted from the hot frame to the cold side. A pinch-off tube was provided so that normal converter processing could be performed. Thermocouples were mounted in the outer case and hot frame to monitor hot and cold interface temperatures.

The module test stand is illustrated in Figure VIII-13 with the module mounted in position. An ARMCO iron thermal conductivity standard with calibrated differential thermocouples wires (constantan) located at a fixed spacing is clamped to the hot frame of the test module. This standard is used to measure the heat transfer through the module at steady-state hold points during the fire transient (for thermal conductance calculations).

Figure VIII-12. Six-Couple Fire Module

TABLE VIII-15
SIX-COUPLE FIRE TEST MODULE DESCRIPTION

MODULE DESIGN SUMMARY

Module Length	2.93 inches nominal
Module Diameter	1.86 inches nominal
No. Couples	Six
Couple Type	PMC-2 Design or Equivalent
Thermal Insulation	a) Min-K 130I (powdered) packed into Voids Around T/E Kegs b) Microquartz (pad) at Outer Wall Adjacent to Cold Frame
Cold Frame	Aluminum Type 2024-T4
Hot Frame and Cylindrical Wall	Stainless Steel Type 304
Outside Case (End Cover	Inconel 625
Couple Hardware	Identical to PMC-2 Design
Module Orientation During Test	Axis Horizontal
Backfill Gas	Argon (10 psia at BOL Conditions)

Figure VIII-13. Fire Module Test Stand

Copper foil is placed between the module and standard to enhance heat transfer. A cylindrical heater rated at 75 watts is placed around the standard to control the hot frame temperature and to represent heat transferred from the heat source to the converter. Heater control is manual.

A 500-watt heater block is clamped to the module cold frame which is used to control the cold frame temperature during the fire transient. A temperature sensing controller is used to provide automatic adjustment of heat to the cold frame. As an aid to control the cold frame temperature during the post-fire cooldown period, a cooling block is attached to the heater block. Liquid carbon dioxide is used as a coolant to remove heat from the module. The module and heater assembly is enclosed by Min-K 2000 insulation to minimize heat losses. Cylindrical guard heaters are located around the insulation adjacent to the standard and module to minimize radial thermal losses. Differential thermocouples are placed in the insulation to sense and automatically control the radial temperature drops. The fire module test stand is enclosed within a steel container as a safety precaution.

The calculated temperature input conditions for the converter hot frame and outer case are illustrated in Figures VIII-14 and VIII-15 for 30-minute 1475°F and 1850°F fires. These conditions represent anticipated thermal environments from transportation accidents but can also be considered a reasonable approximation to a potential site fire environment.

3. 1475°F Fire Test Results

The first simulated 1475°F fire test was performed at 3M on November 19, 1969. Results of these tests were considered preliminary in nature but indications were that the conductance values after test were slightly higher than pretest values. Verification tests were considered necessary to fully evaluate the module during post-test examination.

The second 1475°F fire test was performed at 3M on December 3, 1969 with results similar to the first fire test. Refinements were made in accountability of thermal losses and calculational techniques were revised. Results are listed in Table VIII-16. It was observed from a post-test inspection of the module internals that the cold end segment (bismuth-antimony telluride) of the "P" leg melted. This is normal since during this test the element

Figure VIII-14. Calculated Response of SNAP-23A Generator Components During 1475°F Half-Hour Fire

Figure VIII-15. Calculated Response of SNAP-23A Generator Components During 1850°F During Half-Hour Fire

TABLE VIII-16
THERMAL CONDUCTANCE VERSUS TEMPERATURE
BEFORE AND AFTER A 1475°F FIRE

Converter Hot Frame (°F)	Converter Outer Case (°F)	Pre-Fire Conductance (watts/°F)	Post-Fire Conductance (watts/°F)
1050	155	1.05	----
1100	400	1.11	----
1150	600	1.22	----
1170	600	----	1.34
1155	400	----	1.19
1130	190	----	1.07

temperature exceeded the melting temperature of the material. The followers were pushed out of the cold frame, replacing the molten p-leg segments, and were in good contact with the hot leg segments. The followers after shifting became locked in the cold frame due to warping and could not be removed. The reduced length of the "P" legs would account for the increased conductivity observed after the test.

4. 1850°F Fire Test Results

Following the second 1475°F fire test, the test stand was prepared for the 1850°F fire test. Steady-state equilibrium values were obtained with a new fire test model on December 5, 1969 and December 9, 1969. Fire test runs were continued to simulate the temperature distribution shown on Figure VIII-15. Equilibrium values were obtained at 400°F and 600°F on the outer case, before and after peak heating. Table VIII-17 lists the corresponding hot frame and outer case temperatures before and after peak heating. Thermal conductance values are also listed for the appropriate data points.

Post-test inspection indicated that the module experienced some melting in

TABLE VIII-17
THERMAL CONDUCTANCE VERSUS TEMPERATURE
BEFORE AND AFTER AN 1850° F FIRE

Converter Hot Frame (°F)	Converter Outer Case (°F)	Pre-Fire Conductance (watts/°F)	Post-Fire Conductance (watts/°F)
1073	171	0.976	----
1038	161	1.04	----
1088	400	1.17	----
1089	400	1.12	----
1128	600	1.23	----
1129	600	1.23	----
1131	602	1.22	----
1106	598	----	1.97
1104	600	----	2.02
1103	599	----	1.96
1055	402	----	1.88
1054	401	----	1.91
1052	400	----	1.87
960	202	----	1.97

addition to that noted in the 1475°F fire test. As before, the cold segment bismuth-antimony-telluride (BAT) material in the p-legs melted. The followers shifted out of the cold frame and squeezed the BAT material out and around the p-leg hot segment elements. The net result was a decreased length and increased area for the p-leg elements. The n-leg elements were skewed and the thermoelectric material had coated the hot and cold caps, indicating that the n-legs had experienced some melting. Both p and n-leg materials were in good contact with the hot junction and followers. The top of the aluminum cold frame melted down to the first couple, with the molten material running into the void area around the springs. The increased conductivity is probably due to the reduced heat path resulting from the shortening and widening of the legs due to melting.

5. Conclusions

The results from the simulated fire test indicate the following:

- a) A 1475°F one-half hour fire will cause little change in the SNAP-23A converter thermal conductance. If changes occur, the thermal conductance will increase rather than decrease.
- b) During an 1850°F one-half hour fire, melting of components in the thermopile and aluminum heat accumulator can be expected. However, the thermal conductance of the converter will increase and help reject heat from the heat source.

REFERENCES

- VIII-F1 MMM-3824-1203, "SNAP-23A Converter Fire Safety Evaluation Test Plan and Procedures, "September 1969.

IX. PROCEDURAL REQUIREMENTS, SITING, AND RADIATION DOSE RATES

Each phase of a SNAP-23A mission must be subject to procedural controls in order to ensure system performance and safety. The amount of detail prepared for each procedure is a variable, dependent on the criticality of the operation. Although it is not possible at this time to specify the exact number and sequence of procedures required for the SNAP-23A, it is feasible to divide the types of procedures into the following categories:

- Assembly and Disassembly
- Maintenance
- Handling and Transportation
- Test and Operation
- Siting

The procedures will be dependent in large measure on the radiation dose rates associated with the various configurations encountered from fuel capsule loading to system deployment. A discussion on the expected dose rates is included and completes this section.

A. ASSEMBLY AND DISASSEMBLY

Assembly and disassembly procedures are required for the heat source, the subsystems, and the system. These are step by step, carefully controlled and supervised operations. Where the heat source is a part of the operation, either directly or indirectly, the steps of the procedure are rechecked and verified prior to its initial use.

Time is often a critical factor in determining the implementation of a particular procedure. For instance, the elapsed time between the installation of the heat source into its support ring and mounting the converter on the heat source stud will play a significant role in establishing the assembly technique to prevent thermal damage. The total detail needed for capsule assembly will not be required for every assembly and disassembly; a certain amount of discretion must be exercised.

1. Heat Source

Certain aspects of the heat source assembly process have already been written into procedure form. All welding is leak checked, for example, to verify capsule and heat source integrity.

The acceptable leak rate is 1×10^{-8} cc/sec. Component and subsystem assembly is performed by WANL and ORNL personnel under the direction of WANL representatives. The WANL Quality Assurance Department will follow through on all operations.

2. System

Installation of the heat source and converter are among the more critical steps of system assembly and disassembly. The converter installation determines, to a large extent, the thermal impedance between the converter and the heat source. This in turn affects the converter hot junction temperature and the eventual power output.

B. MAINTENANCE

Maintenance procedures will cover the two related areas of preventive maintenance and emergency maintenance. Preventive maintenance (PM) is a planned endeavor. Its primary purpose is to leave the system in better condition than it was prior to the procedure. Like an assembly procedure, PM is performed in detail. A major PM procedure might be the changing of converters with the system on site. In this case prior analysis of thermal conditions would be necessary to minimize the possibility of damaging the system. Determining which PM can be performed in the field, and which must be performed in the factory is a major decision.

Emergency maintenance, in contrast to preventive maintenance, is unplanned and may be performed by untrained personnel. Every emergency situation cannot be preconceived and covered by a procedure. Therefore, emergency procedures must be in a general form, denoting especially what should not be done rather than what should be done. Diagnostic techniques and allowable time elements are also necessary.

1. Heat Source

No specific heat source maintenance is envisioned at this time. Should the need

arise through an unforeseen circumstance, detailed sequencing must be written down and approved before any action is undertaken. This will apply in field maintenance as well as shop maintenance.

2. System

Except for PM on the converter and power conditioner, the system is designed to be maintenance free. Prior to performing PM on these two components, analyses must be performed and decisions reached on the limits and conditions for in-place maintenance. Criteria must be established based on test data readily obtainable through a series of diagnostic procedures. Maintenance personnel should be trained on operating units in a controlled, factory environment before field maintenance is attempted.

3. Shipping Container

The shipping containers are designed as reusable equipment. Prior to each shipping operation, the containers should be preassembled, load tested, and visually inspected for damage. Criteria for repair or replacement must be established prior to the inspection. Lifting rigs must also be pretested with overloads before the actual lift, or on a periodic basis.

C. HANDLING AND TRANSPORTATION

Any special equipment designed for the system, the heat source, or the shipping containers should be described and its operations explained. Special techniques for loading and unloading the unit of vehicles must be considered. Quality control in the form of an objective observer is a necessity during all operations. Details such as the sequence of tie down, bolt torques, and load limits of materials handling equipment must be defined. Each step of each procedure must be checked for completeness as well as accuracy.

1. Heat Source

The heat source is shipped in a specially designed shipping container that protects the heat source from damage during credible transportation accidents. The single accident which can be damaging to the heat source in the shipping container is earth burial. In the event of earth burial, the surface temperature would no longer be controllable. To obviate

this type of accident, route selection for transporting the heat source from ORNL to WANL has been restricted to major, limited access roadways where the potential for earth burial is minimal. Shipment will be in accordance with applicable AEC and DOT regulations. The preshipment radiation survey will include both swipe tests for removable activity and dose rate measurements at preset distances from the surface of the shipping container.

2. SNAP-23A System

The SNAP-23A system will be shipped in a specially designed container that is capable of protecting the heat source from damage in any credible transport accident. Prior to the first system shipment, a decision will be reached regarding whether to ship the generator with the converter in place or with a dummy heat rejection radiator. As in the heat source shipping situation, only those accidents that could conceivably cause over-temperature of the heat source must be considered in safety procedures. However, all practical measures will be taken to prevent impairment of the system operating capability. Earth burial is treated in the same manner as it was for the heat source shipping container. Damage to the heat rejection system by impact, fire, etc., is a potential problem area. Replacing the actual heat rejection system with a heat resistant dummy can provide a safety benefit.

D. TEST AND OPERATION

Test and operating procedures are naturally tied to the particular test or operation being performed. Key factors to be considered are the preparations for each hardware operation, the detailed steps, and the post-test requirements. Objective observers are as much a part of the test as are the detail requirements. Pass or fail criteria should be included where applicable. Data sheets prepared as part of the procedures make a convenient recording medium. The overall sequence of tests and operations must be controlled by a single document. Deviations from the plan should undergo a significant review before they are permitted.

1. Heat Source

A minimum of testing will be required on the heat source assembly. There should

not be any reason to disturb the containment welds or remove any sections of shielding. Tests involving the heat source as part of the system are treated in the next section.

2. System

System tests are a concern of this document only if they provide a mechanism that could eventually lead to fuel release or shielding degradation. The primary danger to the populace arises from any situation which creates severe overheating and eventual capsule melt. Consideration should be given to avoiding any such possibilities when the system is under test or continuous operation.

E. SITING

The SNAP-23A radioisotope power system is designed to operate unattended for 10 years. It may be used in more than one location during this time period. Some of the applications considered for the 60-watt (e) system are as follows:

- 1) Seismological Stations
- 2) Weather Stations
- 3) Light Houses
- 4) Microwave Repeater Stations
- 5) Ocean Moored Buoys
- 6) Survival Shelters

In general, the SNAP-23A system can be transported and placed into service at numerous locations without danger to surrounding population. It is very probable that response of the system to the environments encountered at most sites can be determined from the data in various sections of this report and the final SNAP-23A Safety Report. No special precautions or siting procedures are anticipated unless the siting evaluation reveals that a substantial probability exists for earth burial, high velocity projectile impact, etc. Assuming the probability exists, engineered safeguards may be justified as a means of obtaining a selected level of confidence that a generator hazard can be eliminated.

When a location is selected, an "Operational Site Safety Analysis Report" will be prepared for the specific application and installation. This report can, in general,

provide detailed information on the following subjects:

- 1) Site Location
- 2) Geography
- 3) Population Density and Distribution
- 4) Hydrology
- 5) Meteorology
- 6) Geology
- 7) Atmospheric Radioactivity
- 8) Description of Nearest Manned Base

In addition to site evaluation, the Operational Site Safety Analysis Report (SAR) must contain recommendations for site protection features (engineered safeguards) if they are required. Though a significant amount of effort has been expended on preliminary safety analyses, these have been directed toward establishing a defined hazard severity associated with a specified system response to a variety of hypothetical environments and conditions. In the preparation of the Site SAR, the accident and hazard potential of each site must be compared to the appropriately defined hazard severity described in this document. Situations may arise, however, where a potential site is in an area indigenous to specific environmental considerations which could produce a situation not comparable to those associated with a defined hazard severity. Such sites are not automatically disqualified from consideration, but they must be evaluated to determine if there are feasible engineered safeguards that will effectively offset the additional hazard severity.

In selecting a site, both the natural and man-made hazard potential must be considered. The inherent natural hazard potential for system damage must be clearly defined for each candidate site. For example, seismological stations may be in areas of high natural hazard potential. Likewise, desert locations are subject to sandstorms, and in a severe storm the system may be buried or partially buried. Mountainous locations such as might be preferred for microwave relay stations or weather stations are subject to rock slides and avalanches.

The natural hazard potentials of a particular site may be related to a defined hazard severity analysis contained in this document. If not, the particular hazard in question must be analyzed in the Operational Site Safety Analysis Report.

Man-made hazards may be either existing or predicted. If the site is in a remote area used for hunting, there is an existing hazard potential from hunters firing at the system. If the system is located on an ocean buoy, there is a predictable hazard potential from collision with ocean vessels. It is important that the site evaluation consider both present and future circumstances in preparing the Site Safety Analysis Report.

If an existing or predictable hazard potential disqualifies the candidate site, consideration should be given to the use of engineered safeguards to make the site useable. For example, bulletproof armor plate can be arranged to protect the system from stray or directed projectiles. A small enclosure or chain link fence may be sufficient to ward off falling rocks or retard vandalism. A concrete foundation may be required if the ground tends to soften at any time in the year. In particular, permafrost will present special problems best solved by those who deal with it on a regular basis.

Obviously, judgement must be employed to determine whether a particular site is worth the economic investment in protective features. This will depend not only on the site qualification, but also on the availability of alternate sites and their inherent problems.

F. RADIATION DOSE RATES

The normal radiation dose rates associated with each major phase of the SNAP-23A deployment are noted in this section. The operations analysis sequence for the system is shown in Figure IX-1. It can be seen from this sequence that all fuel encapsulation and heat source assembly operations are performed by ORNL personnel. Procedures and safeguards are consistent with established AEC practice. The radiation dose rates for the base fuel capsule dictate that all fueling operations prior to and including shield installation be performed in hot cells. Fueling, closure, decontamination, and swipe test checking of the inner liner are accomplished before installation and closure of the outer capsule shell. The radiation shield is installed after the completed and decontaminated capsule has been

Figure IX-1. SNAP-23A Operations Analysis Sequence

subjected to a number of tests, including leak, dimensional, dye penetrant, ultrasonic checks, swipe tests, colorimetry, and dose rate measurements. Table IX-1 presents the radiation levels anticipated for the several major configurations encountered with the SNAP-23A System. These are nominal values projected from test data and calculations. The point of reference is on the side of the unit; dose rates are taken radially from the fuel capsule geometric center.

Section VIII-A and Appendix I present the initial calculations and test data employed to determine the allowable working times for assembly and maintenance personnel. The test data were obtained using a fuel capsule with 165,700 curies of SrTiO_3 . This was the first available fuel loading in a SNAP-23A capsule. The data in Table IX-1 are scaled from the original test data and calculations to represent a maximum reference fuel loading of 185,000 curies. The scaling is a direct ratio and it is slightly conservative since the shielding properties of the additional fuel are not considered. The table is divided to show the allowable working period for both 1.25 rem and 3.0 rem allowable quarterly dose rates. Data for the shipping container configurations were obtained by extrapolating the heat source data with an inverse square relationship. Working times have been obtained by taking a ratio of the allowable dose rate to the actual dose rate. The specific configurations are discussed in the following paragraphs.

1. Heat Source Alone

The heat source, including the radiation shield, may be handled at several points in the system life. Prior to installation in the system, the heat source is handled at ORNL and at WANL. On-site maintenance may require exposing part of the heat source, although this is not anticipated. At EOL, the system will probably be disassembled at WANL and the heat source shipped to ORNL for final disposal. During these time periods, Table IX-1 indicates that an assembly or maintenance worker is restricted to 28 hours of operation working at normal arm extension around the heat source. The restriction is arrived at by comparing the whole body dose at 50 cm against the hand and forearm dose at the surface. The latter is 42 hours and the former is 28 hours. The 28 hour period, therefore,

TABLE IX-1

RADIATION DOSE RATE DATA FOR VARIOUS PHASES OF HEAT SOURCE AND SYSTEM HANDLING

Reference Fuel Loading = 185,000 curies

Configuration	Measured or Calculated Dose Rate (mrem/hr)	Scaled Dose Rate (mrem/hr)	Allowable Radiation Worker Exposure Time per Calendar Quarter		
			Whole Body 1.25R	3.0R	Hands and Forearms 18.75R
Bare Fuel Capsule					
50 cm	808×10^3	903×10^3	N/A	N/A	N/A
100 cm	208×10^3	232×10^3	N/A	N/A	N/A
Heat Source Assembly					
Surface	400	447	2.8	6.7	42
50 cm	40	45	28.0	67.0	420
100 cm	12	13.4	93.3	224.0	1400
SNAP-23A System*					
Surface	160	179	7.0	16.8	105
50 cm	25	28	44.6	107.0	670
100 cm	9	10	125	300.0	1875
Heat Source Shipping Container*					
Surface		53	23.6	56.6	345
50 cm		15	83.0	200.0	1250
100 cm		7	178.0	430.0	Continuous
System Shipping Container*					
Inner Container					
Surface		45	27.8	67.0	415
50 cm		13	96.0	231.0	1440
100 cm		6	208.0	500.0	Continuous
Air Transport Configuration					
Surface		13	96.0	231.0	1440
50 cm		6	208.0	500.0	Continuous
100 cm		4	313.0	Continuous	Continuous

* Extrapolated from test data

becomes limiting. The 2.8 hour period is not considered because there is no reason for a person to be working with his body against the heat source surface.

It must, of course, be understood that these estimates are not substitutes for dose rate monitoring equipment. Wrist film badges and body film badges are recommended for all personnel intimately concerned with heat source operations.

2. Heat Source in Shipping Container

The heat source will be shipped from ORNL to WANL at BOL and from WANL to ORNL at EOL. The heat source shipping container will be employed for all heat source shipments. In the event of an accident, it may be required that the heat source be shipped to ORNL direct from the field. The heat source shipping container is about 42 inches in diameter. The dose rate at the container's surface will be 53 mrem/hr, and at 50 cm it will be approximately 15 mrem/hr. The 50 cm value is employed here to best simulate the operations involved in loading and unloading the shipping container. The working time estimated at this distance is 83 hours.

3. SNAP-23A System

The SNAP-23A system will be subjected to numerous performance tests both at WANL and in the field. In addition, fueling, defueling, and maintenance will bring the operations personnel into close contact with the system. Table IX-1 shows that the whole body dose rate at 50 cm is more severe than the hand and forearm dose rate at the surface. The 44.6 hours is therefore considered the limiting time. Again, this value is only an estimate to be employed in conjunction with monitoring devices. Since the actual time required to test the system may well exceed this estimate, operating procedures and test equipment should be designed to minimize radiation exposures.

4. SNAP-23A System in Shipping Container

The system shipping container should be employed for all interplant or site transportation requirements. As presently conceived, the shipping container can be employed both with and without its air transport capability (see Section V-B). The inner container without the air transport capability is a cube, 4 feet on a side. The predicted

radiation levels at the closest point to the system are approximately the same as those predicted for the heat source shipping container. The working times around the container are not a problem. When the air transport capability is added to the inner container, the cube size is increased to approximately 7 feet on one side. The radiation levels at this distance are low enough to permit unlimited working access (>2000 hours per quarter).

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APPENDIX A

MATERIALS PROPERTY DATA

A. INTRODUCTION

The analyses presented in this report have utilized a number of thermophysical properties for the materials employed in the SNAP-23A System. The objective of this appendix is to briefly summarize the properties actually used for each different material.

The various SNAP-23A System components considered in this PSAR have been fabricated from the following materials:

<u>Material</u>	<u>Component</u>
Strontium Titanate	Fuel
Hastelloy-C	Fuel capsule liner
	Fuel capsule strength member
U-8 w/o Mo	Biological shield
OF Copper	Heat Accumulator
Inconel 625	Base assembly outermost shell
	Rain Cover
	Radiator protective cover
	T/E converter cold frame outer case
304 Stainless Steel	Heat source jacket
	T/E converter hot frame
	Hold down bolt
Inconel 718	Heat source jacket
	Heat source support system
Min-K 2002	Insulation
1100 Aluminum	Radiator fins
2024 Aluminum	T/E converter cold frame
Tantalum	Fuel capsule diffusion barrier

B. STRONTIUM TITANATE, SrTiO_3

1.	<u>Thermophysical Properties</u>	<u>References</u>
a.	Density - 18.2 lb/in^3	A-1
b.	Melting Point - 3700°F	A-2
c.	Boiling Point - $4530 - 5430^\circ\text{F}$	A-3
d.	Vapor Pressure - $3 \times 10^{-7} \text{ atm. at } 2966^\circ\text{F}$	A-4
e.	Thermal Conductivity - 4.18 BTU/ft. hr. $^\circ\text{F}$ at 70°F 1.56 BTU/ft. hr. $^\circ\text{F}$ at 1472°F	A-1
f.	Specific Heat - 0.154 BTU/lb. $^\circ\text{F}$ at 1200°F 0.159 BTU/lb. $^\circ\text{F}$ at 1600°F 0.162 BTU/lb. $^\circ\text{F}$ at 2000°F 0.166 BTU/lb. $^\circ\text{F}$ at 2500°F 0.168 BTU/lb. $^\circ\text{F}$ at 3000°F 0.169 BTU/lb. $^\circ\text{F}$ at 3300°F	A-5
g.	Coefficient of Thermal Expansion - $6.22 \times 10^{-6} / ^\circ\text{F}$ ($212 - 1300^\circ\text{F}$)	A-1
2.	<u>Radioactive Properties</u>	
a.	Activity* 149 curies of Sr-90/watt 34.5 curies of Sr-90/gm of SrTiO_3	A-6, 7, 8
b.	Specific Power* - $0.23 \text{ watt/gm of } \text{SrTiO}_3$	A-5
c.	Power Density* - $11.5 - 13.1 \text{ watt/in}^3 \text{ of } \text{SrTiO}_3$	A-5
d.	Half-life Sr-90 27.7 years Y-90 64.2 hours	

* These values will vary somewhat depending on the age of the fuel as well as on processing and manufacturing variables.

- e. Decay Product - zirconium A-5
- f. Radiation Emitted A-5
- Sr-90 $\longrightarrow$ Y-90 100% beta 0.54 Mev
- Y-90 $\longrightarrow$ Zr-90 ~ 100% beta 2.26 Mev
- g. Radiation Attenuation
- The tenth value layer for the high energy component of bremsstrahlung is 1.6 in. of lead.

C. HASTELLOY-C

1. Composition of Cast Alloy, Weight Percent A-9

Ni	47-60	Si	1.00 maximum
Cr	15.50 - 17.50	Mn	1.00 maximum
Mo	16.00 - 18.00	C	0.12 maximum
W	3.75 - 5.25	V	0.20 - 0.40
Fe	4.50 - 7.00	P	0.04 maximum
Co	2.50 maximum	S	0.03 maximum

2. Thermophysical Properties

- a. Density - 0.323 lb/in³ A-9
- b. Melting Range - 2310-2450°F A-9
- c. Emissivity A-12

<u>Temp, °F</u>	<u>ε, Polished</u>	<u>ε, Stably Oxidized</u>
400	0.11	0.90
800	0.17	0.90
1200	0.22	0.90
1600	0.31	0.92
2000	----	0.96

- d. Thermal Conductivity A-9, 10
 10.8 BTU/ft. hr. °F at 1100°F
 15.1 BTU/ft. hr. °F at 1400°F
 16.2 BTU/ft. hr. °F at 1600°F
 17.4 BTU/ft. hr. °F at 2000°F
- e. Specific Heat A-11
 0.124 BTU/lb. °F at 1040°F
 0.133 BTU/lb. °F at 1290°F
 0.143 BTU/lb. °F at 1540°F
 0.152 BTU/lb. °F at 1790°F
 0.162 BTU/lb. °F at 2040°F
- f. Coefficient of Thermal Expansion A-9, 12
 $7.44 \times 10^{-6}/^{\circ}\text{F}$ 70-1000°F
 $7.73 \times 10^{-6}/^{\circ}\text{F}$ 70-1200°F
 $8.07 \times 10^{-6}/^{\circ}\text{F}$ 70-1500°F
 $8.20 \times 10^{-6}/^{\circ}\text{F}$ 70-1600°F
 $8.5 \times 10^{-6}/^{\circ}\text{F}$ 32-1800°F
 $9.6 \times 10^{-6}/^{\circ}\text{F}$ 70-2000°F

3. <u>Mechanical Properties</u>	A-13			
	<u>70°F</u>	<u>1000°F</u>	<u>1600°F</u>	<u>1800°F</u>
a. Modulus of Elasticity in Tension, psi $\times 10^{-6}$	29.8	24.8	19.5 (1500°F)	----
b. Tensile strength, psi $\times 10^{-3}$	121	99.3	56.4	31.7
c. Yield strength (0.2% offset) psi $\times 10^{-3}$	57.8	43.9	37.5	18.2
d. Elongation, % (in 2 in.)	47.5	52	47	49

D. URANIUM - 8% MOLYBDENUM

1. Composition, weight percent A-14

Depleted Uranium	92%
Molybdenum	8%

2. Thermophysical Properties
 - a. Density - 0.629 lb/in^3
 - b. Melting Range - $\sim 2190^\circ\text{F}$
 - c. Thermal Conductivity
 - $8.0 \text{ BTU/ft. hr. }^\circ\text{F at } 158^\circ\text{F}$
 - $12.0 \text{ BTU/ft. hr. }^\circ\text{F at } 440^\circ\text{F}$
 - $20.8 \text{ BTU/ft. hr. }^\circ\text{F at } 1160^\circ\text{F}$
 - $20.8 \text{ BTU/ft. hr. }^\circ\text{F at } 1300^\circ\text{F}$
 - d. Specific Heat
 - $0.040 \text{ BTU/lb. }^\circ\text{F at } 81^\circ\text{F}$
 - $0.040 \text{ BTU/lb. }^\circ\text{F at } 440^\circ\text{F}$
 - $0.060 \text{ BTU/lb. }^\circ\text{F at } 1160^\circ\text{F}$
 - $0.080 \text{ BTU/lb. }^\circ\text{F at } 1870^\circ\text{F}$
 - e. Coefficient of Thermal Expansion A-15
 - $7.7 \times 10^{-6}/^\circ\text{F} \quad (70-800^\circ\text{F})$
 - $8.4 \times 10^{-6}/^\circ\text{F} \quad (70-1000^\circ\text{F})$
 - $12.1 \times 10^{-6}/^\circ\text{F} \quad (70-1200^\circ\text{F})$
 - $13.4 \times 10^{-6}/^\circ\text{F} \quad (70-1400^\circ\text{F})$

3. Mechanical Properties A-14
 - a. Modulus of Elasticity - $12 \times 10^6 \text{ psi}$
 - b. Tensile Strength, "as cast" U-8 Mo A-16
 - $65.8 \times 10^3 \text{ psi at } 800^\circ\text{F}$
 - $56.3 \times 10^3 \text{ psi at } 1000^\circ\text{F}$
 - $24.8 \times 10^3 \text{ psi at } 1200^\circ\text{F}$
 - $16.9 \times 10^3 \text{ psi at } 1400^\circ\text{F}$

- c. Yield Strength (0.1% offset) A-14
115,000 - 140,000 psi
- d. Elongation A-16
5.3% at 800°F
10.7% at 1000°F
36.7% at 1200°F
25.3% at 1400°F

E. OXYGEN FREE (OF) COPPER

- 1. Composition A-13
99.95% Cu
- 2. Thermophysical Properties
 - a. Density - 0.323 lb./in³
 - b. Melting Point - 1981°F
 - c. Thermal Conductivity
226 Btu/ft hr °F at 68°F
200 Btu/ft hr °F at 800°F
 - d. Specific Heat
0.092 Btu/lb °F at 68°F
0.106 Btu/lb °F at 1100°F
 - e. Coefficient of Thermal Expansion
 $9.8 \times 10^{-6}/^{\circ}\text{F}$ (68-572°F)
- 3. Mechanical Properties
 - a. Modulus of Elasticity in Tension
 17×10^6 psi

- b. Tensile Strength
 - Annealed 32,000–35,000 psi
 - Hard 45,000–55,000 psi
- c. Yield Strength (0.5% ext.)
 - Annealed 10,000–11,000 psi
 - Hard 40,000–50,000 psi
- d. Elongation (in 2 in.)
 - Annealed 45–55%
 - Hard 6–20%

F. INCONEL 625

1. Composition, percent

Nickel	Balance	Manganese	0.50 maximum
Chromium	20.0–23.0	Silicon	0.50 maximum
Molybdenum	8.0–10.0	Aluminum	0.40 maximum
Iron	5.0 max.	Titanium	0.40 maximum
Columbium + Tantalum	3.15 – 4.15	Carbon	0.10 maximum
Cobalt	1.00 max	Sulfur	0.015 maximum

2. Thermophysical Properties

- a. Density – 0.305 lb/in³
- b. Melting Range – 2350–2460°F
- c. Thermal Conductivity
 - 9.1 Btu/ft hr °F at 800°F
 - 11 Btu/ft hr °F at 1200°F
 - 13 Btu/ft hr °F at 1600°F

- d. Specific Heat
0.098 Btu/lb °F
- e. Coefficient of Thermal Expansion
 - $7.6 \times 10^{-6}/^{\circ}\text{F}$ (70-800°F)
 - $8.2 \times 10^{-6}/^{\circ}\text{F}$ (70-1200°F)
 - $8.8 \times 10^{-6}/^{\circ}\text{F}$ (70-1600°F)

G. 304 STAINLESS STEEL

A-13

1. Composition, weight percent

Iron	65-74
Chromium	18-20
Nickel	8-12
Manganese	2 maximum
Silicon	1 maximum
Carbon	0.08 maximum
Sulphur	0.030 maximum

2. Thermophysical Properties

- a. Density - 0.29 lb/in^3
- b. Melting Range - 2550 - 2650°F
- c. Emissivity -
 - 0.2 unpolished
 - 0.85 oxidized and weathered
- d. Thermal Conductivity
 - 9.4 Btu/ft hr °F at 212°F
 - 9.8 Btu/ft hr °F at 400°F
 - 12.0 Btu/ft hr °F at 800°F
 - 14.3 Btu/ft hr °F at 1200°F

A-21

e. Specific Heat

A-21

0.120 Btu/lb °F at 100°F

0.129 Btu/lb °F at 400°F

0.135 Btu/lb °F at 600°F

0.138 Btu/lb °F at 700°F

f. Coefficient of Thermal Expansion

A-21

$9.0 \times 10^{-6}/^{\circ}\text{F}$ (68-200°F)

$9.4 \times 10^{-6}/^{\circ}\text{F}$ (68-400°F)

$9.8 \times 10^{-6}/^{\circ}\text{F}$ (68-600°F)

$10.0 \times 10^{-6}/^{\circ}\text{F}$ (68-800°F)

$10.1 \times 10^{-6}/^{\circ}\text{F}$ (68-1000°F)

$10.5 \times 10^{-6}/^{\circ}\text{F}$ (68-1200°F)

H. INCONEL 718

A-18

1. Composition, Percent

Nickel (+ cobalt)	50.00-55.00	Cobalt	1.00 Maximum
Chromium	17.00-21.00	Carbon	0.08 Maximum
Iron	Balance	Manganese	0.35 Maximum
Columbium (+ Tantalum)	4.75-5.50	Sulphur	0.015 Maximum
		Silicon	0.35 Maximum
Molybdenum	2.80-3.30	Phosphorous	0.015 Maximum
Titanium	0.65-1.15	Boron	0.006 Maximum
Aluminum	0.20-0.80	Copper	0.30 Maximum

2. Thermophysical Properties

a. Density - 0.297 lb/in^3

b. Melting Range - 2300-2437°F

- c. Thermal Conductivity
 - 10.3 Btu/ft hr °F at 800°F
 - 12.3 Btu/ft hr °F at 1200°F
 - 14.4 Btu/ft hr °F at 1600°F
- d. Specific Heat
 - 0.104 Btu/lb °F at 70°F
- e. Coefficient of Thermal Expansion
 - $8.1 \times 10^{-6}/^{\circ}\text{F}$ (70-800°F)
 - $8.6 \times 10^{-6}/^{\circ}\text{F}$ (70-1200°F)
 - $9.5 \times 10^{-6}/^{\circ}\text{F}$ (70-1600°F)

I. MIN-K 2002

A-19

1. Composition

Silica	75%
TiO ₂	16%
Microquartz fiber	4%
Astroquartz fiber	5%

2. Thermophysical Properties

- a. Density - 0.0116 lb/in³ A-20
- b. Thermal Conductivity, Btu/ft hr °F A-15

<u>Air-Filled</u>	<u>Krypton-Filled</u>	<u>Temperature</u>
0.020	0.017	400°F
0.022	0.018	600°F
0.025	0.019	800°F
0.029	0.021	1000°F
0.035	0.023	1200°F

c. Specific Heat in Air (Min-K 2000)

A-20

0.234 Btu/lb °F at 600°F

0.256 Btu/lb °F at 1000°F

0.264 Btu/lb °F at 1200°F

0.268 Btu/lb °F at 1400°F

0.276 Btu/lb °F at 1800°F

0.280 Btu/lb °F at 2000°F

J. 1100 ALUMINUM

A-13

1. Composition

99.0% aluminum minimum

2. Thermophysical Properties

a. Density - 0.098 lb/in³

b. Melting Range - 1190 - 1215°F

c. Emissivity in Air

0.030 at 77°F

Approximately 0.8 in fire conditions

d. Thermal Conductivity

128 Btu/ft hr °F at 77°F

e. Specific Heat

0.214 Btu/lb °F at 68°F

0.22 Btu/lb °F at 212°F

f. Coefficient of Thermal Expansion

$13.1 \times 10^{-6}/^{\circ}\text{F}$ (68-212°F)

$14.2 \times 10^{-6}/^{\circ}\text{F}$ (68-572°F)

$15.2 \times 10^{-6}/^{\circ}\text{F}$ (68-932°F)

K. 2024 ALUMINUM

1. Composition, percent A-13

Aluminum	92.4 - 94.7	Iron	0.50 max.
Copper	3.8 - 4.9	Silicon	0.50 max.
Magnesium	1.2 - 1.8	Zinc	0.25 max.
		Chromium	0.05 max.
Manganese	0.3 - 0.09	Other	0.15 max.
2. Thermophysical Properties
 - a. Density - 0.100 lb/in³ A-13
 - b. Melting Range - 935-1180°F A-13
 - c. Emissivity - A-22
 - in air - 0.035-0.07 at 68°F
 - d. Thermal Conductivity A-22
 - 109 Btu/ft hr °F at 77°F
 - 110 Btu/ft hr °F at 200°F
 - 114 Btu/ft hr °F at 400°F
 - 109 Btu/ft hr °F at 600°F
 - 100 Btu/ft hr °F at 800°F
 - e. Specific Heat A-22
 - 0.20 Btu/lb °F at 70°F
 - 0.22 Btu/lb °F at 200°F
 - 0.23 Btu/lb °F at 400°F
 - 0.25 Btu/lb °F at 600°F
 - 0.28 Btu/lb °F at 800°F
 - f. Coefficient of Thermal Expansion A-13
 - $13.1 \times 10^{-6}/^{\circ}\text{F}$ (68-212°F)
 - $13.9 \times 10^{-6}/^{\circ}\text{F}$ (68-572°F)

L. TANTALUM

A-23

1. Thermophysical Properties

- a. Density - 0.60 lb/in^3
- b. Melting Point - 5425°F
- c. Thermal Conductivity
 - $31.1 \text{ Btu/ft hr } ^\circ\text{F at } 68^\circ\text{F}$
 - $38.8 \text{ Btu/ft hr } ^\circ\text{F at } 1055^\circ\text{F}$
 - $40.9 \text{ Btu/ft hr } ^\circ\text{F at } 1523^\circ\text{F}$
 - $42.9 \text{ Btu/ft hr } ^\circ\text{F at } 2022^\circ\text{F}$
 - $45.0 \text{ Btu/ft hr } ^\circ\text{F at } 2580^\circ\text{F}$
- d. Specific Heat
 - $0.0340 \text{ Btu/lb } ^\circ\text{F at } 32^\circ\text{F}$
 - $0.0350 \text{ Btu/lb } ^\circ\text{F at } 441^\circ\text{F}$
 - $0.0365 \text{ Btu/lb } ^\circ\text{F at } 1341^\circ\text{F}$
 - $0.0385 \text{ Btu/lb } ^\circ\text{F at } 2241^\circ\text{F}$
 - $0.041 \text{ Btu/lb } ^\circ\text{F at } 3141^\circ\text{F}$
- e. Coefficient of Thermal Expansion
 - $3.67 \times 10^{-6}/^\circ\text{F (68-932}^\circ\text{F)}$
 - $4.45 \times 10^{-6}/^\circ\text{F (68-2732}^\circ\text{F)}$

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APPENDIX B

SAFETY THERMAL MODEL DESCRIPTION AND FIRE ENVIRONMENT

A. INTRODUCTION AND BACKGROUND

Nuclear safety evaluations of the SNAP-23A generator and heat source were implemented with the aid of several analytical thermal models. Since each of these tools has been used extensively on analyses performed to date, they have been summarized for reference as follows:

1) Initial Overall Safety Model. This was a 72-node thermal model developed early in the program to provide transient and steady-state thermal data related to safety. With the exception of ground burial analyses, the data obtained with this model have been largely updated using the current thermal model.

2) Simplified 25-Node Model. The complexity of the early 72-node model in the fuel capsule region resulted in appreciable computer time. A smaller, 25-node model was developed to permit economical parametric studies.

3) Current 65-Node Safety Model. The present analytical thermal model was developed as a result of all the preliminary analyses with the 72-node and 25-node models. It avoids the computer time limitations of the earlier models and provides more accurate temperature profiles in critical regions of the generator.

B. COMPUTER CODE AND CURRENT ANALYTICAL THERMAL MODEL DESCRIPTION

1. Thermal Analyzer Code

The Hittman Associates TAP-4 code was used as the basic network from which the various thermal models were prepared. TAP-4 is general in construction, using a network of conductors and capacitors to describe a heat transfer nodal network with transient properties solved through simultaneous finite difference equations. Solution of a particular problem requires definition of nodal arrangement and the designation of built-in arithmetic functions and subroutines that will enable solution of the generalized code logic.

2. Analytical Model

The 65-node safety model used to perform thermal safety analyses is shown in Figure B-1. The basic features of this model are discussed below.

a) Detail in the area of the capsule was minimized to improve running time. However, the fuel, liner, capsule, diffusion barrier and associated gaps were taken into account in calculating the fuel temperature. Figure B-2 shows the detail of gaps and wall thickness used in calculating the connectors in this area.

The connectors through the gaps are composed of both radiation and conduction through helium gas. The radiation and conduction heat transfers in these areas were assumed to be in the form:

$$Q_{\text{rad}} = \sigma A \epsilon_{\text{eff}} (T_2^4 - T_1^4)$$

and

$$Q_{\text{cond}} = k_{\text{He}} \frac{2 \pi L}{\ln r_2 / r_1} (T_2 - T_1)$$

where

T_2 is the associated temperature of the inner wall

T_1 is the associated temperature of the outer wall

The effective connector 1 (shown in Figure B-2) is given by the equation

$$\begin{aligned} \frac{1}{Y_{\text{eff}}} = & \frac{1}{Y_{\text{fuel}}} + \frac{1}{Y_{\text{rad}} + Y_{\text{cond}}} + \frac{1}{Y_{\text{liner}}} \\ & + \frac{1}{Y_{\text{rad}} + Y_{\text{cond}}} + \frac{1}{Y_{\text{capsule}}} \end{aligned} \quad (\text{B-1})$$

The radiation heat transfer is a function of the fourth power of temperature. Temperatures from previous thermal model results (from the 72-node model which has detailed connectors in this region) were used to linearize the radiation terms so that an effective connector No. 1 could then be calculated. A similar approach was used on connector No. 2. The assumption

Figure B-1. Safety Thermal Model

Figure B-2. Basis of Thermal Model in Fuel Capsule Region

used to linearize the fourth power term is only accurate for moderate changes in temperature. If large changes in temperature occur, error is introduced into the effective connector. In most cases analyzed by these models, large changes in fuel, capsule, and inner shield temperature do not occur.

b) The connectors in the fuel region were calculated assuming uniform heat generation throughout the fuel volume. By solving the differential equation (see Figure B-3 for details):

$$\frac{d^2 t}{dr^2} + \frac{1}{r} \frac{dt}{dr} + \frac{q'''}{k_f} = 0 \quad (\text{B-2})$$

and by using the boundary condition:

$$\left. \frac{dt}{dr} \right|_{r=0} = 0 \text{ and } t_{r=0} = t_o$$

The temperature, as a function of r , can be shown to be in the form

$$t(r) = -q''' \frac{r^2}{4k_f} + t_o \quad (\text{B-3})$$

where

t_o is the centerline temperature

q''' is the volumetric heat generation rate

k_f is the thermal conductivity

r is the radius

The equation used for conduction heat transfer in TAP-4 is

$$T_1 = \frac{Q_1 + Y_{1,2} T_2}{Y_{1,2}} ;$$

therefore

$$T_1 - T_2 = Q_1 / Y_{1,2} \quad (\text{B-4})$$

Figure B-3. Fuel Region Notation

where

T_1 and T_2 are the nodal temperatures

Q_1 is the heat flow between nodes 1 and 2

$Y_{1,2}$ is the connector between nodes 1 and 2

Using the equation of the temperature distribution in a solid cylinder with internal heat generation,

$$T_1 - T_2 = q''' \frac{R_2^2}{4k_f} \quad (B-5)$$

where

$$q''' = \frac{Q}{\pi R_1^2 L}$$

R_2 is radius node 2

R_1 is radius node 1

Equating Equations B-4 and B-5, the connector between two nodes in the fuel can be computed by

$$Q/Y_{1,2} = \frac{Q}{\pi R_1^2 L} \frac{R_2^2}{4k_f}$$

$$Y_{1,2} = 4\pi L k_f \left[\frac{R_1^2}{R_2^2} \right] \quad (B-6)$$

Using the above equation, the connector in the fuel region gives more realistic fuel center-line temperatures.

c) The biological shield and insulation are divided into three layers each. The number of nodes provides a reasonable balance between accuracy and computer time usage.

d) The effect of the perforated cover around the radiator is accounted for in a realistic fashion. Nodes 62 and 64, representing the cover, are connected to the ambient node 72. The connector between ambient temperature and the fins (nodes 63, 64, and 65) also includes convection and radiation heat transfer.

e) Nodal detail in the heat accumulator neck was minimized. Results from the previous 72-node model showed very little temperature gradient in the neck because of the high thermal conductivity of copper.

f) The nodal network in the heat source is arranged so that it can be readily removed from the complete system. This was done to provide a capability to evaluate the heat source in ambient and transient fire conditions. The nodes on the outer heat source can easily be connected to an ambient node, requiring an appropriate change in the connector. This connector includes both radiation and convection heat transfer as a function of the surface-to-ambient delta temperature. The natural convection, h , was calculated using the relationship

$$h = .24 \left(\frac{\Delta T}{L} \right)^{0.25} \quad (B-7)$$

where

ΔT is the temperature difference between the surface and ambient

L is the characteristic length for the node

During a fire transient, the heat input is assumed to be radiation. A surface emissivity of 0.8 and a fire emissivity of 1.0 are assumed.

Before the fire, the surface emissivity is assumed to be 0.5.

During and after the fire, the surface is assumed to have an emissivity of 0.8.

g) Paths for miscellaneous heat losses are incorporated in distinct program logic for evaluation of parasitic effects.

h) The converter model includes nodes which have been placed in the hot frame, T/E hot end, T/E cold end, and cold frame. This four-node simulation can be replaced by a simplified two-node model, with nodes in only the hot and cold frames. The two-node model was used to evaluate temperature conditions where the total converter conductance must be determined as a function of temperature and then inserted into the generator thermal model as tabular functions (e.g., to support the converter fire test).

The thermal model consists of 65 nodes and approximately 140 connectors. The number of nodes was selected to provide an optimum balance between accuracy and computer time. Where appropriate, thermal properties were input to the model as functions of temperature. Material thermal properties are described in Appendix A.

Transient safety analysis of the system requires a detailed representation of the radiator, particularly in the evaluation of system response to a fire. These details are optionally incorporated into the complete analytical safety model of the system. Several effects have been accounted for by the radiator fin model:

- a) Heat of fusion of the aluminum
- b) Reduction in fin area as fins melt from tip to root
- c) Radiation from the steel enclosing case to five points on each fin

The model used to perform the analysis of both the 1475°F and the 1850°F fire is shown in Figure B-4. The fins are divided into five nodes. Three nodes are then connected to each other, to the upper section of the perforated cover, and to the side section of the perforated cover.

All surfaces involved are assumed to have an emissivity of 0.8. Since the analytical representation of a fire is by radiation only (a function of area, ϵ_{eff} , and T^4), the reduction in area due to melting significantly affects the heat input. The fins see the fire through the holes in the perforated cover so that radiation heat input is of the form:

$$Q_{\text{input}} = F A \epsilon_{\text{eff}} \sigma (T_2^4 - T_1^4)$$

where F is the view factor between the fins and the fire. The cover quickly reaches a temperature close to the fire temperature (and then radiates to the fins at this temperature). This is why the five fin nodes are connected to the upper and lower perforated cover.

As the fin outer node reaches the melting point of aluminum, the net heat input to the node is summed until the heat of fusion of the node is accounted for. The node is then disconnected from the nodal map and the nodal map rearranged so that the second node is connected to the ambient node. This process is repeated for each node that is melted.

Figure B-4. Detail of Radiator Portion of Thermal Model

If the fire results in complete melting of the fins, the heat input is assumed to feed directly into the cold frame on the converter. The heat of fusion for the aluminum cold frame is accounted for by the same method used with the fins. The temperatures through the T/E converter are allowed to rise and have an upper bound governed by the radiation heat input from the fire and the diffusivity of the SNAP-23A system.

3. Earth Burial Analyses

Earth burial results were performed with the initial safety model and the simplified 25-node model, using the nodal network shown in Figure B-5. Fourteen earth nodes are used to provide a temperature distribution around and below the upright system. Steady-state earth burial results were obtained by using the 72-node model with the cylindrical earth nodes. The size of the earth nodes was varied according to the burial depth of the generator.

4. Heat Source Analysis

Transient thermal analysis of the heat source assembly was made by extracting a portion of the generator model already discussed and revising the boundary conditions. Thus, all nodes, admittances, and capacitances within the heat source remained unchanged. Thermal paths from the heat source to the insulation and converter were revised from conduction to radiation, when appropriate. Capability existed within the model to vary the exterior temperature as a function of time.

C. FIRE ENVIRONMENT USED WITH ANALYTICAL MODEL

1. 1475°F One-Half Hour Fire (Standard Transportation Fire)

The 1475°F fire was studied because it represents the hypothetical accident environment specified by the AEC (Reference B-2), DOT (Reference B-3), and IAEA (Reference B-4). Surviving this fire environment in the appropriate packaging for transport implies that the SNAP-23A system can meet the nominal transport requirements for thermal environments. The fire has a duration of one-half hour and an emissivity of 0.9. The package emissivity is taken as 0.8. Survival implies that cooling of the package is not initiated until at least 3 hours after the fire is ended.

Figure B-5. SNAP-23A Ground Burial Model

The applicability of the 1475°F environment is amply demonstrated by the data presented in References B-5 and B-6.

2. Standard Building Fire

Under storage conditions, the SNAP-23A system or the heat source alone could, hypothetically at least, be exposed to a fire of different characteristics than the 1475°F, 30-minute fire of hydrocarbon fuel. For a building with normal combustible loading, the fuel is assumed to be cellulose-base materials and the resulting time-temperature profile (Reference B-7) is as follows:

<u>Time (hours)</u>	<u>Temperature (°F)</u>
0	0
0.25	1420
0.50	1560
0.75	1640
1.00	1700

Reference B-7 does not specify the emissivity values to be used in conjunction with the standard fire. SNAP-23A analyses have considered the fire to have an effective emissivity of one and the package being heated to have an emissivity appropriate to its surface condition. This latter value has been conservatively taken as 0.8 to represent an oxidized surface.

3. 1850°F One-Half Hour Fire

Considerable evidence exists that certain types of fires that could be encountered during a transportation accident achieve peak temperatures greater than 1475°F. Reference B-7 reveals that measurements made for various types of petroleum fuel yielded mean flame temperatures ranging from 1700°F to 2190°F. These experiments were for relatively small pools of fuel. More recently, Sandia Corporation (Reference B-5) conducted tests with JP-4 aviation-type fuel. On the basis of results from these tests, Reference B-8 concludes that a large free burning fire can be numerically represented by an 1850°F black body source. A relatively large pool of fuel would be required to achieve a fire duration of one-half hour. Russian experiments (Reference B-7) indicated that for pool diameters greater than 1 meter,

the rate of burning for gasoline or kerosene was about 4 millimeters per minute and independent of diameter. Fire temperature was on the order of 1475°F . Furthermore, for large pools flame height was about 1.7 times pool diameter. The Russian experiments used the same scale as for a transportation accident and, therefore, give a fair indication of what might be expected in practice. On the basis of available data, an 1850°F fire was taken as the maximum credible thermal environment to which the SNAP-23A system can be exposed. The 1850°F value is undoubtedly conservative since it is an upper bound on reported data for fires that are large enough and of sufficient duration to be considered.

REFERENCES

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- B-3 Federal Register, Vol. 33, No. 194, October 1968, "Department of Transportation, Hazardous Materials Regulations Board."
- B-4 Regulations For Safe Transport of Radioactive Materials, IAEA Safety Series No. 6, 1967.
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- B-6 Sousselier, Y., "Fire Tests of Packaging for Radioactive Materials," Presented at International Symposium for Packaging and Transportation of Radioactive Materials, SC-RR-65-98, January 1965.
- B-7 Messenger, L. M., Fairbairn, A. "The Transport of Radioactive Materials Interim Recommendations for the Application of Environmental Tests to the Approval of Packaging," UKAEA, 1965.
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APPENDIX C

FUEL CAPSULE STRESS ANALYSIS

A number of stress analyses have been performed during the SNAP-23A fuel capsule design effort to verify that the capsule will survive the 9040 psi hydrostatic pressure that results from capsule burial in 20,000 feet of seawater, as required by the system safety design criteria. A preliminary stress analysis on the fuel capsule (Reference C-1) indicated compliance with the design criteria. The analysis consisted of an approximate manual solution to arrive at the stress intensity of the capsule closure weld. The outer cylindrical shell and cap thickness, and weld penetration requirements were sized from the preliminary analysis.

An improved version of the Westinghouse-developed FEATS computer program was used for the current analysis since its treatment of plastic strains is more accurate than the previous code used. The FEATS finite element code can be used to analyze plane stress or axisymmetric structures subjected to mechanical or thermal loads. In the finite element method, the continuous structure is replaced by a system of elements which are interconnected at joints or nodal points. Equilibrium equations, in terms of unknown nodal point displacements, are used in the code to arrive at the solution.

The detailed capsule stress model developed is illustrated in Figure C-1. A series of exploded sections are provided for clarity. The major emphasis in this computer model has been concentrated in the root of the end cap weld. In this particular region, grid (element) spacings of 0.001 inch width, gradually increasing in size with increasing distance from the root, have been utilized. The specific material properties used are shown in Table C-1.

Figure C-1. Capsule Stress Model

TABLE C-1
CAPSULE STRESS MODEL MATERIALS PROPERTIES

<u>Material</u>	<u>Elastic Modulus (psi)</u>	<u>Poisson's Ratio</u>	<u>Plastic/Elastic Modulus Ratio</u>	<u>Yield Strength (psi)</u>
Hastelloy-C	30×10^6	0.3	0.0046	52,000
Weld	30×10^6	0.3	0.0087	50,000
Fuel	10×10^6	0.3	0.005	20,000

The effective strains developed in more critical elements (at the weld root) are summarized in Table C-2. It is apparent from the data presented that the maximum calculated effective strain is compressive and less than two percent at the base of the weld. From this detailed computer analysis, it has been concluded that the reference capsule design will meet the 10,000 psi hydrostatic pressure requirement. Hydrostatic tests on SNAP-23A fuel capsules containing a fuel simulant have experimentally verified this safety design requirement, as described in Section VIII B.

TABLE C-2
CALCULATED EFFECTIVE STRAINS IN CAPSULE WELD REGION

<u>Element Number</u>	<u>Effective Strain (Percent)</u>
119	0.61
120	0.68
127	0.62
128	0.73
136	0.84
137	0.81
138	0.84
139	0.80
147	1.95
148	1.45
155	1.48
156	1.36

REFERENCES

- C-1 WANL-3800-001, SNAP-23A Quarterly Progress Report, March-May 1967.

APPENDIX D

ATMOSPHERIC FUEL RELEASE

A. INTRODUCTION

This appendix reports the results of a ground level release evaluation. It must be noted that only a sustained, very high temperature environment can both breach the capsule and cause fuel vaporization. This environment is not credible for the SNAP-23A System. (The environments for which a release could be postulated include either a high-yield explosion, causing partial vaporization of all components or a large fire lasting perhaps 20 or more hours with sustained high temperatures.) The brief analyses presented here are therefore academic and are only intended to give a complete documentation of SNAP-23A nuclear safety.

B. SOURCE TERM

Although the environment which could lead to an airborne fuel release is considered not credible for the SNAP-23A system, the approach taken in this appendix has been to define the source term consistent with a design thermal environment. Therefore, the fuel is assumed to be directly exposed to an 1850°F temperature.

The release rate is based on the estimated rate of vaporization of SrTiO_3 at 1850°F. The vapor pressure of SrTiO_3 is not known as a function of temperature; however, it is thought that the vapor pressure curve for SrO obtained from Reference D-1 can be taken as a conservative estimate of that for SrTiO_3 (Reference D-2). This is supported by limited data from Reference D-3, which gives a vapor pressure of 3×10^{-7} atm for SrTiO_3 at 2966°F. Figure D-1 shows the vapor pressure curve of SrO and the data point for SrTiO_3 . The value for SrTiO_3 is about 35 percent lower than that for SrO , indicating that the SrO data provides a conservative estimate of the SrTiO_3 vapor pressure.

The mass loss rate is related to the vapor pressure by the equation:

$$R = \frac{P}{\sqrt{2\pi m k T}} \quad (D-1)$$

Figure D-1. Estimated Vapor Pressure of Strontium Titanate

where

R is the mass loss rate, molecules/cm²-sec

m is the mass of molecule, gm/molecule

k is the Boltzmann's constant, gm-m²/sec²-molecule-°K

T is the absolute temperature, °K

P is the vapor pressure, gm/m-sec²

Equation D-1 gives the mass loss rate in a vacuum. At higher pressure some of the molecules that leave the surface are replaced by vaporized molecules which strike the surface and recombine. This has the effect of reducing the mass loss rate. For this analysis it has been conservatively assumed that all of the vaporized molecules are immediately swept away so that they cannot recombine with the surface. Even in a violent fire, this assumption could cause the mass loss rate in Equation D-1 to be conservative by as much as a factor of 100.

Equation D-1 can be written as

$$R = \left(4.31 \times 10^{-11} \frac{\text{gm}}{\text{cm}^2 \text{ sec}} \right) P \sqrt{\frac{M}{T}} \quad (\text{D-2})$$

where

P is the vapor pressure in atmospheres

M is the molecular weight, amu

T is the temperature, °K

Substituting values for SrTiO₃ in an 1850°F fire,

$$P = 1.085 \times 10^{-6} \text{ atmospheres}$$

$$M = 186 \text{ amu}$$

$$T = 1850^\circ\text{F} = 1283^\circ\text{K}$$

Then,

$$R = 1.83 \times 10^{-12} \frac{\text{gm}}{\text{cm}^2 \text{ sec}}$$

It was assumed that about one-half of the total fuel pellet surface area is available for vaporization at a given time due to the initial stacking of the pellets and the design strength of the containment. This results in a release rate of

$$Q = (1.83 \times 10^{-12} \frac{\text{gm}}{\text{cm}^2 \text{ sec}}) (600 \text{ cm}^2) (34.5 \frac{\text{ci}}{\text{gm}})$$

$$Q = 3.78 \times 10^{-8} \text{ ci/sec}$$

All fuel released is assumed to be an inhalable aerosol with the particulate having aerodynamic diameters of less than 10 microns.

It is thought that the vaporized activity would remain in the form of SrTiO_3 , which is nearly insoluble. However, due to the lack of information on the form of vaporized SrTiO_3 , this analysis was performed for both soluble and insoluble Sr-90.

C. DISPERSION MODEL

The mathematical model used to study the release of activity to the atmosphere is taken from Reference D-4 and is based on the model suggested by Pasquill and modified by Gifford. It gives the concentration at some point downwind from a continuous source, assuming no deposition or reaction at the earth's surface. For this analysis the concentration was calculated only along the plumb centerline for a ground level release. The equation from Reference D-4 is then

$$A(x, z) = \left(\frac{Q}{\pi \sigma_y \sigma_z \bar{u}} \right) \exp \left[\frac{-z^2}{2 \sigma_z^2} \right] \quad (\text{D-3})$$

where

A is the activity concentration at height z and downwind distance x

Q is the activity release rate

$\bar{u}$ is the mean wind speed

σ_y, σ_z is the standard deviation of plumb concentration distribution in the horizontal and vertical directions

z is the height of the receptor

This is the same as Sutton's equation (Reference D-5), where σ_y and σ_z are related to Sutton's C_y and C_z through equations of the form

$$\sigma_y = \frac{1}{\sqrt{2}} (C_y)(x)^{\frac{2-n}{2}}$$

where n is a stability index, varying between zero and one.

Since the exact site of hypothetical ground release is unknown, the probable meteorologic conditions cannot be predicted. Newcombe (Reference D-6) points out that the poorest dispersion of nuclear material occurs when an inversion (stable or Pasquill's stability category class F) condition exists just above ground level. Also, Reference D-7 presents isopleth calculations for stable, neutral, and unstable conditions. The stable condition produces the largest isopleths, due to the limited vertical and horizontal dispersion of activity. Thus, a stable condition constitutes a worst case and is assumed for this analysis. It was assumed that a person remains in a fixed position downwind of the source for 30 minutes. Values of σ_y and σ_z as a function of downwind distance x for class F stability were taken from Reference D-4. According to Reference D-4, these values apply only for open, level country and overestimate the concentrations for other areas. Also, the empirical values of σ_y and σ_z are representative of a sampling time of about 10 minutes. For the 30-minute sampling time considered here, concentrations would most likely be smaller than those given by Equation D-3 because of increased meander in the wind direction.

Calculations were made for $z = 1.675$ m (the approximate height of a man) and for two wind speeds, 1.0 m/sec and 5.0 m/sec. These wind speeds are typical for inversion conditions. Table D-1 presents the parametric term $\frac{A_u}{Q}(x)$ and the activity concentration $A(x)$ for the assumed release rate and wind speeds.

The activity inhaled by a person on the plume centerline is $I = A B t$, where

I is the activity inhaled

A is the activity concentration

B is the breathing rate = $347.0 \frac{\text{cc}}{\text{sec}}$ for a standard man (Reference D-6)

t is the exposure time = 30 minutes

TABLE D-1
ACTIVITY CONCENTRATION, $A(X)$, AND INHALED ACTIVITY, $I(X)$, FOR WIND SPEEDS, U , OF
1 METER/SEC AND 5 METERS/SEC

Downwind Distance x (meters)	$\frac{Au}{Q}$ $\left[\frac{(\text{ci/m}) (\text{m/sec})}{(\text{ci/sec})} \right]$	$u = 1 \text{ meter/sec}$		$u = 5 \text{ meter/sec}$	
		$A(\text{ci/m}^3)$	$I(\text{ci})$	$A(\text{ci/m}^3)$	$I(\text{ci})$
10^2	2.68×10^{-2}	1.12×10^{-9}	7.05×10^{-10}	2.25×10^{-10}	1.40×10^{-10}
2×10^2	9.60×10^{-3}	3.64×10^{-10}	2.28×10^{-10}	7.30×10^{-11}	4.56×10^{-11}
5×10^2	2.04×10^{-3}	7.70×10^{-11}	4.81×10^{-11}	1.54×10^{-11}	9.60×10^{-12}
10^3	6.64×10^{-4}	2.50×10^{-11}	1.56×10^{-11}	5.00×10^{-12}	3.12×10^{-12}
2×10^3	2.32×10^{-4}	8.80×10^{-12}	5.50×10^{-12}	1.76×10^{-12}	1.10×10^{-12}
5×10^3	6.25×10^{-5}	2.36×10^{-12}	1.47×10^{-12}	4.72×10^{-13}	2.95×10^{-13}
10^4	2.52×10^{-5}	9.55×10^{-13}	5.95×10^{-13}	1.91×10^{-13}	1.19×10^{-13}
2×10^4	1.06×10^{-6}	4.02×10^{-13}	2.51×10^{-13}	8.05×10^{-14}	5.05×10^{-14}
5×10^4	3.57×10^{-6}	1.35×10^{-13}	8.45×10^{-14}	2.70×10^{-14}	1.69×10^{-14}
10^5	1.71×10^{-6}	6.50×10^{-14}	4.06×10^{-14}	1.30×10^{-14}	8.10×10^{-15}

The quantity $I(x)$ is also shown in Table D-1.

D. RADIOLOGICAL CONSEQUENCE ON MAN

The consequences of inhaling a given amount of activity I can be expressed in terms of dose to the critical organ by means of a simple body model. The hazard severity can then be estimated by comparing the calculated dose to the allowable dose for persons in unrestricted areas.

The model used here is based on ICRP data given in Reference D-8 for particles of unspecified size, shape, and density. The distribution of particulates is assumed to be as shown in Table D-2 (from Reference D-8).

TABLE D-2
DISTRIBUTION OF PARTICULATES IN RESPIRATORY
TRACT OF STANDARD MAN

	<u>Readily Soluble Compounds</u>	<u>Other Compounds</u>
Exhaled	25%	25%
Deposited in Upper Respiratory Passages and Subsequently Swallowed	50%	50%
Deposited in the Lungs (Lower Respiratory Passages)	25% (this is taken up into the body)	25%

Of the 25 percent of "Other Compounds" which go to the lower respiratory passages of the lungs, half (i.e., 12.5 percent of amount inhaled) will be eliminated from the lungs and swallowed within the first 24 hours. The other half will go from the lungs into body fluids with an effective biological half-life of 120 days.

A simple body model has been defined by the following assumptions:

- 1) All inhaled activity swallowed takes 24 hours to clear the lungs ie., it remains in the lung for an average of 12 hours.
- 2) All activity which goes from the lungs to body fluids goes to the blood, then to the critical organ according to the fraction given in Reference D-4.
- 3) Soluble activity from the lungs of GI tract is deposited in the bone via the blood in a time period which is negligible (no time constant required).
- 4) Activity passing through the gastro-intestinal tract is subject to blood and organ uptake according to the fractions given in Reference D-4.

The body model assumptions are very conservative. For instance, soluble activity will take less than 24 hours to clear the lung; a negligible amount of insoluble activity would pass from the GI tract to the blood, etc. However, the resulting model seems adequate to assess the airborne release hazard for the SNAP-23A System.

Figure D-2 shows the model, including fractions between organs, residence times in each organ, and paths which result in a significant dose to an organ. It has been assumed that the GI tract receives a dose from the activity passing through the tract and also from the activity which is absorbed into the blood from the GI tract. Thus, some of the activity which enters the GI tract is considered twice on leaving the organ.

Given the data in Figure D-2, the calculation of burden and dose as functions of time and inhaled activity I is straightforward. The rate of burden accumulation for each organ that receives a significant dose is

$$\dot{q}_{UL} = -f_2 I \frac{1}{t_1}$$

Upper lung, linear retention
 $0 < t < t_1$

$$\dot{q}_{LL} = -q_{LL} \lambda_L$$

Lower lung, exponential retention

$$\lambda_L = \frac{0.693}{(T_{1/2})_L} = \frac{0.693}{120 \text{ days}} = 5.78 \times 10^{-3} \text{ day}^{-1}$$

CONSTANTS

Path	Fractions	Soluble Sr-90	Insoluble Sr-90
Upper Lung → Blood	f_1	0.25	0.0
Upper Lung → GI	f_2	0.50	0.625
Upper Lung → Lower Lung	f_3	0.0	0.125
GI → Blood	f_4	0.3	0.3
GI Passage	f_5	1.0	1.0
Blood → Bone	f_6	0.3	0.3

Figure D-2. Body Model for Airborne Release Analysis

$$\dot{q}_{GI} = 0$$

GI tract, constant burden for 31 hours

$$\dot{q}_B = -q_B \lambda_B + f_6 q_{LL} \lambda_L$$

Bone, exponential retention

$$\lambda_B = \frac{0.693}{6.4 \times 10^3 \text{ day}} = 1.08 \times 10^{-4} \text{ day}^{-1}$$

Integrating and substituting the initial conditions for each organ, the equations for $q(I, t)$ are

$$q_{UL} = f_2 I \left(1 - \frac{t}{t_1}\right), \quad 0 < t < t_1 = 24 \text{ hours} \quad (D-4)$$

$$= 0 \quad \text{for } t > 24 \text{ hours}$$

$$q_{LL} = f_3 I e^{-\lambda_L t} \quad (D-5)$$

$$q_{GI} = f_5 f_2 I, \quad 0 < t < t_2 = 31 \text{ hours} \quad (D-6)$$

$$= 0, \quad t > t_2$$

$$q_B = f_6 f_3 I \left(\frac{\lambda_L}{\lambda_B - \lambda_L} \right) \left(e^{-\lambda_L t} - e^{-\lambda_B t} \right) + f_6 (f_1 + f_4 f_2) I e^{-\lambda_B t} \quad (D-7)$$

As mentioned, the 31 hour passage time through the GI tract is negligible for bone dose calculations. Thus, the bone receives immediate burdens from the GI-blood and upper lung-blood paths. This is seen by setting $t = 0$ in the last equation.

The dose rates are given by equations of the form

$$\dot{D}_i(I, t) = \frac{E_i}{M_i} q_i(I, t)$$

where

$\dot{D}_i(l, t)$ is the dose rate to organ (i) as a function of inhaled activity and time

M_i is the mass of organ (i)

$q_i(l, t)$ is the burden in organ (i) as a function of inhaled activity and time

$E_i = \sum [EF(RBE) n]$, effective energy delivered to organ (i) per disintegration

where

E is the average energy per disintegration

F is the fraction delivered to the critical organ

RBE is the relative biological effectiveness

n is the accounts for heterogeneous distribution in the bone

m is the mass of critical organ in grams

Values are listed in Reference D-8.

Then, the doses are

$$D_i(l, t) = \frac{E_i}{M_i} \int_0^t q_i(l, t) dt,$$

assuming the organ mass is constant over the integral time. The doses resulting from the burdens in Equations D-4 through D-7 are

$$D_{ul}(l, t) = \frac{1}{2} \frac{E_L}{M_L} f_2 I(24 \text{ hr}) = \frac{E_L}{M_L} f_2 I(12 \text{ hr}), t > 24 \text{ hr} \quad (D-8)$$

$$D_{LL}(l, t) = \frac{E_L}{M_L} \frac{f_3 I}{\lambda_L} \left(1 - e^{-\lambda_L t} \right) \quad (D-9)$$

$$D_{GI}(l, t) = \frac{E_{GI}}{M_{GI}} f_5 f_2 I(31 \text{ hr}), t > 31 \text{ hours} \quad (D-10)$$

$$D_B(I, t) = \frac{E_B}{M_B} f_6 I \left\{ (1 - e^{-\lambda_L t}) \left(\frac{f_3}{\lambda_B - \lambda_L} \right) + (1 - e^{-\lambda_B t}) \left(\frac{1}{\lambda_B} \right) \right. \\ \left. \left[(f_1 + f_4 f_2) - \frac{f_3 \lambda_L}{(\lambda_B - \lambda_L)} \right] \right\} \quad (D-11)$$

The values used in the above equations are (Reference D-8) as follows:

<u>Lung</u>	$E_L = 5.62 \times 10^7 \frac{\text{rem gm (of organ)}}{\text{ci day}}$
	$M_L = 10^3 \text{ gm}$
	$\lambda_L = 5.78 \times 10^{-3} \text{ day}^{-1}$
<u>GI tract</u>	$E_{GI} = 4.55 \times 10^7 \frac{\text{rem gm (of organ)}}{\text{ci day}}$
	$M_{GI} = 314 \text{ gm (average of four regions, weighted by residence time in each region)}$
<u>Bone</u>	$E_B = 2.81 \times 10^8 \frac{\text{rem gm (of organ)}}{\text{ci day}}$
	$M_B = 10^4 \text{ gm}$
	$\lambda_B = 1.08 \times 10^{-4} \text{ day}^{-1}$

The fractions f_1 through f_6 are tabulated in Figure D-2 for both soluble and insoluble Sr-90.

E. RESULTS

Equations D-8 through D-11 were used to find the critical organ dose, following inhalation of either soluble or insoluble Sr-90. Calculations of the dose to both the lung

Figure D-3. 50-Year Dose for 30 Minute Exposure to Airborne Release of Soluble and Insoluble Strontium-90 as a Function of Distance Downwind from the Source. (Wind Speed = 1.0 m/sec)

and bone were made for the first 50 years (lifetime) following exposure. As expected, the bone is the critical organ. The dose results are shown in Figure D-3 as a function of downwind distance of the receptor.

The values in Figure D-3 can be compared to the allowable critical organ doses to persons in unrestricted areas, which is 25 rem for a 50-year period (Reference D-9). It can be seen that this hazard results in doses which are well below the allowable limits, assuming the person is more than 100 meters from the source.

The analysis was not extended to distances less than 100 meters because:

- a) The Pasquill atmospheric dispersion coefficients for distances less than 100 meters involves a questionable extrapolation of the data.
- b) The dispersion equation is not valid close to the source for the assumed release environment, ie., a violent fire.

D. CONCLUSIONS

This analysis is based on a conservatively estimated source term and on a simple, but conservative body model. It shows that an airborne fuel release from the SNAP-23A system will not constitute a hazard to a person standing 100 meters or more downwind from the source.

No analysis has been made for persons less than 100 meters downwind, since the dispersion equation is not valid close to a violent fire. In reality, an event sufficient to release airborne fuel would also raise the contaminants several hundred feet before they could disperse downwind. Probably very little airborne contamination would be present less than 100 meters from the source.

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APPENDIX E

FUEL RELEASE TO THE OCEAN ENVIRONMENT

A. INTRODUCTION

The proven ability of the fuel capsule to withstand at least 20,000-ft ocean depths is equivalent to precluding a fuel release for burial below about 99 percent of the ocean surface. Nevertheless, one of the purposes of a complete safety analysis is to evaluate the effects of a fuel release in order to thoroughly understand the hazards involved with the operation or transportation of a radioisotope fueled generator. In the event of seawater release, the potential hazard of Sr-90 to the general population would be through the consumption of contaminated seafood.

This appendix considers a deep water and a shallow water release. Shallow water is loosely defined as those near-shore areas where the water depth varies up to approximately 200 feet. Only a continuous release of fuel has been evaluated since an instantaneous release has no physical significance with the SrTiO_3 fuel form. For academic interest, References E-1 and E-2 present the effects of an instantaneous release in deep water and shallow water, respectively. Consequences of an ocean fuel release are expressed by comparing seawater contamination to a maximum permissible continuous concentration (MPCC).

B. MAXIMUM PERMISSIBLE CONTINUOUS CONCENTRATION, MPCC

The index used to relate local seawater contamination to a permissible maximum is the maximum permissible continuous concentration proposed in Reference E-3. In this relationship, the maximum permissible seawater concentration is equated to allowable body burdens. Ingestion of the radioactivity is assumed to be taken up by man through the food chain. The equation that expresses this relationship is :

$$\text{MPCC} = \left[\frac{S_q' I_{ne}}{I_{nb}'} \right] \left[\frac{(K + B)}{B} \right] \quad (\text{E-1})$$

where

- MPCC = the maximum permissible continuous concentration for seawater, $\mu\text{c/cc}$
 S = the exposure factor = 0.1
 q' = the quantity of Sr-90 in bone at the permissible total body burden = 2 μc
 I_{ne} = the concentration of elemental strontium in seawater = 8.2×10^{-3} g/liter
 I_{nb} = the total quantity of elemental strontium in bone = 1.3×10^{-1} grams
 K = the radiological decay constant of Sr = 6.9×10^{-5} days $^{-1}$
 B = the biological decay constant for Sr for man = 3.8×10^{-5}

It is found that for Sr-90

$$\text{MPCC} = \left[\frac{(0.1)(2.0)(8.2 \times 10^{-3})}{1.3 \times 10^{-1}} \right] \left[\frac{(6.9 \times 10^{-5} + 3.8 \times 10^{-5})}{3.8 \times 10^{-5}} \right]$$

$$\text{MPCC} = 3.5 \times 10^{-2} \mu\text{c/liter} = 3.5 \times 10^{-5} \mu\text{c/cc}$$

C. DEEP OCEAN RELEASE

1. Source Term

A continuous release of radioactive fuel to the marine environment would occur since the dissolution rate of the SrTiO_3 pellets is finite. Reference E-4 cites a dissolution rate for SrTiO_3 of $1.0 \text{ mg/cm}^2\text{-day}$. For the Sr-90 content only, this dissolution rate would be equivalent to a release rate, q , of 5.79×10^{-4} curies/sec, assuming a total pellet surface area of 1395 cm^2 .

2. Dispersion Model

The above release rate was used in conjunction with the following equations which employ the Okubo and Pritchard solution (Reference E-5) modified for a continuous release

$$S_c(r, t) = \frac{q/D}{2\sqrt{\pi\omega r}} \int_{y=\frac{r}{\omega t}}^{y=\infty} e^{-y^2} dy \quad 0 < t \leq \tau_o \quad (\text{E-2})$$

$$S_c(r, t) = \frac{q/D}{2\sqrt{\pi\omega r}} \circ \frac{2}{\sqrt{\pi}} \int_{y=\frac{r}{\omega t}}^{y=\left[\frac{r}{\omega(t-\tau_o)}\right]} e^{-y^2} dy \quad t > \tau_o \quad (E-3)$$

where:

S_c is the concentration at distance, R , from source at time, T , $\mu\text{c}/\text{cm}^3$

q is the continuous release rate, curies/sec

D is the diffusion layer thickness, meters

ω is the diffusion velocity, 2.4 cm/sec (Reference E-7)

$\frac{2}{\sqrt{\pi}} \int_0^y e^{-y^2} dy$ is the error function

t is the time, sec

r is the radial distance from source, meters

τ_o is the total time required for complete dissolution

τ is the time span during fuel release, seconds

It was found for a source strength of 173,000 curies and a dissolution rate of 5.79×10^{-4} curies/sec that $\tau_o = 3.11 \times 10^8$ seconds; that is all of the Sr-90 fuel would be in solution after this time period. This value was substituted into Equations E-2 and E-3, and the continuous fuel concentration was calculated as a function of time and distance from the source. Figure E-1 is a plot of the results. The value of the error function for each calculation was taken directly from Reference E-6.

Figure E-1. Seawater Concentration of Sr-90 Versus Time for a Continuous Release

It is seen from Figure E-1 that the concentration of Sr-90 at 1000 meters from the source is below the MPCC at all times. Within a radius of approximately 190 meters from the source, the MPCC will be exceeded for about 10 years. The severity of the hazard is difficult to evaluate, due to a lack of information on the relationship between man and deep sea life. At this time, however, activity uptake by deep sea life in such a small area is not considered a serious hazard to man. (Capsule integrity has been demonstrated for deep ocean environment and hence the hazard has an extremely low probability of occurring.)

D. SHALLOW OCEAN RELEASE

1. Source Term

It has been assumed, as in the deep water release study, that the release rate is
 $M = 5.79 \times 10^{-4}$ curies/sec

2. Dispersion Model

A continuous fuel release model derived from the Carter and Okubo model (Reference E-7) was used to determine a shallow water fuel release from the SNAP-19 fuel capsule (Reference E-8). The model differs from deep sea release models in that it takes into account the actual pattern of flows and the random small scale motions which contribute to mixing. The model also incorporates a current velocity. The equation derived for fuel concentration along the plume centerline is

$$\frac{S}{M} = \left(\frac{V^{3/4}}{4\pi A_x^{1/4} A_y^{1/4} A_z^{1/2} U^{1/2}} \right) \exp \left[- \frac{V^{1/2} x^2}{4t A_x^{1/2} A_y^{1/2} U} \right] \quad (E-4)$$

where:

$$U = 1 + \frac{1}{12} \left(\Omega_y^2 \frac{A_y}{A_x} + \Omega_z^2 \frac{A_z}{A_x} \right) t^2$$

$$V = 1 + \left(\frac{1}{3} \Omega_y^2 \frac{A_y}{A_x} + \frac{1}{12} \Omega_z^2 \frac{A_z}{A_x} \right) t^2$$

u is the current velocity, meters/sec

x is the distance along centerline of plume, meters

t is the time, seconds

M is the release rate

A_x, A_y, A_z are eddy diffusivities in the x, y , and z directions, respectively, (meters)²/sec

$\Omega_x, \Omega_y, \Omega_z$ are shear values in the x, y , and z directions, respectively, sec⁻¹

Equation E-4 is based on a coordinate system that moves with the center of the plume. This equation was used to determine the seawater concentration variation with time and distance from a SNAP-23A fuel capsule.

The values used in Equation E-4 are:

$$A_x = A_y = 3.9 \times 10^{-1} \text{ m}^2/\text{sec}$$

$$A_z = 10^{-3} \text{ m}^2/\text{sec}$$

$$\Omega_y = 10^{-4} \text{ sec}^{-1}$$

$$\Omega_z = 2 \times 10^{-3} \text{ sec}^{-1}$$

$$u = 10^{-1} \text{ m/sec}$$

Substitution of the above values in Equation E-4 with varying t and x yields the curves shown in Figure E-2. At 100 meters from the source and beyond, the Sr-90 concentration is well below the MPCC value.

From the results presented in Figure E-2, it can be concluded that a continuous release of the SNAP-23A fuel inventory in shallow ocean water does not present a significant safety hazard.

E. RELEASE AFTER 280 YEARS

After 280 years, the fuel inventory will have decayed to approximately 170 curies. The concentration curves for both deep and shallow ocean release can be directly

Figure E-2. Seawater Concentration of Sr-90 Versus Time for a Continuous Release in Shallow Water

reduced by a factor of 1000. The concentration would probably exceed the MPCC only for a distance on the order of 1 meter from the source. Burial of the generator in shallow coastal waters will relieve the hydrostatic pressure on the capsule and tend to make corrosion the dominant failure mode. The low corrosion rate of Hastelloy-C will provide shallow water fuel containment until fuel release is no longer a hazard.

F. PHYSICAL AND BIOLOGICAL CONSIDERATIONS RELATED TO A STRONTIUM-90 FUEL RELEASE IN THE OCEAN

In Reference E-1, a broad discussion of physical and biological effects was presented. A brief summation of this discussion is presented here to complete the overall picture of an ocean release of the strontium-90 fuel.

A number of factors must be considered in studying the consequences of a fuel release from a safety standpoint. The physical forces that cause the fuel to be either concentrated in a specific area or spread over a large region must be known. Types of marine life and their relative abundance must be known and factored into a safety analysis. The manner in which marine life takes up released radioisotopes, specifically Sr-90 and its daughter products, should be understood and related to its effect on man via his food chain.

Prior to the complete dispersion of the Sr-90 fuel by turbulent diffusion with or without the aid of bottom currents, the fuel will be involved in many interactions with the physical and biological environment. The physical interactions include diffusion and currents, absorption by bottom sediments, and upwelling. Biological aspects include the following:

- 1) Uptake of radioisotopes by phytoplankton and zooplankton
- 2) Benthonic animal uptake
- 3) Uptake by demersal fish (plaice, rays, etc.)
- 4) Pelagic fish (surface dwelling fish) contamination

Figure E-3 depicts the physical and biological modes of a fuel release through the hydrosphere.

Figure E-3. Possible Modes of Fuel Release Through the Hydrosphere

A number of physical transport processes exist in the ocean which result in either a movement of water along the bottom of the ocean or in a vertical direction. Included in these processes are the following:

- 1) Bottom currents of cold, polar water
- 2) Turbidity currents resulting from a geophysical disturbance
- 3) Upwelling due to the wind shear at the air-water interface

Another type of physical transport process is the transfer of radioactive particles into bottom sediments which possess an affinity for them. However, the rate of removal of Sr-90 from seawater by the marine sediments is very low. Hence, this transport mechanism need not be considered.

Biological transfer processes include diurnal migration of zooplankton and seasonal migrations of mobile benthonic organisms.

The actual uptake of radioisotopes by marine life in a contaminated area depends directly upon the type of life in the immediate area. Since the species of marine life vary greatly from the surface waters to the great depths, the uptake problem is just as involved as the different varieties of sea life in the sea.

The uptake of Sr-90 and other radioisotopes by marine life has been under study for a number of years. It has been found that different species of marine life are affected by different radioisotopes. Some species concentrate the radioisotopes in certain parts of their bodies, while other reject them entirely.

The vertical distribution of marine life in the ocean has been defined by a number of marine biologists. (See References E-9, E-10, and E-11.) With a knowledge of the vertical distribution of marine life, the uptake and concentration of Sr-90 by various forms of marine life can be evaluated. In recent years, this area of marine biology has received considerable attention by many researchers. Some experimental results and their significance are discussed in References E-12 through E-21.

G. CONCLUSIONS

1. Deep Water Release

a) An assumed continuous release model yields results which are always less than the MPCC except for a region no greater than 190 meters in radius from the source.

b) A fuel release at the design depth of the capsule (20,000 feet) would not directly affect marine animals which constitute part of man's food supply. The exact nature of the hazard cannot be determined since little is known about deep sea marine life uptake.

c) Fuel released in water depths of approximately 9000 feet or less could possibly be taken up by local marine life. However, the degree of uptake would be very limited because of the small release area where the permissible concentrations are expected.

d) Although lower forms of marine life concentrate large amounts of radioisotopes in their bodies, the amount of activity transferred decreases the higher one goes up the food chain. By the time man is reached, it is quite possible that the severity of the hazard would be decreased appreciably.

2. Shallow Water Release

From the results of this study it can be concluded that a continuous shallow water release of the SNAP-23A fuel inventory will not present a significant nuclear safety hazard. Seawater contamination would be localized to a small area, and the marine population would not be excessively exposed to the released fuel. Thus, marine life uptake would be at a minimum.

It should be noted that the probability of a fuel capsule rupture is extremely low in shallow water. The SNAP-23A fuel capsule has been designed for absolute containment of the fuel inventory in hydrostatic pressure environments of 10,000 psi (equivalent to > 20,000 feet of depth). Hence, only seawater corrosion would eventually destroy fuel containment in the shallow depths since large hydrostatic pressures are nonexistent. It is expected that the low corrosion rate of Hastelloy-C will provide shallow water fuel containment until fuel release is no longer a hazard.

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APPENDIX F

EXPLOSION-OVERPRESSURE STUDY

A. INTRODUCTION

A preliminary safety evaluation of the SNAP-23A system (Reference F-1) concluded that the safety criterion originally established for overpressure (Reference F-2) was too severe. It is the purpose of this appendix to review the explosion environment that is applicable to SNAP-23A, present an accident model, and evaluate the SNAP-23A system response using the accident model.

B. THE CRITICAL ACCIDENT

It is reasonable to expect that terrestrial sites selected for SNAP-23A deployment will be relatively free from hostile environments because of prior site safety evaluations and judicious choice of site locations. On the other hand, transportation does not represent a controlled environment and does introduce the possibility of accidents resulting in explosions. Without completely reviewing the wide range of potential transport accident environments, it suffices to say that a tank truck or airplane accident represents the major source of an explosive hazard. At least one major explosion accident with a propane carrier is known to have occurred some seven years ago in Pennsylvania. In this particular case the truck, carrying 9000 gallons of liquid propane skidded, rolled over, ruptured and the contents subsequently exploded. The explosion apparently produced moderate overpressure at some distance since it severely damaged nearby structures. Propane should result in somewhat greater TNT yield than gasoline and thus the 9000 gallons (maximum DOT permitted load) is assumed to be a maximum credible environment.

C. BACKGROUND ON EXPLOSION OVERPRESSURE

1. Definitions

Explosion - Overpressure calculations presented here use a standard approach based on TNT equivalent yield. While this approach has shortcomings for explosions which result

from an air-propane mixture, it is consistent with the bulk of the literature and eliminates the use of little understood empirical equations. Some of the factors involved in an explosion overpressure or blast wave are as follows. (These are described or used in many references, e.g., References F-3, F-4, F-5.)

Overpressure, p - Pressure above ambient immediately behind explosive shock front. Can be determined by established relations for reference TNT explosions.

Dynamic Pressure, q - Pressure caused by particle velocity behind shock front.

$$p = \frac{1}{2} \rho u^2 = \frac{5}{2} \cdot \frac{p^2}{7p_o + p}$$

where p_o is ambient pressure and u is local velocity of shock front.

Reflected Overpressure, p_r - Pressure above ambient felt at the surface of a structure subject to a blast overpressure.

$$p_r = 2p \left(\frac{7p_o + 4p}{7p_o + p} \right)$$

also,

$$p_r = 2p + (\gamma + 1) q \quad (\gamma = 1.4)$$

The ratio of p_r (felt by a structure) to p (free air overpressure) varies from 2 to 8.

Pulse Duration, t_d - Time required for overpressure to decay to ambient pressure (see Figure F-1).

Impulse - Area under the positive pressure versus time curve. Impulse may vary for explosions having the same p and t_d . Impulse is a major factor in representing the damage potential of an explosion. (See Figure F-2).

Shock Front Mach Number, M_x - Ratio of local velocity to speed of sound.

Figure F-1. Typical Overpressure-Time History

Figure F-2. Dynamic Load for Cylindrical Structure

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2. Scaling Relations

The scaling law for explosions is based on fundamentals of geometric similarity. Two explosions can be expected to give identical blast wave intensities at distances which are proportional to the cube root of the respective energy release. The following relations are sufficient to define a typical overpressure in a standard atmosphere.

$$\text{yield factor} = \lambda = \left(\frac{W}{W_o} \right)^{1/3}$$

where:

W = energy release for explosion desired

W_o = energy release for reference explosion

The properties of the reference explosion can be obtained from Table XI of Reference F-3 (among others).

$$\text{Scaled distance} = d_s = \frac{(\text{actual distance, } d)}{\lambda}$$

The scaled distance is used as the independent variable in a reference explosion table. Desired properties of the blast environment can be interpolated in the table. These properties include p/p_o , M_x , and t_d .

3. TNT Yield

If the explosion is a bare spherical charge of TNT, overpressures are readily characterized. For other explosives, the most difficult part of overpressure determination is to estimate the equivalent TNT yield of the chemical involved. Usually, the yield has a very broad range and for this reason is characterized in probability terms. For example, the yield of a liquid oxygen and hydrogen explosion has been roughly determined through the work of Sandia and Douglas Corporations for the Saturn launch vehicle as illustrated in Figure F-3. The yield varies somewhat for different amounts of propellant involved (a mixing problem). Note that the effective yield of these liquid propellants is in the range of one to 15 percent. A solid propellant can be detonated to produce very high yields of 50-100 percent TNT depending on the propellant. On the other hand, the yield of typical

Figure F-3. TNT Yield Occurrence Probability (Typical for LO_2/LH_2)

terrestrial hydrocarbon fuels such as gasoline, butane or propane can be expected to be significantly lower than the yield of liquid propellants. Even with perfect mixing, the flame speed of a gasoline explosion, as occurs in an automobile engine, is relatively slow. The requirement to mix the fuel with air produces a low density mixture as compared to the compact TNT type explosive. Energy from a low density air fuel mixture will be released slowly. There is also a spatial relationship which must be considered in a fuel explosion. To achieve a 16:1 air fuel mixture for an optimum propane explosion requires substantial area. For 9000 gallons, this will be much greater than 20 feet by 20 feet and will not be circular. At a distance of 20 feet from the center of the explosion, the overpressure from a propane blast will be much less than that from a TNT source of overall equivalent yield.

4. Close Field Liquid Propellant Explosion

Some evidence suggests that actual overpressures near a liquid propellant explosion are somewhat less than would be predicted by a straight TNT equivalence approach. Reference F-6 explores this area in some detail. How close is "near" can be evaluated through the use of an equation. In this "close field," overpressures are less than predicted for TNT equivalence. In the "far field," TNT equivalence is adequate. The boundary separating the two regions is defined by:

$$\text{CFR} = \text{close field radius} = \left(\frac{15W}{\pi \rho_o} \right)^{1/3} \text{ ft.}$$

W is the far field TNT yield and ρ_o is ambient density.

Parametric curves in Reference F-6 enable a TNT equivalence overpressure to be reduced for close field effects. The results are valid for $\text{LH}_2 - \text{LO}_2$ and RPI-LO_2 propellant explosions. It is not known whether other fuel oxidizer combinations (e.g., propane-air) will exhibit the same close field overpressure characteristics.

D. CALCULATED OVERPRESSURE ESTIMATES

The analytical model described above was used to evaluate the overpressure produced by exploding 9000 gallons of propane. For estimation purposes, it will be assumed that the

propane is released to the local area surrounding the carrier and the center of the resulting explosion is about 30 feet from the unsheltered SNAP-23A generator. Calculation of overpressure has been made with the aid of Table XI of Reference F-3. The equivalent TNT yield is not known, but a parametric approach can be used to establish that peak overpressures will be much less than the ultraconservative criterion applied to date (28,000 psi after Reference F-2).

The explosion characteristics are summarized in Table F-1 which has been included primarily because it gives some insight into the way in which typical explosion parameters vary. Calculated overpressure and reflected overpressure are shown in Figure F-4.

The expected yield for hydrocarbon fuel explosions will be lower than indicated by Figure F-3. Thus, a two percent TNT yield is a reasonable upper bound for a highly unlikely SNAP-23A explosion environment. The maximum reflected pressure resulting from a two percent yield at 30 feet is 330 psi. The pressure loading which a cylindrical object will experience is indicated by Figure F-2. The difference in shape between Figures F-1 and F-2 is due to the traverse of the idealized spherical shock around the cylindrical SNAP-23A target.

All values in Table F-1 and Figure F-4 are calculated assuming no close field effect. Although no data exist to substantiate a reduction in TNT equivalence, it is likely that a close field does exist for propane-air. The significance of the close field effect, as given in Reference F-6, is revealed in the following analysis.

Consider again the case for the propane two percent TNT yield at 30 feet.

$$CFR = \left[\frac{(15)(860)}{\pi(0.002378)(32.2)} \right]^{1/3} = 174 \text{ feet}$$

Therefore, 30 feet is well within the close field region.

From Figure 7 of Reference F-6, p is effectively reduced by $2/3$.

$$p_{\text{close field}} = (2/3)(73.5) = 49 \text{ psi}$$

$$p_r_{\text{close field}} = 192 \text{ psi}$$

TABLE F-1
EFFECT OF EXPLODING 9000 GALLONS OF PROPANE

<u>Assumed TNT Yield</u>	<u>d</u>	<u>λ</u>	<u>d_s</u>	<u>p/p_o</u>	<u>p</u>	<u>p_r</u>	<u>t_d</u>
1%	20	0.603	33.2	7.64	112	575	6.3
2	20	0.760	26.3	13.0	191	1130	5.7
4	20	0.957	20.9	19.7	290	1865	2.6
8	20	1.21	16.6	30.5	450	3100	1.0
100	20	2.80	7.2	142	2090	16170	2.1
1%	30	0.603	49.7	2.7	39.7	148.5	9.3
2	30	0.760	39.5	5.0	73.5	330	7.4
4	30	0.957	31.3	8.65	127	676	6.1
8	30	1.21	24.9	14.2	209	1260	5.6
100	30	2.80	10.7	70.3	1032	7700	1.4

NOTES:

- a. 9000 gallons propane $\approx$ 43,800 lb.
- b. Reference explosion is a 2000 lb spherical charge of TNT
- c. $p_o = 14.7$ psi
- d. These data are not corrected for close field effects

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Figure F-4. Effect of Yield on Peak Overpressure and Peak Reflected Overpressure

E. PROBABILITY OF A SEVERE OVERPRESSURE FOR SNAP-23A

Reference F-1 suggested some probability values for explosion during transport of the SNAP-23A generator. This subject has been reviewed and there are two significant additions: (1) the probability of achieving an arbitrary TNT yield value and (2) some actual statistics for tank truck vehicles (Reference F-7). Consider the probability of achieving the two percent TNT yield overpressure for 9000 gallons of propane at a distance of 30 feet. The entire probability may be evaluated as follows:

P = probability of achieving the critical accident

P = $(P1) (P2) (P3)$

where,

$P1$ = accident with a dangerous cargo carrier in 3000 miles of transport

= 1.8×10^{-4} (estimated using data from References F-8 and F-9)

$P2$ = accident producing secondary explosion

= 4.4×10^{-3} (calculation from data in Reference F-7)

$P3$ = achieving two percent equivalent TNT yield

< 0.55 from Figure F-3

thus,

$P < (1.8 \times 10^{-4}) (4.4 \times 10^{-3}) (0.55)$

$P < 4.35 \times 10^{-7}$

F. RESPONSE OF SNAP-23A SYSTEM TO EXPLOSIVE OVERPRESSURE

The response of the SNAP-23A System has been determined by using the blast environment from the two percent TNT yield explosion at a distance of 30 feet. The specific components analyzed are the protective cover around the fins and the radiator fins.

1. Protective Cover

The response of the Inconel 625 protective cover to a blast pressure load can be approximated by considering the cover to be a perforated cylindrical shell. Schuman (Ref. F-10)

postulated a failure criterion for blast loaded cylindrical shells, without holes, based on actual test data. The failure criterion for the explosive blast was a radial permanent deformation equal to approximately five to 10 percent of the original shell diameter. Based on the dimensions of the protective cover, the energy release from the accident model explosion and the mechanical properties of Inconel 625, the critical pressure required to cause an unperforated shell to fail has been computed to be 6030 psi. The presence of the holes causes a gross reduction in the magnitude since there is a 60 percent void in the protective cover shell. Employing efficiency factors to account for holes in pressurized vessels according to A.S.M.E. (Ref. F-11) it was computed that the protective cover is only 20 percent efficient. Therefore, the critical external pressure required for the protective cover to fail is reduced to 1200 psi. Since the incident overpressure from Table F-1 is only $p = 73.5$ psi, it is concluded that the protective cover will survive the blast.

2. Radiation Fin

A cover with a small open area surrounding the fins would reduce the overpressure of the fins. However, there is some evidence (Reference F-11) that covers with more than 50 percent open area will not attenuate the blast wave and reduce damage behind the cover. The crossover where protection is not achieved seems to occur at about 50 percent open area. It is assumed in this analysis that the protective cover, consisting of 60 percent void, does not alter the pressure load of Figure F-2.

The dynamic response of the family of radiator fins to blast overpressure is a complex problem. If one can ascertain the net pressure versus time profile for each individual fin, its response to the explosion can be determined by the method outlined in Harris and Crede (Reference F-12). From Figure 49.18 of Reference 12 the deflection of a structure is estimated based on its static strength characteristics, its natural frequency, and the blast pressure versus time environment.

The SNAP-23A radiator fins are approximately 2.6° apart and extend around the periphery of the hub. The forward fins, with respect to the shock front (see sketch below), will experience a minimum pressure differential across them due to nearly equalized pressures in the bays.

The side fin could be a critical case and lends itself readily to an approximate analysis since its pressure load can be estimated. The side fin sees an overpressure pulse, exclusive of any reflected overpressure contribution (Reference F-3, p. 123), consisting of two parts:

- (a) Side overpressure magnitude for the time span equal to the fin thickness divided by the shock front velocity.
- (b) Drag pressure over the time span until pressure relief is obtained.

These two overpressures are roughly equal in magnitude. The side fin, therefore, experiences the pressure profile shown in Figure F-5. Figure F-6 is an equivalent elastic-plastic curve which has been generated to approximate the stress-strain curve for the 1100 aluminum fin material.

The ductility ratio, λ , defined in Reference F-12, has been computed as the maximum acceptable deflection, X_m , divided by the yield deflection, X_y :

$$\lambda = \frac{X_m}{X_y} = \frac{\text{ductility strain}}{\text{equivalent yield strain}} = \frac{0.3}{0.04} = 7.5$$

This magnitude of λ implies a ductile structure as would be expected for the fin material.

The effective natural period of the fin, considered to be a uniformly loaded beam with one end simply supported and the other clamped, is computed:

$$\tau = \frac{2 \pi}{15.4} \sqrt{\frac{\mu l^4}{E I}}$$

where:

Figure F-5. Side Fin Pressure Profile

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Figure F-6. Equivalent Elastic-Plastic Curve for 1100 Aluminum

- μ = mass per unit length of fin
 l = length of the fin = 6.625 in.
 E = effective elastic modulus from curve above =

$$\frac{10500 \text{ psi}}{0.04 \text{ in./in.}} = 0.2625 \times 10^{-6} \text{ psi}$$

- I = area moment of inertia of fin
 τ = 0.03 sec.

The ratio:

$$\frac{\text{pressure load duration}}{\text{effective natural period}} = \frac{t_d}{\tau} = \frac{\sim 0.0159 \text{ msec.}}{0.03 \text{ sec.}} \\ = 0.00053$$

The time the side fin experiences the overpressure is extremely short compared to its effective natural period.

Due to the short loading time and the close proximity of neighboring fins, the fin would not experience the overpressure load uniformly over its entire leading surface area. However, in determining the effective resistance of the fin structure, r_y , the closest approximation to this case was the uniformly loaded beam consisting of simply supported and clamped ends. From the equation for τ , and the equations for the beam of maximum deflection and bending moment, r_y was derived to be:

$$r_y = \frac{\sigma_{\epsilon y} b h^2}{0.586 l}$$

where:

- $\sigma_{\epsilon y}$ = the effective yield stress from the elastic pastic curve = 10500 psi
 b = average width of fin = 8 in.

$$h = \text{fin thickness} = 0.062 \text{ in.}$$

$$\therefore r_y = 83 \text{ psi}$$

The ratio of the maximum overpressure, p_m , acting on the fin to its yield resistance can now be determined:

$$\frac{p_m}{r_y} = \frac{73.5 \text{ psi}}{83 \text{ psi}} = 0.885$$

From the dynamic response curve (Figure 49.18 of Reference F-12) for the input parameters $\frac{t_d}{\tau} = 0.00053$ and $\frac{p_m}{r_y} = 0.885$, the resulting λ is indicated to be less than one.

This implies that the yield strain of the fin is not exceeded and no permanent deformation takes place.

While the foregoing analysis is approximate, it indicates that the SNAP-23A over-pressure accident environment will not lead to failure of the SNAP-23A System through the loss of heat rejection capability.

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APPENDIX G

IMPACT ANALYSES

A. INTRODUCTION

Impact represents one of the more severe postulated environments to which the SNAP-23A system may be exposed. Potential impact accidents range in severity from short drops during handling and storage for the high velocity impacts which occur as a result of an airplane crash. This appendix is concerned only with the immediate mechanical effects of impact (as differentiated from corrosion or thermal problems where the failure mechanism may have been originated by mechanical impact damage).

Four specific configurations are treated. These are the bare heat source, the heat source in its shipping container, the SNAP-23A system, and the SNAP-23A system in its shipping container. In general, the impact accidents for the heat source are treated in considerable depth, and the detailed analyses are available in references. The accidents posing a problem for the SNAP-23A system generally occur during transportation and involve a complex overall configuration that is not amenable to transient stress analysis on a detailed component basis. The approach used for SNAP-23A system impacts is generalized, so that it serves as a tool for the selection of a shipping container configuration as well as for evaluation of the fuel capsule and biological shield integrities.

The primary purpose of all impact evaluations is to assure that safety criteria are met for the postulated credible accident. The criteria require that the fuel be absolutely contained within the fuel capsule walls and that radiation levels be less than 1000 mr/hr at 3 feet from an accessible surface of the configuration evaluated.

B. Heat Source Handling Accidents

1. Introduction and Summary

The accident situation postulated as credible during heat source handling is a free fall from a height of 6 feet onto a non-yielding surface. This accident could occur, for example, when the heat source is removed from the heat source shipping container and

placed into the generator base assembly. For this situation the heat source surface temperature is assumed to be 1100°F.

The impact orientations of the heat source used in this analysis are top, bottom, side, and corner. The method of analysis utilized is based on the energy capacity of the material and the dynamic stress at impact. Conservatism in the approach is obtained by purposely ignoring the resistance and strength of the outer Inconel steel shell and the attenuation of the copper material between the Inconel and U-8 Mo.

Only in the top impact case is the copper material considered. In this later case, the deformation of the copper accumulator stud resulting from the 6 foot free fall is found to be 1.6 inches. The strength and resistance of the outer Inconel shell are conservatively ignored. The stress transmitted by the copper in this case is found to be 4940 psi. Compared to the 26,400 psi yield strength of the U-8 Mo at 1100°F, the value is acceptable, and failure or severe damage to the biological shield will not occur.

The bottom impact case, ignoring the resistance and strength of the Inconel shell and the attenuation of the copper layer, produces a maximum dynamic stress of 32,974 psi in the biological shield. The indentation or deformation based on the contact area at the bottom of the shield is found to be 0.115 inch. Since the U-8 Mo at 1100°F is quite ductile and the attenuation and resistance afforded by the copper and Inconel shell have been ignored, the values calculated are considered acceptable. The impact stress at the fuel capsule U-8 Mo interface is found to be 16,500 psi. This value is also acceptable.

Indentation or deformation resulting from the side impact case is calculated to be 0.217 inch. As in the bottom impact case, the resistance to deformation and strengthening of the Inconel shell is conservatively ignored. The maximum dynamic stress in the U-8 Mo due to the impact velocity resulting from the 6 foot free fall is the same as that determined for the bottom impact case. Based upon the conservatism inherent in the analysis and the magnitude of the values calculated, severe damage to the biological shield in the event of a side impact situation is not probable.

Corner impact at the bottom edge of the heat source represents the most severe impact orientation. The complexities of the geometry in the main U-8 Mo shell and the U-8 Mo plug cannot be rigorously analyzed. Based on the energy capacity of the U-8 Mo, the maximum indentation is found to be 0.27 inch. As in the previous case, the resistance to indentation afforded by the Inconel shell and the attenuation of the copper are conservatively ignored. As long as the close tolerances at the U-8 Mo plug and main U-8 Mo biological shield are maintained, the two parts may be assumed to act as one unit. The conservatism inherent in the analysis would indicate that severe damage to the biological shield in the event of a corner impact situation is unlikely.

The present uncertainty in the ductility of the shield at temperatures below 1100°F requires that low ductility be assumed when the heat source is allowed to cool below this temperature. Impact of the heat source in this condition would not significantly alter the extent of shield deformation but could produce a fracture in the shield wall. However, the irregular features that such a crack would exhibit and the effectiveness of the ductile steel jacket in preventing partition of the shield at the crack boundary should prevent any significant change in radiation level.

2. Calculational Details

The following summarizes the impact calculations performed in detail. (These are also reported in Reference G-1.)

Top Impact (Case I, Figure G-1)

The impact resistance of the copper accumulator at 1200°F* is determined by the parabolic stress-strain law (Reference G-2). This relationship, generated by Bell for the dynamic stress-strain behavior of annealed copper at 1200°F, is as follows:

* 1200°F property data are used throughout since U-8 Mo 1200°F rather than 1100°F data are available.

Figure G-1. Top Impact

$$\sigma = \alpha \left(1 - \frac{T}{T_M} \right) \epsilon^{1/2}$$

where

σ is the stress

T is the state temperature

T_M is the melting temperature

ϵ is the strain

The value for α is given by Bell as

$$\alpha = 2.7 \times 10^4 \text{ psi}$$

and the operating (state) temperature of the system is

$$T = 1200^\circ\text{F} = 922^\circ\text{Kelvin}$$

with

$$T_M = 1358^\circ\text{Kelvin}$$

Therefore, the dynamic stress-strain behavior for copper at the instant of impact is given by

$$\sigma = 8640 \epsilon^{1/2}$$

The potential energy U of the copper per unit volume of material is

$$U = \frac{2}{3} \beta \epsilon^{3/2}$$

where

$$\beta = \alpha \left(1 - \frac{T}{T_M} \right)$$

Neglecting the strength of the outer shell, the displacement of the copper accumulator can be expressed as a function of the maximum longitudinal deformation of the accumulator.

$$x = \epsilon_{\max} \ell \text{ (displacement of heat source assembly during impact)}$$

$$\ell = 5 \text{ inches (length of accumulator)}$$

The energy dissipated by the accumulator during deformation is therefore

$$E = \frac{2}{3} \beta \epsilon^{3/2} A \ell$$

$$E = \frac{2}{3} \beta x^{3/2} \ell^{-1/2} A$$

where

A is the cross section area of the accumulator

ℓ is the length of the accumulator

and

$$A = \frac{\pi}{4} (3.7)^2 = 10.75 \text{ in.}^2$$

$$\ell = 5 \text{ in.}$$

$$E = 27,600 x^{3/2} \text{ in.-lb}$$

Equating the energy dissipated by the copper accumulator to the kinetic energy of the heat source assembly unit,

$$\frac{1}{2} m v^2 = 27,600 (x)^{3/2}$$

yields a deformation (x) of mass (m) of the heat source = 800 lb

$$x = \left[\frac{(235.8)^2 \left(\frac{400}{386} \right)}{27,600} \right]^{2/3}$$

$$x = 1.6 \text{ inches}$$

The stress in the copper resulting from this deformation is

$$\sigma = 8640 \epsilon^{1/2}$$

$$\sigma = 8640 \left(\frac{x}{l} \right)^{1/2}$$

$$\sigma = 4940 \text{ psi}$$

This is a conservative evaluation of the accumulator deformation since the strength of the Inconel shell is neglected. The values are realistic for bare copper.

Bottom Impact (Case II, Figure G-2)

The stress in the U-8 Mo at the impact interface with a non-yielding surface (see Figure G-2) is

Figure G-2. Bottom Impact

$$\sigma_B = \rho c v$$

where

$$\rho = \frac{0.624}{386} \quad (\text{mass density of U-8 Mo}), \quad \frac{\text{lb-sec}^2}{\text{in.}^4}$$

$$c = \left(\frac{E}{\rho} \right)^{1/2} \quad (\text{velocity of stress wave}), \quad \text{in./sec}$$

$$E = 12 \times 10^6 \quad (\text{elastic modulus}), \quad \text{lb/in.}^2$$

and

$$\sigma_B = 32,974 \text{ psi}$$

The maximum strain is determined by

$$\epsilon_{\max} = \frac{v}{c}$$

which yields

$$\epsilon_{\max} = 0.00275 \frac{\text{in.}}{\text{in.}}$$

The stress at the U-8 Mo to fuel capsule interface is determined by:

$$\alpha_i = \frac{\rho_H C_H V}{1 + \frac{\rho_H C_H}{\rho_U C_U}}$$

where

V is the impact velocity, $\frac{\text{in.}}{\text{sec}}$

$$\rho_H C_H = 143.3 \quad (\text{impedence of Hastelloy material}), \quad \frac{\text{lb-sec}}{\text{in.}^3}$$

$$\rho_U C_U = 140 \quad (\text{impedence of U-8 Mo material}), \quad \frac{\text{lb-sec}}{\text{in.}^3}$$

which yields

$$\sigma_i = 16,500 \text{ psi}$$

The impact surface is limited to the U-8 Mo plug diameter (D), Figure G-2, of 8.4 inches. The deformation Y is

$$Y = \frac{\text{Vol}}{A} \quad (\text{inches})$$

where

A is the area at impact interface

$$A = \frac{\pi}{4} D^2$$

and

Vol = volume of material dissipating the kinetic energy

The volume of material required to absorb the kinetic energy of the heat source unit is determined by

$$\frac{1}{2} mv^2 = WH = \int \text{vol } \sigma d\epsilon$$

where

W = 800 lb (weight of assembly)

H = 72 inches (free fall distance)

$\int \sigma d\epsilon$ is the area under the stress-strain curve at 1200°F

By use of the above values and the area under the idealized stress-strain curve shown in Figure G-3, the volume of U-8 Mo required to dissipate the kinetic energy of the heat source is found to be

$$\text{Vol} = 6.36 \text{ in.}^3$$

Therefore, the deformation Y is

$$Y = \frac{6.36}{\pi(4.2)^2}$$

$$Y = 0.115 \text{ in.}$$

Figure G-3. Idealized Stress-Strain Curve for
U-8 Mo at 1200°F

Side Impact (Case III, Figure G-4)

The maximum stress during impact on the surface is the same as Case II. The energy of impact is absorbed in 6.36 cubic inches of shield material; thus

$$\sigma_S = \sigma_B = 32,974 \text{ psi}$$

$$\sigma_i = 16,500 \text{ psi}$$

$$\text{Vol} = 6.36 \text{ in.}^3$$

The deformation Y is determined by the following equations:

$$Y = r \left(1 - \cos \frac{\alpha}{2} \right)$$

$$\alpha = \left(\frac{Vol}{\ell} + \frac{r^2}{2} \sin \alpha \right) \frac{360}{\pi r^2}$$

where

$$r = 6.2 \text{ in.}$$

$$\ell = 13.9 \text{ in.}$$

Solving for α yields

$$\alpha = 30^\circ$$

Therefore,

$$Y = 6.2 (1 - \cos 15^\circ)$$

$$Y = 0.217 \text{ in.}$$

Figure G-4. Side Impact

The evaluation of the deformation Y of the heat source assembly after falling 6 feet is conservative since any attenuation and resistance by the copper and Inconel surrounding the U-8 Mo shield are ignored. One can expect the effect to be less severe than shown by this approach.

Corner Impact (Case IV, Figure G-5)

The volume of material required to dissipate the impact energy is approximated by that volume within the section of an oblate spheroid shown in Figure G-5.

The deformation Y is determined by

$$Y = R_1 - \left(R_1^2 - a^2 \right)^{1/2}$$

$$Y = R_2 - \left(R_2^2 - b^2 \right)^{1/2}$$

where

$$R_1 = 6.7 \text{ in.}$$

$$R_2 = 3.2 \text{ in.}$$

From the preceding analysis, the volume of required material was found to be 6.36 in.^3 . Equating this volume to the volume of an oblate spheroid yields three equations with three unknowns. Solving for "a" yields:

$$\text{Vol} = \frac{2}{3} \pi a b^2$$

$$a = 1.9 \text{ in.}$$

Therefore, the deformation Y is found to be

$$Y = 0.27 \text{ in.}$$

The value of "Y" determined above is conservative since attenuation and resistance by the copper and Inconel surrounding the U-8 Mo shield are ignored.

Figure G-5. Corner Impact of Heat Source

C. HEAT SOURCE IN SHIPPING CONTAINER DURING TRANSPORT

1. Introduction

Through the use of analytical methods the integrity of the heat source and shipping container assembly was investigated for an accident environment. In Section VI-B it is established that a 30-foot free fall onto a non-yielding surface is a hypothetical representation of a head-on collision of vehicles at probable impact velocities. Due to the wide range and magnitude of the various parameters associated with this type of collision, the 30-foot free fall criterion offers a realistic approach for assessing the integrity of the packaged unit in an accident environment.

The analyses consist of five sections. The first three of the sections are concerned with determining loads imposed on the inner container and heat source based on an impact velocity of 44 ft/sec attained by a free fall of 30 feet. The impact attitudes of the container package treated in these three sections are end-on, side, and corner. A fourth section treats the puncture resistance of the unit based upon a free fall from a height of 40 inches onto a 6-inch diameter mandrel 8 inches in length. The results of these analyses are then used in the fifth section, which presents the analysis of the heat source loadings. The shipping container configuration depicted in Figure G-6 utilizes thick layers of crushable honeycomb to absorb impact energy and reduce the loading to the heat source assembly. Details of the shipping container are given in Section V.

2. Summary of Results

For the end-on impact case, the end plate is assumed to strike a rigid surface after free fall from a 30-foot height. The energy absorbed by the outer container is neglected. During this impact the inner 5-inch layer of honeycomb was crushed 2.2 inches and the outer 12-inch layer about 0.5 inch. Neither of these values approaches 70 percent of the original thickness which is the limit beyond which the crush force will rapidly increase above a constant and tolerable value.

Figure G-6. Shipping Container Assembly

In the case of the side impact, the outer can is assumed to strike a rigid surface after dropping from a 30-foot height. Again, the impact velocity is 44 ft/sec. The outer can is assumed to behave as a membrane with no capability of withstanding bending loads. All of the energy absorption takes place in the side honeycomb. The side honeycomb temperature at the inner container averages about 250°F at maximum ambient temperature and 94°F under the coldest ambient temperature. The thickness of the side honeycomb is 11 inches. For this impact the honeycomb is crushed 5 inches or 45 percent of its total thickness. This is again less than the 70 percent constant pressure crush limit.

The corner impact case is more severe than either the side or end case from the standpoint of the percentage of the initial thickness of honeycomb that is crushed. In this case, it was assumed that impact occurs on the corner of the shipping container after a 30-foot drop onto a non-yielding surface. Impact velocity is 44 ft/sec, and the system center of gravity was assumed to be directly above the corner so that rotation does not occur. It is assumed that the honeycomb temperature is constant at 125°F in the crushed area. This is the maximum temperature case and will lead to the greatest crush distance. Neither the side nor end honeycomb in a corner impact is crushed in its "preferred" direction. The kinetic energy due to impact is absorbed in an 11.6 inch corner deformation. The total thickness of honeycomb available in the corner orientation is approximately 22 inches. Therefore, only 53 percent of the initial thickness is crushed, which is well within the 70 percent allowable.

Neither the end nor side honeycomb sections adjacent to the inner container are computed to crush. Since the side and end honeycomb structures do not receive support from one another, these layers must be sufficiently rigid so that they crush locally rather than shearing or bending at overhanging ends.

The analyses indicate that the SNAP-23A/FPMS heat source shipping container can survive a 30-foot drop in any orientation without affecting the integrity of the heat source. Hence, containment of the radioactive fuel and effectiveness of the radiation

shielding are assured. None of the heat source components were loaded to a point approaching their mechanical strength in a transportation accident environment.

Deformation of the outer container for an end-on drop was evaluated to be 2.7 inches, while corresponding values of side-on and corner drops were 5.0 inches and 11.6 inches, respectively. These deformations are sufficiently small such that external radiation levels following the 30-foot drop will remain well below permissible limits (1000 mrem/hr at 3 feet from the external surface). The maximum surface radiation level following a free drop accident results from a side-on impact in which the dose rate at the surface increases from 50 mr/hr to 65 mr/hr.

The analyses also verify the capability of the shipping configuration to withstand the hypothetical puncture accident regardless of orientation at impact. In this postulated accident, the heat source in the shipping container drops through a displacement of 40 inches onto a 6-inch cylindrical mandrel. The kinetic energy at impact equals 10,700 foot-pounds.

The maximum penetration of the honeycomb section at the outer end is approximately 6.4 inches, neglecting the energy absorbed in punching the shell. The maximum g-load is less than that developed during the 30-foot drop case, and the penetration is less than 70 percent of the honeycomb thickness.

For lateral impact of the shipping container on the mandrel, the 28.4 lateral loading is lower than that of the axial case. The final penetration depth where the kinetic energy was absorbed is about 9.7 inches, which is an acceptable value.

3. Calculational Details

The following sections present the basic calculations performed to verify the impact capabilities of the heat source shipping container. These results also appear in Reference G-3.

a) End-On Impact. In this case the end plate strikes a rigid surface after dropping from a 30-foot height. Impact velocity is 44 ft/sec. The end honeycomb temperature at the inside averages about 195°F at maximum ambient temperature. The crush pressure increases with impact velocity and decreases with temperature.

Honeycomb Axial Crush Strength = 725 psi

Dynamic Factor = 1.04 (Reference G-4 for 44 ft/sec)

Temperature Factor = 0.88 (Reference G-5 for 195°F)

Crush pressure at Inside = $725 \times 1.04 \times 0.88 = 660$ psi

Crush Force = p_A (Area of 5 inch inner layer) = $660 \times \pi \times 11.25^2 = 263,000$ lb

The honeycomb temperature at the outside near the cover plate averages about 100°F at this condition. The crush pressure will vary with the dynamic and temperature factors again:

Temperature Factor = 0.98 (Reference G-5 for 100°F)

Crush Pressure at Outside = $725 \times 1.04 \times 0.98 = 740$ psi

Crush Force = p_A (Total Area of Bottom) = $740 \times \pi \times 22.75 = 1,204,000$ lb

For this case, we can safely neglect the energy absorbed by the outer container and simplify the system to a one-dimensional, two mass system.

First, assuming that the crushable medium between the masses does not crush at impact, the force decelerating the system would be 1,204,000 lb, and the constant deceleration would be obtained from:

$$F = MA$$

$$A = \frac{F}{M} = \frac{Fg}{W}$$

Looking at the g loading,

$$G = \frac{A}{g}$$

$$G = \frac{F}{W} = \frac{1,204,000}{3212 \text{ total weight}} = 374 \text{ g's.}$$

Thus, both masses would decelerate at this rate. The force that must be transmitted through the crushable medium between the masses is then 374 (weight of mass 1) or $374 \times 1969 = 737,000$ lb. Since this cannot occur, the crushable material between the masses will crush.

At impact we have:

Looking at Mass 2, the net deceleration force is $1,204,000 - 263,000$ or $941,000$ lb.

$$G = \frac{941,000}{1243} = 757 \text{ g's, which is the deceleration rate of the outer container.}$$

We have linear motion with constant deceleration:

$$S = V_o t + 1/2 A t^2$$

$$V = V_o = A t$$

$$A = \text{Constant} = (-757) (32.2) = -24,400 \text{ ft/sec}^2$$

$$\text{Time to Stop} = \frac{V_o}{A} = \frac{44}{24,000} = 0.001805 \text{ sec}$$

Crush distance of outer honeycomb

$$\begin{aligned} &= (0.001805) \left[(44) - (1/2 \times 24,400 \times 0.001805) \right] \\ &= (0.001805) \left[(44) - (22) \right] = 0.0397 \text{ ft or } 0.476 \text{ in.} \end{aligned}$$

Mass 1 also decelerates at a constant rate.

$$G = \frac{263,000}{1969} = 133.7 \text{ g's}$$

$$A = (-133.7) (32.2) = -4300 \text{ ft/sec}^2$$

$$\text{Time to stop} = \frac{44}{4300} = 0.01024 \text{ sec}$$

$$\text{Distance traveled} = (0.01024) (22) = 0.225 \text{ ft or } 2.70 \text{ in.}$$

$$\text{Crush distance of honeycomb between masses is } 0.225 - 0.0397 = 0.185 \text{ ft or } 2.22 \text{ in.}$$

Checking the energy relationships,

Initial Kinetic Energy = Work Done on Honeycomb

$$1/2 \frac{W}{g} (\text{total}) V_o^2 = \sum \int f dx$$

$$1/2 \frac{(3212)(44)^2}{32.2} \text{ ft lb} = 0.0397 \times 1,204,000 + 0.185 \times 263,000 \text{ lb}$$

$$96,500 \text{ ft lb} = 47,800 + 48,700 \text{ ft lb}$$

$$96,500 \text{ ft lb} = 96,500 \text{ ft lb}$$

Thus, we have an energy check on the solution.

During this impact, the inner 5-inch layer of honeycomb crushed 2.22 inches and the outer 12 inch layer 0.476 inch. Neither of these values approaches 70 percent of the original thickness, which is the limit beyond which the crush force would increase above the assumed constant value.

The above accelerations (or g loads) and displacements were obtained using the maximum predicted temperatures. Under these conditions, the honeycomb crush strengths are minimum. This tends to produce the worst crushing (maximum displacements) conditions but the minimum loading condition.

The honeycomb temperature at the inner container can be as low as 60°F when the ambient temperature drops to the minimum level. At this temperature, the temperature factor for use in calculating the inside crush pressure is essentially 1.0 and the pressure (including dynamic effects) will be about 754 psi. During impact when the honeycomb crushes, the g loading will be higher than the above value for warmer honeycomb. The g load of the inner container will be 152.5 g's rather than the above 133.7 g's. The g load of the outer components will also be higher than the above 757 g's, but this is not a critical area.

b. Side Impact. In this case the side of the outer can strikes a rigid surface after dropping from a 30-foot height. Again, the impact velocity is 44 ft/sec. The outer can is assumed to behave as a membrane with no capability of withstanding bending loads. All of the energy absorption is assumed to take place in the side honeycomb. The side honeycomb temperature at the inner container averages about 250°F at maximum ambient temperature. The dynamic factor on crush pressure is still about 1.04.

Honeycomb Radial Crush Strength = 350 psi

Temperature Factor = 0.84

Crush Pressure at Inside = $350 \times 1.04 \times 0.84 = 306$ psi

The projected area of the inner container and insulation and shim layers is approximately 875 in.². The force required to crush the inner honeycomb is then (assuming a constant pressure normal to the wall of the inner container):

Crush Force = $306 \times 875 = 267,000$ lb

Under these conditions, the outer honeycomb temperature averages about 150°F.

Again, the dynamic factor remains the same, or

Temperature Factor = 0.94

Crush Pressure at Outside = $350 \times 1.04 \times 0.94 = 342$ psi

The area of honeycomb crushing at the outside varies with crush distance since we have a cylinder crushing against a flat surface.

The width of the flat on the bottom is $2\sqrt{r^2 - (r - \delta)^2} = 2\sqrt{\delta(2r - \delta)}$.

The effective length of the outer crush area will be essentially equal to that of the inner container since the end honeycomb is oriented in the weak direction radially, and does not support the ends of the side honeycomb that overlap the inner container. The end honeycomb itself will not absorb appreciable energy in this direction either.

Thus, the outer crush force is a function of crush distance.

$$\text{Crush Force} = (342)(37)(2) \sqrt{\delta(2r - \delta)} = 25,300 \sqrt{\delta(2r - \delta)} \text{ lb}$$

The system can be represented according to the following diagram:

Actually, for this drop height the honeycomb between the masses will not crush. In fact, it will not crush until the deceleration loading reaches $267,000/1969$ or 135 g's . When this deceleration loading is reached, the force at the outside is $(135)(3212)$ or $434,000\text{ lb}$.

This occurs at a δ of 7.8 in. The force versus deflection curve at the outside of the honeycomb is shown in Figure G-7. The area under the curve at the 7.8 inch deflection is the energy absorbed while the two masses are rigidly held relative to each other (inner honeycomb is not crushed). This energy absorbed is approximately $2,318,000\text{ in. lb}$ or $193,000\text{ ft lb}$. The initial kinetic energy (total) from a 30-foot drop is only $96,500\text{ ft lb}$. Thus, enough energy would be absorbed at approximately a 5.0 inch deflection of the outer can wall to stop both masses.

The outside force at this 5.0 inch deflection is $360,000\text{ lb}$. The g load is $360,000/3212$ or 112 g's .

The thickness of the honeycomb is 11 inches. The 5.0 inch deflection is 45 percent of the thickness, again less than the 70 percent constant pressure crush limit.

Under the coldest ambient temperature, the inside temperature drops to 94°F , and the outside temperature to 12°F . The temperature factors become 0.98 and about 1.05. The crush pressures will increase to about 358 psi at the inside and 382 psi at the outside. The outside crush distance will decrease to about 4.75 in. to absorb all of the energy. The increased crush pressure of the honeycomb will increase the force at this deflection to about $391,000\text{ lb}$ (122 g's). The increased crush pressure will also increase the force at which the inner honeycomb will crush to about $312,000\text{ lb}$ (158 g's).

Figure G-7. Force Versus Deflection at Outer Container

Thus, the deceleration rate will be a maximum of 122 g's under the cold conditions and, again, the inner honeycomb will not crush.

The side honeycomb consists of four sectors, held together at the outside by the outer shell. Neglecting friction,

the deceleration of the inner container tends to produce a separation at the bottom sectors. A normal force is applied to each lower sector of magnitude of the inertia force multiplied by $\sqrt{2}/2$.

$$\text{This is } 122 \times 1969 \times \sqrt{2}/2 = 170,000 \text{ lb}$$

Summing moments about the pivot point:

$$\text{Shell Tension Force} \times 2r = 170,000 \times r \cos 45^\circ$$

$$\text{Shell Tension Force} = 170,000 \cos 45^\circ / 2 = 60,000 \text{ lb}$$

For a 1/8 inch shell of 35-inch effective length,

$$\text{tension} = 60,000 / (1/8 \times 35) = 13,700 \text{ psi}$$

This is quite acceptable in view of the 36,000 psi yield strength.

c) Corner Impact. The corner impact case is more severe than either the side or end case with respect to the honeycomb crush distance relative to the initial thickness.

In this case it is assumed that impact occurs on the corner of the shipping container after a 30-foot drop onto a non-yielding surface. Once again, impact velocity is 44 ft/sec. It is assumed that the system center of gravity is directly above the corner so that rotation does not occur.

It is assumed that the honeycomb temperature is constant at 125°F in the crushed area. This is the maximum temperature case where the crush distance will be greatest. Neither the side nor end honeycomb is crushed in its "preferred" direction. Figure G-8 obtained from Reference G-6 shows the reduction of honeycomb crush strength as the direction of compression is varied.

The bottom honeycomb will be crushed at about 40° from its "preferred" direction and the side honeycomb approximately 50° from its preferred direction. The strength reduction factors will be about 0.41 and 0.33, respectively. The temperature factor will be about 0.96. The dynamic factor will still be about 1.04. The crushing pressures will be

crush pressure - end honeycomb, $725 \times 0.41 \times 0.96 \times 1.04 = 296$ psi

crush pressure - side honeycomb, $350 \times 0.33 \times 0.96 \times 1.04 = 115$ psi

Figure G-8. Honeycomb Strength Variation with Crush Direction

The area of honeycomb being crushed will vary with the distance the corner is crushed.

Using Figure G-9 as a reference, the projected area shown in the lower figure can be found as follows:

$$s = \frac{\delta}{\sin \beta}$$

$$A_{\text{projected}} = \frac{2\alpha}{2\pi} \pi r^2 - r(r-s)$$

$$\text{but } \alpha = \sin^{-1} \sqrt{\frac{s(2r-s)}{r}}$$

$$A_{\text{projected}} = r^2 \sin^{-1} \sqrt{\frac{s(2r-s)}{r}} - (r-s) \sqrt{s(2r-s)}$$

Finally,

$$A_{\text{actual}} = \frac{1}{\cos \beta} \left[r^2 \sin^{-1} \sqrt{\frac{s(2r-s)}{r}} - (r-s) \sqrt{s(2r-s)} \right]$$

This area is the total area of honeycomb being crushed. A portion of this area is side honeycomb and a portion is end honeycomb.

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Figure G-9. Honeycomb Projected-Area Analysis

d exists when $\delta > h/\cos\beta$

$$d = s = \frac{h}{\tan \beta}$$

$$A_{\text{side honeycomb}} = \frac{1}{\cos\beta} \left[r^2 \sin^{-1} \sqrt{\frac{d(2r-d)}{r}} - (r-d) \sqrt{d(2r-d)} \right]$$

$$A_{\text{end honeycomb}} = A_{\text{total}} - A_{\text{side honeycomb}}$$

Finally, the crush force

$$F = (p_{\text{side honeycomb}})(A_{\text{side honeycomb}}) + (p_{\text{end honeycomb}})(A_{\text{end honeycomb}})$$

$$r = 22.8 \text{ in.} \quad p_{\text{end honeycomb}} = 296 \text{ psi}$$

$$h = 12 \text{ in.} \quad p_{\text{side honeycomb}} = 115 \text{ psi}$$

$$\beta = 40^\circ$$

Thus, δ appears when $\delta \approx 9$ in.

δ	A total	A side	A end	Total Force
3 in.	115 in. ²	--- in. ²	115 in. ²	34,100 lb
6	317	---	317	93,800
10	643	51	592	180,900
15	1066	375	691	247,600

The force versus deflection curve is plotted in Figure G-10. The 96,500 ft-lb of kinetic energy will be absorbed with an 11.6 inch corner deflection. The total thickness of honeycomb (in this orientation) is about 22 inches. We have reached only 53 percent of the thickness in crushing, considerably less than the 70 percent allowable. Extra honeycomb is available for impact at other angles where rotations, etc., introduce uncertainties into the analysis. The maximum g load in this case is the force at the 11.6 inch deflection divided by the weight, or 202,000/3212, or 62.9 g's.

In the cold condition, this honeycomb area would be at a temperature averaging about 0°F. The temperature factor would increase to about 1.07, increasing both crush pressures by the ratio of 1.07/0.96 or 1.11. The maximum g load in this case is about 66.2 g's.

The corner g load can be resolved into axial and lateral components.

$$g_{\text{axial}} = g_{\text{corner}} \times \sin 50^\circ = (66.2)(0.766) = 50.7 \text{ g's}$$

$$g_{\text{lateral}} = g_{\text{corner}} \times \cos 50^\circ = (66.2)(0.644) = 42.6 \text{ g's}$$

Neither the end nor side honeycomb adjacent to the inner container will crush since the g levels for crushing in these areas are a minimum of 133.7 and 135 g's, respectively.

Neither the side nor end honeycomb receives support from the other. In corner impact cases, these layers must be sufficiently rigid so that they crush locally rather than fail due to shearing or bending at overhanging ends.

Figure G-10. Force Versus Deflection at Corner

The bottom honeycomb has an 11 inch overhang. The shear force at the overhang can equal the crushing pressure x the overhang, and the moment can equal 1/2 of the pressure x the overhang length squared.

$$V = (725)(11) = 7980 \text{ lb/in.}$$

$$M = \frac{(725)(11)^2}{2} = 43,800 \text{ in. lb/in.}$$

The minimum shear strength of the honeycomb core is 390 psi. A 12 inch depth of core alone would carry a shear load of 4680 lb. The two facings have a total thickness of 0.2 inch. The shear strength of the facings is 23,000 psi. Thus, the facings can carry 4600 lb/in. shear. The combined strength of core and facings is 9280 lb/in. Thus, the shear load can be carried without any backup support behind the honeycomb.

The moment of inertia of the facings above is 7.02 in.⁴/in. The facing stress

$$\delta \text{ facing} = \frac{MC}{I} = \frac{(43,800)(6)}{7.02} = 37,400 \text{ psi}$$

The facings above can carry the bending load since the facing tensile strength is about 39,000 psi. The core will also add bending support as will the backing honeycomb.

The shear force and bending moment in the side honeycomb is:

$$V = (350)(5) \text{ or } 1750 \text{ lb/in.}$$

$$M = (350)5^2/2 = 4380 \text{ in. lb/in.}$$

The core will carry a shear load of (11)(228) or 2510 lb/in. The three 0.030 in. facings will carry (3)(0.030)(23,000) or 2070 lb/in. The total shear capability is 4580 lb/in.

The moment of inertia of the core alone is 1.81 in.⁴/in. The facing stress is

$$\delta \text{ facing} = \frac{MC}{I} = \frac{(4380)(5.5)}{1.81} = 13,100 \text{ psi}$$

This is acceptable in view of the 39,000 psi strength.

In the corner impact case, the bottom honeycomb provides the stiffness to prevent excessive bulging of the end plate. The moment at the center of the honeycomb plate (neglecting cover plate) from Reference G-7, assuming the entire inertial load is applied uniformly to the plate, is

$$M = \frac{3 + \mu}{16} r^2 P$$

P , in our case for 50.7 g's is

$$50.7 \times 3212/\pi \times 22.8^2 = 99.7 \text{ psi}$$

$$M = (3.3/16)(22.8)^2(99.7) = 10,700 \text{ in. lb/in.}$$

$$\sigma_{\text{facing}} = \frac{(10,700)(6)}{7.02} = 9140 \text{ psi}$$

This is well within the strength of the facing. No facing yielding will occur.

The total axial load in this case is 50.7×3212 or 163,000 lb. The shear load at the edge of the cover plate is $163,000/\pi \times 45.7$ or 1135 lb/in. The peak shear stress in the plate is $1.5 V/t = (1.5)(1135)/0.125 = 13,600 \text{ psi}$. This is well within the 25,000 psi shear allowable (yield) for CORTEN steel.

Radial loads due to bulging of the plate at local areas during impact will be taken at the lip. There are 120 bolts. The axial load per bolt will be $163,000/120$ or 1360 lb. The bolts are 0.375 UNC, and have a stress area of 0.0773 in^2 . The bolt stress is thus 17,600 psi, well within the 30,000 psi yield stress of commercial bolts.

d) Inner Container Load Transmission. The inner container must transfer the deceleration force from the honeycomb to the heat source after the 30 foot drop. It must not distort excessively and must provide near uniform sand pressure on the heat source. The kinetic energy at impact is absorbed by the honeycomb crushing and is dependent on crush strength and crush area. The container must, therefore, retain the crush area to provide the necessary energy absorption.

The stress level of the inner container during impact will not exceed yield except in local areas. During end impact, the heat source and support ring exert pressure on the sand which is transmitted downward to the inner container cover. Since the top and bottom covers are supported by the honeycomb, the pressure load is transmitted directly to the honeycomb. The angle of repose of sand for 1 g is approximately 33° (Reference G-8).

Thus, the inner container cylindrical wall is pressurized and hoop tensile stresses result. The internal pressure and the margins of safety are presented in Table G-1.

During lateral impact, load is transmitted from the heat source assembly and support ring through the sand to the honeycomb. However, the pressure directly below the heat source assembly is greater than that at the sides. The inner container must withstand these pressure variations in addition to transmitting the force. Again, the results are tabulated in Table G-1.

4. Heat Source Stress Analysis

A stress analysis of the heat source under sustained accelerations was performed using the FEATS computer program. The axial and lateral models are shown in Figures G-11 and G-12. The models use 338 and 208 elements, respectively. The four major materials, Inconel 718, 304 SS, copper, and U-8 moly, were appropriately represented in the model. The fuel capsule loading effects were included by adding a "dummy" low strength material of appropriate density. A suitable pressure load was applied at the outside surface of the heat source to counterbalance the inertia loading.

The output of the program includes stresses, strains, and displacements, both elastic or elasticplastic. This analysis remained entirely elastic at the maximum g loads. A summary of the maximum stresses for each of three cases is listed below for a 1 g loading.

<u>Maximum Effective Stress Per g</u>	<u>Bottom Loading</u>	<u>Neck Loading</u>	<u>Lateral Loading</u>
Inconel 718	15.00 psi	27.90 psi	15.40 psi
304 Stainless Steel	8.59	24.70	----
U-8 Moly	8.46	11.34	9.48
Copper	4.23	4.53	4.39

TABLE G-1
INNER CONTAINER IMPACT LOAD SUMMARY

<u>Loading Condition</u>		<u>Maximum Load</u>	<u>Margin of Safety</u>
Bottom Impact (152.5 g's)		Internal Pressure 727 psi	65%
Lateral Impact (122 g's)	Section at Heat Source	Tension = 4186 lb Moment = 300 in. lb	36%
	Section Adjacent to Heat Source	Tension = 3140 lb Moment = 270 in. lb	65%
Corner Impact (66.2 g's or 50.7 g's axial and 42.6 g's lateral)		Not Critical

Figure G-11. Mechanical Model of Heat Source (Axial)

Figure G-12. Mechanical Model of Heat Source (Lateral)

The various loads (worst temperatures) are summarized below:

	<u>Axial Acceleration</u>	<u>Lateral Acceleration</u>
Bottom Impact	152.5 g's	----
Top Impact	-152.5	----
Side Impact	-----	+122 g's
Corner Impact	+ 50.7	+ 42.6

It is now possible to determine directly the maximum stresses for all but the corner impact. In the corner impact, it is conservatively assumed that the maximum stresses due to axial and lateral loadings are additive directly. The true maximum will be less than this since the maximum stresses under axial and lateral loads do not occur at the same point.

<u>Maximum Effective Stress</u>	<u>Bottom Impact</u>	<u>Top Impact</u>	<u>Side Impact</u>	<u>Corner Impact</u>
Inconel 718	2285 psi	4250 psi	1880 psi	2071 psi
304 Stainless Steel	1310	3760	----	1253
U-8 Moly	1290	1730	1156	979
Copper	645	690	536	417

The strengths of the materials at temperature (1200°F) are as follows:

Inconel 718	125,000 psi (yield)
304 Stainless Steel	14,000 psi (yield)
U-8 Moly	24,600 psi (yield)
Copper	8,000 psi (ultimate at high strain rates)

Thus, none of the heat source components are loaded at levels approaching their strengths. No heat source damage is expected under any of the impact conditions.

5. Stress Wave Considerations

Most impact analyses deal in considerable depth with stress wave propagation in the impacting object. The subject shipping container differs from the typical impacting object

in that the object is surrounded by a light, low strength, crushable medium. When impact occurs, stress waves must first be initiated in the honeycomb and transmitted to the inner container through the honeycomb.

The stress wave magnitude that can be transmitted through the honeycomb is limited to the crush strength of the honeycomb. Honeycomb is generally thought of as a rigid, perfectly plastic solid in the design range (rectangular stress-strain curve). Since the material obviously is capable of some elastic strain, we will consider it to be an elastic-perfectly plastic material (inclined elastic stress-strain curve to the yield stress and a constant stress in the plastic range equal to the yield stress).

At impact, a compressive elastic stress wave equal to the crush stress propagates through the honeycomb at velocity C , where

$$C = \sqrt{\frac{E}{\rho}} \quad (\text{Reference G-9})$$

The honeycomb has the following values of

$$E = 0.25 \times 10^6 \text{ psi}$$

$$\tau = 0.00457 \text{ lb/in.}^3$$

$$\rho = 1.175 \times 10^{-5} \text{ lb sec}^2/\text{in.}^4$$

$$C = 146,000 \text{ in./sec or } 12,100 \text{ ft/sec}$$

$$\sigma_1 = 745 \text{ psi (highest crush pressure at inner container). For this portion of the analysis, temperature variation through the honeycomb has been neglected, resulting in a constant crush stress.}$$

A plastic wave would follow the elastic wave in a strain hardening material at a velocity equal to $\sqrt{\frac{E_p}{\rho}}$, where E_p is the slope of the plastic line. Since we have assumed that $E_p = 0$, the plastic wave does not propagate and the plastic damage will be confined initially to the portion of the honeycomb where the stress wave is initiated.

At the interface between the honeycomb and the inner container, the following relationships from Reference G-9 should hold for an elastic case, where the subscripts I, R, and T refer to incident, reflected, and transmitted waves, respectively:

$$\sigma_I + \sigma_R = \sigma_T$$

$$V_I - V_R = V_T$$

$$\sigma_T = \sigma_I \left(\frac{2}{1 + \frac{\rho_1 C_1}{\rho_2 C_2}} \right) \quad \sigma_R = \sigma_I \left(\frac{1 - \frac{\rho_1 C_1}{\rho_2 C_2}}{1 + \frac{\rho_1 C_1}{\rho_2 C_2}} \right)$$

$$\rho_1 C_1 \ll \rho_2 C_2$$

From the above relationship:

$$\rho_1 C_1 / \rho_2 C_2 \approx 0.0113$$

$$\text{Thus, } \sigma_R = 0.976 \sigma_I$$

Therefore, a reflected compression wave would return from the interface back through the honeycomb (of magnitude $0.976 \times$ the initiated compression wave) in an elastic case. The compressive stress would be the sum of $\sigma_I + \sigma_R$ or $1.976 \sigma_I$. But in our case, σ_I is

the crush strength of the material so that $\sigma_I + \sigma_R$ cannot exceed σ_I . Therefore, $\sigma_R = 0$, and the local plastic damage will occur in the honeycomb at this interface after the initiated wave reaches the interface. Therefore, $\sigma_T = \sigma_I + \sigma_R = \sigma_I$ since $\sigma_R = 0$. The stress wave that reaches the inner container cannot exceed the magnitude of the crush force of the honeycomb, approximately 754 psi. The tensile strength of the most critical component in the container, the shield, from Reference G-10, is a least 24,000 psi, more than an order of magnitude higher than the level of any pressure wave that will be transmitted into the container.

6. Puncture

The following analysis verifies the capability of the unit to withstand the hypothetical puncture accident.

In this postulated accident, the unit drops through a displacement of 40 inches onto a cylindrical mandrel. The kinetic energy at impact equals 10,700 ft lb. The mandrel diameter is 6 inches, and its area is 28.3 in.² The package must be capable of impacting with any side normal to the mandrel.

The outer end honeycomb crush pressure varies from about 710 to 760 psi, at low impact velocity, depending on temperature. The maximum penetration of the honeycomb would be 6.4 inches at the lower pressure, neglecting the energy absorbed in punching the shell. The maximum g load would occur with the higher pressure just as the shell is sheared.

The force required to punch the shell would be:

$$\begin{aligned} F_{\text{pen}} &= \pi d t \tau_{\text{(ult)}} \\ &= \pi \times 6 \times 0.125 \times 37,000 = 87,200 \text{ lb} \end{aligned}$$

The force crushing the honeycomb is

$$F = 760 \times 28.3 = 21,500 \text{ lb}$$

The total force applied to the unit is 108,700 lb, creating a g load of 33.8 g's.

This g load is less than that developed during the 30-foot drop case and the penetration is less than 70 percent of the honeycomb thickness. Therefore, the unit will survive this accident satisfactorily.

The outer side honeycomb crush pressure at low impact velocity varies from 329 to 367 psi. The g loading will thus be lower than that in the axial penetration accident. If we ignore the energy required to punch through the shell and assume a constant crush pressure, we obtain a penetration depth that exceeds the honeycomb thickness.

The unit cannot be shown to be adequate under the above conservative assumptions. It will be necessary to include the energy absorption in penetrating the shell and the "bottoming" of the honeycomb. The mandrel edge may have a 1/4 inch radius. This will prevent the outer shell from shearing until penetration is at least one shell thickness. Energy absorbed in denting the shell is still neglected.

The initial kinetic energy to be absorbed is 128,400 in. lb (107.00 ft lb). Figure G-13 shows the force as a function of mandrel penetration. After shearing the shell (assumed to shear in one thickness at 35,000 psi), the force drops to the crushing strength of the honeycomb, which is a minimum of $329 \text{ psi} \times 28.3 \text{ in.}^2$, or 9310 lb. After a penetration depth of 70 percent of the 11 inch thickness, or 7.7 inches, the honeycomb will begin to "bottom out" and the crushing pressure will increase.

Reference G-14 presents the load versus deflection curves for 5052 aluminum honeycomb (same material as this unit) through all three phases of compression. During the first phase, the load is a near constant value as the honeycomb collapses into well rounded folds. Then, these folds crush until they are completely flattened in phase 2. In the final phase, the honeycomb behaves as a porous material that is much stiffer. Compression in this phase produces a lateral material flow that increases cross-sectional area. The compression force curve in Figure G-13 has been extended beyond 7.7 inches, using the data presented in this reference.

The final penetration depth where the kinetic energy will be absorbed is about 9.7 inches. The crushing force at this depth is only about 85,000 lb, less than that required to penetrate the shell. The total force developed as the shell is sheared is the 82,400 lb shearing force plus the 9310 lb honeycomb crushing force, or 91,710 lb. This would increase to about 92,790 lb with cold honeycomb. This produces a 28.9 g lateral load that is acceptable since it is less than that developed in the 30-foot drop test discussed above.

Figure G-13. Force Required to Penetrate Shipping Container with 6 Inch Mandrel

D. SNAP-23A SYSTEM TRANSPORTATION ACCIDENTS

1. Impedance Of An Aircraft Fuselage

The impedance of an aircraft fuselage has been determined using fuselage crash test data reported by Greer, et.al. (Reference G-11). The predicted and actual responses of simulated aircraft fuselages are presented from impact tests at subsonic velocities onto various inclined media. The impedance has been derived from the bending moment due to impact versus fuselage station graphs and the fuselage deformation data. The local slope at each fuselage station has been determined by conjugate beam theory. The bending energy has been determined as the area under the moment-slope diagram. Finally, the effective spring constant or impedance of the fuselage has been derived after equating the bending strain energy to the tensile strain energy from axial impact.

The predicted moment versus station curve due to the impact of a representative fuselage model is shown in Figure G-14. The moment curve can be approximated by a triangular profile. The origin ($x=0$) and the end point ($x = 290$ in.) for this moment distribution correspond to stations 30 and 320, respectively. The loading is applied to a beam (the fuselage) whose length is 290 in. From Figure G-14, the peak of the triangular distribution is 1150 in.-kips. The assumption that the fuselage is a simply supported beam is justified in that failure of the fuselage due to impact is in the primary bending mode. The shape of the deformation versus beam distance curve for the simply supported beam is similar to the primary free-free bending mode shape exhibited by the models (Reference G-11, Figure 2.3). To find slopes (θ) of the deflection curve of the fuselage, the conjugate beam theory can be applied to the following approximate moment loading of the model fuselage:

$$\frac{M_o}{EI} = \frac{1150 \times 10^3 \text{ in. lb}}{10^7 \text{ psi} \times 1518 \text{ in.}^4} = 0.757 \times 10^{-4} / \text{in.}$$

$$\Sigma M_B = 0 \Rightarrow 290 R_1 = \left(\frac{M_o}{EI} \times \frac{155}{2} \right) \left(\frac{155}{3} + 135 \right) + \left(\frac{M_o}{EI} \times \frac{135}{2} \right) \left(\frac{2}{3} \times 135 \right)$$

$$R_1 = \theta_A = \frac{M_o}{290 EI} \left[77.5 (187.4) = 67.5 (90) \right]$$

$$\text{and } \theta_A = 0.538 \times 10^{-2} \text{ radians}$$

The reaction, R_1 , is equivalent to the slope at A, θ_A .

Now, solving for the second reaction, or the slope at B:

$$\theta_{155} = R_1 - \left(\frac{M_o}{EI} \times \frac{135}{2} \right) = 0$$

$$R_2 = 0.757 \times 10^{-2} \times \frac{1.35}{2} = 0.512 \times 10^{-2} \text{ radians}$$

The remainder of the slope diagram is developed and shown in Figure G-15 by computing the equivalent fictitious shear at various x distances. The curve is extended to station 0 ($x = -30$) and station 352 ($x = 322$), the fuselage end.

The bending moment versus slope or angular displacement is plotted in Figure G-16 using the data from Figure G-14 and G-15. Now, the total bending energy due to impact, the area under the $M-\theta$ diagram, is equated to the tensile strain energy $\frac{K \delta^2}{2}$ in order to establish the effective spring constant K of the fuselage:

$$\text{bending strain energy} = \int M d\theta \equiv \text{tensile strain energy} = \frac{K \delta^2}{2} = 14,350 \text{ in. lb}$$

Figure G-14. Comparison of Fuselage Bending Moment Due to Impact

The deformation δ is assumed to be the axial deformation the fuselage undergoes during axial impact. Again, from Reference G-11 the deformation due to crushing in the vicinity of the forward bulkhead is on the order of 2-1/2 in. Therefore, the effective spring constant or stiffness of the fuselage can be computed to be:

$$\int M d\theta = 14350 \text{ lb in.} = \frac{1}{2} K \delta^2 = \frac{1}{2} K (2.5 \text{ in.})^2$$

$$\text{or } K = 4600 \text{ lb/in.}$$

Lower fuselage shell load versus deformation curves have also been used to calculate K . These data bracket the derived value of 4600 lb/in.

The impact impedance parameter ρC for the aircraft fuselage can now be derived. Axial force is proportional to the resulting deformation:

$$F = K \delta .$$

$$\text{Substituting stress, } \sigma , \text{ and } \sigma = E \epsilon = E \frac{\delta}{l} :$$

$$\sigma A = K \delta$$

$$\frac{E \delta A}{l} = K \delta \quad \text{implies} \quad E = \frac{K l}{A}$$

By definition, the impact impedance equals the product of density and the sonic speed through the medium:

$$\rho C \equiv \rho \sqrt{\frac{E}{\rho}} \equiv \sqrt{E \rho} \quad \frac{\text{lb} \cdot \text{sec}}{\text{in.}^3}$$

Finally, substituting for E the parameter $\frac{K l}{A}$ determined above:

$$\rho C = \sqrt{\frac{K l \rho}{A}}$$

Figure G-15. Local Slope Along The Fuselage

Figure G-16. Bending Moment Due to Impact Vs. Slope for the Fuselage

To solve for the impedance of the fuselage, its composite density $\bar{\rho}$ must be determined. From Reference G-11, the model fuselage section weights and dimensions were used to compute the total weight and volume:

$$\text{total weight} = 1800 \text{ lb}$$

$$\text{volume} = 5.78 \times 10^5 \text{ in.}^3$$

Therefore, the average density of the aircraft fuselage is computed to be:

$$\bar{\rho} = \frac{Wt}{Vol} = \frac{1800 \text{ lb}}{5.78 \times 10^5 \text{ in.}^3} = 3.12 \times 10^{-3} \frac{\text{lb}}{\text{in.}^3}$$

Finally, the impact impedance parameter for the aircraft fuselage models:

$$\begin{aligned} \rho C &= \sqrt{\frac{K \bar{\rho}}{A}} = \sqrt{\frac{4600 \text{ lbf/in.} \times 352 \text{ in.} \times 3.12 \times 10^{-3} \text{ lbm/in.}^3}{386 \times \frac{\pi}{4} (48 \text{ in.})^2}} \\ &= 0.085 \text{ lb-sec/in.}^3 \end{aligned}$$

This magnitude indicates extremely high impact energy absorption capability.

The impedance derived for the crash model fuselage will have a similar magnitude for a full-size prototype fuselage. From Reference G-12, the elastic body $\sigma = E\epsilon$, and for the case of the same materials in the model and prototype: $\frac{\sigma_r}{\sigma_m} = \frac{\epsilon_p}{\epsilon_m}$ implied by $\frac{E_p}{E_m} \equiv 1$.

For a geometrically scaled model, both the total deformations and the lengths change in the same ratio and, hence, the strains are the same for model and prototype. Also for the case of the same materials:

$$\frac{(\text{Mass})_p}{(\text{Mass})_m} = \frac{(\text{Length})_p^3}{(\text{Length})_m^3} \quad \text{was implied by} \quad \frac{e_p}{e_m} = 1.$$

This results in the impedances of the model and prototype being equal, or

$$\frac{(\rho C)_p}{(\rho C)_m} = \frac{\sqrt{\frac{E_p \rho_p}{\rho_p C_p}}}{\sqrt{\frac{E_m \rho_m}{\rho_m C_m}}} = 1.$$

2. Impedance of Subassemblies

The general approach used for impact analysis of the SNAP-23A system is to concentrate on evaluating the integrity of the fuel capsule. The fuel capsule is assumed to impact on a composite non-rigid medium. From Reference G-13, the pressure at the capsule shield interface due to impact can be expressed as follows:

$$P = \frac{\rho_o C_o V}{1 + \frac{\rho_o C_o}{\rho_B C_B}} \quad (G-1)$$

where

P is the interface pressure or alternatively a limiting stress criterion

$\rho_o C_o$ is the impedance of SNAP-23A fuel capsule

$\rho_B C_B$ is the composite impedance of non-rigid receiver medium

V is the impact velocity

The product $\rho_B C_B$ is a composite of the individual impedances, including the U-8 Mo shield, generator structure, shipping container, and aircraft fuselage. Each impact case postulated utilizes a unique set of impedances in the determination of $\rho_B C_B$.

Equation G-1 can be manipulated and transformed into an impedance form as follows:

$$P = \frac{\text{Force}}{\text{Area}} = \frac{\rho_o C_o V}{1 + \frac{\rho_o C_o}{\rho_B C_B}}$$

and

$$P = \frac{\text{Force}}{\text{Velocity}} = \frac{\rho_o C_o A}{1 + \frac{\rho_o C_o}{\rho_B C_B}} \quad (\text{G-2})$$

Assuming several impedances are in mechanical series, the composite impedance across the system is:

$$Z = \frac{1}{\frac{1}{Z_1} + \frac{1}{Z_2} + \frac{1}{Z_3} + \dots + \frac{1}{Z_n}} = \frac{1}{\sum_{i=1}^n \frac{1}{Z_i}} \quad \text{for } n \text{ composites.} \quad (\text{G-3})$$

In series the displacements and velocities of masses are additive. For the case where $n = 2$,

$$Z = \frac{Z_1 Z_2}{Z_1 + Z_2} = \frac{Z_1}{1 + \frac{Z_1}{Z_2}}$$

Equation G-3 is similar in form to Equation G-2, and it follows that Equation G-2 can be generalized for n components into the following form:

$$\frac{A}{Z} = \sum_{i=1}^n \frac{1}{\rho_i C_i}$$

For a unit area, the impedance of the receiver medium of n components is of the form:

$$\frac{1}{\rho_B C_B} = \sum_{i=1}^n \frac{1}{\rho_i C_i} \quad (\text{G-4})$$

The impact impedances of several components of the SNAP-23A system are listed below.

Hastelloy-C Fuel Capsule:	$\rho_1 C_1$	=	139 lb sec/in. ³	at 1400°F
U-8 Mo Shield	: $\rho_2 C_2$	=	140 lb sec/in. ³	at 70°F
		=	40 lb sec/in. ³	at 1400°F
Copper Accumulator	: $\rho_3 C_3$	=	4 lb sec/in. ³	at 1200°F
Inconel 718 Stainless Steel Shell	: $\rho_4 C_4$	=	133 lb sec/in. ³	at 1200°F
SNAP-23A System Structure Surrounding the Heat Source	: $\rho_5 C_5$	=	(for conservatism, not included in overall Z)	

The impedance $\rho_6 C_6$ of a shipping container suitable for the SNAP-23A system can be synthesized from the overall impact analysis. The approximate impedance of an aircraft fuselage subjected to axial impact against several media was established in Section D.1 to be 0.085 lb sec/in.³

To make further comparisons, the impedances for Ottawa sand (soil) and granite are presented. A non-rigid impact medium would be densely packed Ottawa sand:

$$\rho_S C_S = \sqrt{E_S \rho_S} = \sqrt{\frac{9900 \times 0.0667}{386}} = 1.31 \text{ lb sec/in.}^3$$

A rigid impact medium such as granite gives

$$\rho_G C_G = \sqrt{E_G \rho_G} = \sqrt{\frac{7.3 \times 10^6 \times 0.0955}{386}} = 42.5 \text{ lb sec/in.}^3$$

However, in all cases considering impact on granite, $\rho_G C_G$ was treated as infinite in magnitude.

The composite impedance $\rho_B C_B$ of the receiver medium, with which the capsule interfaces during impact, can be determined for various postulated cases. The applicable component impedances are substituted into Equation G-4, yielding $\rho_B C_B$.

3. Fuel Capsule Stress Due to Impact

Consider in the following sketch, the capsule impacting with a non-rigid composite receiver:

From the conservation of momentum across the wavefronts of discontinuity,

$$A_0 U_0 = A_1 U_1 + A_2 U_2$$

where

U_0 is the initial impact velocity

U_1 is the particle velocity of the capsule at the impact interface

U_2 is the velocity of the receiver

The equation of motion for the capsule is

$$M_0 \frac{dU_2}{dt} + A_2 \rho_2 C_2 U_2 = 0$$

where

$A_2 \rho_2 C_2 U_2$ is the resistance of the receiver analogous to a spring force (kx)

Integrating between limits,

$$\int_{U_0}^{U_2} \frac{dU}{U} = - \frac{A_2 \rho_2 C_2}{M_0} \int_0^t dt$$

Now, letting $t = \frac{2l_0}{C_0}$, the time for an elastic wave to cycle the capsule, the solution is

$$U_2 = U_0 \exp \left[\frac{-A_2 \rho_2 C_2}{M_0} \left(\frac{2l_0}{C_0} \right) \right]$$

Now, for $M_0 = A_0 \rho_0 l_0$ and the assumption $A_0 = A_1 = A_2$,

$$U_2 = U_0 \exp \left[\frac{-2\rho_2 C_2}{\rho_0 C_0} \right]$$

Also,

$$U_1 = U_0 - U_2 = U_0 \left[1 - e^{\left(\frac{-2\rho_2 C_2}{\rho_0 C_0} \right)} \right]$$

The strain at the impact interface of the capsule is

$$\epsilon_1 = \left(\frac{U_0 - U_2}{l_0} \right) t = \frac{U_1}{l_0} \left(\frac{2l_0}{C_0} \right) = \frac{2U_1}{C_0}$$

The dynamic stress on the capsule is

$$\sigma = E_1 \epsilon_1 = \frac{2E_1 U_1}{C_0} = \frac{2E_1}{C_0} U_0 \left[1 - e^{\left(\frac{-2\rho_2 C_2}{\rho_1 C_1} \right)} \right]$$

Finally, substituting ρC^2 for E , the impact stress is determined to be:

$$\sigma = \rho_1 C_1 U_0 \left[1 - \exp \left(\frac{-2\rho_2 C_2}{\rho_1 C_1} \right) \right] \quad (G-5)$$

where the factor 2 has been dropped to yield an average stress.

For a composite receiver, $\rho_B C_B$ from Equation G-4 can be substituted for $\rho_2 C_2$. As $\rho_C C_C \rightarrow 0$ (soft), the impact stress of the capsule approaches zero because the composite receiver absorbs all of the impact energy.

The solution for impact stress is restricted to the first passage of the elastic wave resulting from impact. Reflections of the elastic wave and interactions with the plastic wave propagated from the impact surface would introduce considerable complexity into the analyses. The dynamic interface stress is determined for the primary impact. Secondary impacts or vibration of the capsule within the cavity of the U-8 Mo shield were not considered.

4. Capsule Failure Criterion

The fuel capsule stress due to impact and its mechanical properties at elevated temperatures can be used to formulate the parameters of an impact failure criterion. Based on test data, the criterion can be used to predict the passing or failure of thin and thick-walled cylindrical shells subjected to impact. The SNAP-23A fuel capsule is a thick-walled capsule. Details of the derivation of the impact criterion parameters and the correlation to test data on shells are found in Reference G-14.

Survival of the fuel capsule is defined as no penetration of the outer capsule wall.

The fundamental assumption associated with the impact failure criterion is that the ratio S of dynamic impact stress to shell ultimate stress is related to the ratio M of shell impact momentum to its momentum absorption capacity. This postulated failure criterion relationship was derived to be as follows:

$$S = f(M)$$

or

$$\frac{R}{h} \frac{v_f}{\sigma_u} \sqrt{E_p} = \epsilon_f \left[h^2 v_f \sqrt{\rho/E_{sh}} \right]$$

where

R is the mean shell radius

h is the shell wall thickness

v_f is the shell failure velocity

σ_u is the ultimate strength of shell material

E is the shell elastic modulus

ρ is the density of shell material

$$E_{sh} = 1/2 (\sigma_y + \sigma_u) \epsilon_f$$

σ_y is the yield strength of shell material

ϵ_f is the failure strain or ductility of the shell material.

The term $V_f \sqrt{E \rho}$ (or $V_f \rho C$), a stress parameter, is replaced by σ , computed from Equation G-5.

Figure G-17 is a graphical representation of the impact failure criterion for cylindrical shells. The solid line is an empirical fit through the data points. The curve represents shell failure. Region I is the safe impact or survival region, whereas Region II is the failure region. The capsule's fate is judged by the location of its "S" and "M" coordinates with respect to Region I or II. The range of variables S and M allows the application of the criterion to both thick and thin shells. The thick shells resist failure through a combination of membrane and bending resistance; the thin shells resist failure by undergoing large stretching deformations through membrane action.

5. Shield Failure Criterion

The capsule failure criterion defined in the previous section has also been used to make a rough estimate of the fate of the U-8 Mo shield after low velocity impacts. The ratio of shield radius to thickness is slightly greater than unity. There is negligible data on the fate of cargo in aircraft crashes at velocities above 250 knots (442 fps). Therefore,

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Figure G-17. Impact Failure Criterion for Cylindrical Shells

no confidence can be placed in estimates of the integrity of the massive shield after high velocity aircraft impacts. Enormous magnitudes of energy must be absorbed by the aircraft and component structures in these cases. Defining shield failure cannot be rigorous since the shield already has a joint. Failure is loosely interpreted to mean breaking the shield such that streaming radiation will occur.

6. Impact Cases

The computed results for the SNAP-23A system impact cases are presented in this section.

The impedances, stresses, and impact criterion parameters are summarized for each case in Table G-2. Numbering of the cases has been arbitrarily chosen in the order of increasing relative impact velocity. Case 1, presented in Table G-2, is particularly important because it represents an example of synthesizing the container impedance requirement from analysis. Further discussion on the requirements of the generator containment structure is presented in the next section of this appendix.

The case descriptions which follow represent the general approach to impact by combining all of the foregoing discussions in this Appendix.

Case 1 - The SNAP-23A System in a Cubic Container Impacting at Terminal Velocity onto a Non-Yielding Medium

Aerodynamic analyses reported in Section VI-C indicate that a 5-cubic foot aluminum container weighing 1620 lb yields a terminal velocity impact of 237 fps. Assuming that the container impedance is unknown, its ρC is neglected the first time through.

The composite impedance of the non-rigid shield and steel shell receiver media is computed from Equation G-4 to be $\rho_c C_c = 68 \text{ lb sec/in.}^3$. Capsule dynamic stress due to impact is determined from Equation G-5 to be 248,000 psi.

Referring to the impact failure criterion for cylindrical shells, the stress parameter S for the capsule is 17.7. The momentum parameter M is 0.072 in.^2 for Hastelloy-C at 1400°F .

From Figure G-17, the S coordinate is just below the horizontal line in the safe Region I for the M coordinate which is off scale. It may therefore be concluded that these coordinates result in a marginal survival of the capsule.

Had the container impedance been included, this case would pass with margin. Assuming the container impedance at temperature is on the order of $\rho_c C_c = 40$, the revised $\rho_c C_c$ is 25 lb-sec/in.³ and M does not change. The revised σ and S parameters, however, are 120,000 psi and 8.6, respectively. Therefore, the S coordinate is well within the safe Region I.

Case 2 - Failure Terminal Velocity for the Fuel Capsule Inside the SNAP-23A System and Its Container (Container $\rho C = 40$)

From Figure G-17 the stress parameter S is taken as the threshold value of 20 which divides the pass and fail regions. Then the corresponding stress on the capsule due to impact must not exceed 280,000 psi. The peak allowable terminal velocity of the container such that the capsule survives is computed to be 550 fps from Equation G-5.

Case 3 - The SNAP-23A System and Container in an Aircraft That Crashes into a Non-Yielding Medium During Landing (or Takeoff)

A maximum impact velocity of 130 knots (220 fps) is assumed since this represents the maximum approach speed of a typical jet aircraft (Boeing 707) during landing. The composite impedance of the non-rigid shield, steel shell, and the aircraft fuselage onto a granite medium is 0.0847. The low impedance of the aircraft fuselage completely dominates the composite one. The high energy absorption capability of the fuselage results in the low stress of 440 psi on the capsule wall due to the impact. The coordinates $S=0.0314$ and $M=0.067$ in.² are clearly in the safe region and indicate survival of the capsule for this case. At this low impact velocity, application of the impact criterion also indicates survival of the shield.

TABLE G-2
RESULTS FROM IMPACT CASES INVOLVING THE SNAP-23A
GENERATOR SYSTEM

Accident Cases	Configuration	Relative Impact Velocity (fps)	Impedances Comprising				Receiver Composite Impedance	Capsule Stress (psi)	Shield Stress (psi)	S	M in. ²	Capsule	Shield
			Shield	Steel Shell	Container	Aircraft							
1) Truck Head-on Collision	SNAP-23A System	180	140	133 133			88	280000	256000	20 20	>0.01 >0.01	Pass	Pass
2) Aircraft Crash During Takeoff or Landing	SNAP-23A System + Container + Air- craft	220	140	133	40*	0.085	0.0847	440		0.0314	0.067	Pass	Pass
3) Typical Terminal Velocity Impact After Midair Release	SNAP-23A System + Cubic Container	237	140	133	40*		25	120000		8.6	0.072	Pass	
4) Limiting Terminal Velocity Impact After Midair Release	SNAP-23A System + Container	550	140	133	40*		25	280000		20	>0.01	Pass	
5) Terminal Velocity Impact	SNAP-23A System alone	591	140	133			68	61900		44.2	0.18	Fail	
6) Aircraft Crash Enroute	SNAP-23A System + Container + Air- craft	843	140	133	40*	0.085	0.0847	1690		0.12	0.256	Pass	
7) Head-on Midair Collision Between Aircraft	SNAP-23A System + Container + Air- craft	1690	140	133	40* (2)	0.085	0.0424	1725		0.123	0.513	Pass	

* Estimated Upper Bound

BLANK

Case 4 - The Midair Release of the SNAP-23A System (Without Container) With Impact on a Non-Yielding Medium at Terminal Velocity

The terminal velocity of the 1220-lb SNAP-23A System has been determined to be 591 fps. The composite impedance of the material external to the capsule is 68 lb-sec/in.^3 . The capsule impact stress is 619,000 psi. The corresponding impact parameters are $M=0.18 \text{ in.}^2$ and $S=44.2$. The criterion indicates that the capsule would fail since these coordinates fall into Region II.

Case 5 - The SNAP-23A System and Container in an Aircraft That Crashes Head-On Enroute into a Non-Yielding Medium

The Boeing 707 cruising velocity of 575 mph (843 fps) is taken to be the impact velocity. The composite impedance is the same as for Case 3. The high energy absorption capability of the aircraft fuselage results in low stress, 1690 psi, at the capsule shield interface. The impact coordinates, $S=0.12$ and $M=0.256$, indicate that the capsule will survive.

Case 6 - The SNAP-23A System Inside an Aircraft Which Undergoes a Head-On, Midair Collision With Another Aircraft

The impact velocity is assumed to be the relative velocity, 1690 fps, between two Boeing 707's at their cruising speeds. The composite impedance consists of the non-rigid shield, steel shell, and aircraft fuselage impacting another fuselage. This impedance has been determined to be $0.0424 \text{ lb-sec/in.}^3$. The analytical model predicts that the two fuselages will absorb all of the energy of impact and yield a minimal impact stress on the capsule, $\sigma = 1725 \text{ psi}$. The criterion indicates that for these coordinates, the capsule safely survives the midair collision, assuming that the boundary remains constant at $S=20$ for this off-scale value of $M=0.513 \text{ in.}^2$.

Equation G-5 was used to estimate the impact interface stress between the container and fuselage. The computed loading against a side panel of the fuselage would probably

result in tearing out the panel and subsequent midair release. We may then revert back to Cases 1 and 2, where the generator in its container undergoes free fall to impact.

Case 7 - The Impact Velocity at Which Both Capsule and Shield Survive,
Considering Only the SNAP-23A System

The capsule threshold value for pass/failure corresponds to an S value of 20 (see Case 2). Considering the receiver medium to be the biological shield and outer steel shell only, the limit impact velocity for capsule survival is 268 fps. It has been determined, however, that the shield fails at a velocity of 268 fps. The critical impact stress on the shield for S=20 was estimated to be 256,000 psi. The corresponding velocity limit for the shield to pass is 180 fps. At this velocity, there is no question about the safe survival of the capsule. As for the shield, it could survive a 60 mph head-on collision between two trucks where the impact impedances of the truck structures and shipping container provide the required margin of safety. This would be a worst case truck accident.

7. Bounds on the Impedance of the Generator Shipping Container

In Cases 1 and 2, it was shown that a container whose impedance was 40 lb-sec/in.³ would assure survival of the capsule in some terminal velocity impact situations. In this section, discussion will show that this value is an estimated upper bound, i.e., $\rho_6 C_6 \leq 40 \text{ lb-sec/in.}^3$. For example, a 40 lb-sec/in.³ container impedance is conservatively hard when compared to the impedance of granite which is 42.5 lb-sec/in.³. It should be noted that granite is usually taken to be a non-yielding medium for which impedance is infinite. Thus, this magnitude of impedance for a container would imply high strength but low energy absorption capability.

If a cubic aluminum container is assumed to have the same spring constant as the aircraft fuselage (it would be expected to be stiffer), an approximate impact impedance for it would be

$$\rho_6 C_6 = \sqrt{\frac{KI}{A} \rho}$$

where

A, l is the area and length of the cube

ρ is the $\frac{\text{Weight}}{\text{Volume}}$ = average density of cubic structure

$$\rho_6 C_6 = \frac{4600 \text{ lb/in.} \times 60 \text{ in.} \times 1620 \text{ lb/in.}^3}{386 \times 60^2 \text{ in.}^2 \times 60^3}$$

$$= 0.04 \text{ lb-sec/in.}^3$$

This would be an extremely soft, energy absorbing structure. In order to yield $\rho_6 C_6 = 40$, $K = 46 \times 10^8 \text{ lb/in.}$ is implied. However, this is physically impossible to reach because it implies that the elastic modulus of the composite structure is well beyond aluminum and even steel. From Hooke's law, $\frac{P}{\delta} = \frac{AE}{l} \triangleq K$, a spring constant,

$$\text{or } E = \frac{Kl}{A} = \frac{46 \times 10^8 \text{ lb/in.} \times 60 \text{ in.}}{60^2 \text{ in.}}$$

$$= 77 \times 10^6 \gg E_{\text{Alum}} = 10 \times 10^6 \text{ psi}$$

Normally, the elastic modulus of a composite structure is less than the individual moduli of the components. Therefore, use of the container impedance equal to 40 represents an upper bound and yields conservative results in the impact analyses.

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- G-12 Harris, C. M. and C. E. Crede, Shock and Vibration Handbook, Vol. 2, McGraw-Hill, New York, 1961, pp. 27-7 and 27-10.
- G-13 Kornhauser, M., Structural Effects of Impact, Spartan Books Inc., Baltimore, Maryland, 1964.
- G-14 Haskell, D. F., "Impact Failure Criterion for Cylindrical and Spherical Shells," The Shock and Vibration Bulletin, No. 39, Part 5, December 1968, pp. 21-27.

APPENDIX H

AIRCRAFT ACCIDENT RISK

An analysis of aircraft accident data which yield accident occurrence probabilities is presented herein. Probability data are based on a series of documents published by the Civil Aeronautics Board for the period 1960 through 1962 and the National Transportation Safety Board from 1964 through 1967 entitled "Annual Review of U. S. Air Carrier Accidents." The accident per mile statistics used for SNAP-23A were selected from the category of aircraft called "nonscheduled revenue service - all cargo" defined in the references as follows:

"All cargo revenue flights that are not operated in regular scheduled service such as charter flights and all nonrevenue flights incident to such flights."

All aircraft accident data are summarized in Table H-1. The probability of an accident involving the generator and its shipping container in a chartered cargo flight was computed as 3.3×10^{-8} accidents/mile. This value is an average of the annual statistics tabulated for domestic carriers (2.7×10^{-8} accidents/mile) and international carriers (3.9×10^{-8} accidents/mile) since transportation of the SNAP-23A system by air is assumed to be from WANL to any site in the world.

Three specific accident types have been defined and comprise the total aircraft accident probability. The accident can occur:

- During takeoff or landing
- As any mid-air collision between aircraft
- As any other inflight or cruise accident.

TABLE H-1
AIRCRAFT ACCIDENT DATA

	<u>Category</u>	<u>Frequency Factor</u>	<u>Probability (Accident/ per Mile)</u>
A.	Turbojet and Piston Airplanes		
	1. All accidents for non- scheduled revenue service - all cargo.	1	3.3×10^{-8}
	a. domestic	-	2.7×10^{-8}
	b. International	-	3.9×10^{-8}
	2. Accident during takeoff or landing.	0.83	2.7×10^{-8}
	3. Mid-air collision with another aircraft.	0.063	0.21×10^{-8}
	4. All other inflight accidents (explosions, catas- trophic failures, hitting mountains, etc.)	0.10	0.34×10^{-8}
B.	Helicopter Domestic Service (all accidents)	1	0.76×10^{-6}

The takeoff or landing type of accident is the most probable. The mid-air collision between aircraft is the least probable. However, the mid-air collision is of the greatest interest due to the potentially high impact loads which can occur upon collision and/or upon the ensuing terminal velocity impact of the cargo. The frequency factor of each accident type is shown in Table H-1 for the nonscheduled cargo flights.

The frequency factors have been determined after reviewing the detailed accident resumes for each year of the nonscheduled revenue - all cargo carriers. For the seven years reviewed there were a total of 48 accidents by these special carriers. There were 40 accidents during the take-off or landing phases, five accidents en route or during cruise, and three mid-air collisions. In 18 cases, or 40 percent of the accidents, fire resulted. Fire was present in:

- (1) Thirty-two percent of the take-off or landing accidents
- (2) Sixty percent of the accidents en route
- (3) Two of the three mid-air collisions

The probability of the specific event occurring is also given in accidents/mile. Each was determined by multiplying the frequency factor by the total aircraft accident probability, 3.3×10^{-8} .

Deployment of the generator at or near the site may be by helicopter. Based on the annual statistics for the helicopter domestic carriers non-scheduled revenue service, the accident probability for this transportation mode has been computed to be 0.76×10^{-6} .

APPENDIX I

ANALYTICAL DESIGN OF THE BIOLOGICAL SHIELD

The biological shield of the SNAP-23A System was designed to meet standard radiation dose rate criteria with a minimum weight penalty. The majority of the shielding design effort has been reported in References I-1 and I-2. These references are also the source of the design methods and data reported in this Appendix.

The geometries involved in the SNAP-23A system dictated the use of a computer program with the capability of analyzing complex geometries. The QAD program was selected to perform the analysis because of its speed and flexibility.

The QAD Code was written to facilitate the calculation of both neutron and gamma fluxes, using both multi-energy groups and multi-regions. The gamma ray computations are based on point kernel ray tracing techniques with the use of polynomial buildup factors. The method of analysis involves representing the source by a number of point isotropic sources and computing the distance through all regions. Distances are computed using the line-of-sight from the source points to a desired dose point. From the distance traveled in each region and the material nuclear characteristics within the region, energy-dependent attenuation and energy-dependent buildup factors are applied. After the flux is converted to a dose rate, individual source points are summed into source-energy dependent dose rates.

The code requires the generator geometry to be described by zones (regions of homogeneous compositions). A zone is defined by listing the boundaries which surround it and may be described using up to six boundaries. The program can accept up to 50 homogeneous zones, 50 surfaces, 40 compositions, 20 materials, 8,000 source points, 30 energy groups, and as many receiver-points as desired.

Dose rate accuracy is directly related to the number of source points and the zone description. The number of source points and zones defined are also related to computer time required to calculate the dose rate at a receiver or detector point. For the subject analysis, the source was divided into 1,000 source points and each material was defined as

a new zone. It was felt that this arrangement provided a good compromise between accuracy and computer time.

The material properties of interest were the nuclear radiation spectrum from the SrTiO_3 heat source and gamma ray attenuation properties of SrTiO_3 , copper, and the U-8 Mo shield.

Strontium-90 decays by the emission of a high energy beta particle. The high energy betas (~ 2.27 Mev) produced result in high-energy bremsstrahlung. The nuclear properties of the SrTiO_3 used are listed in Table I-1. All photon emissions listed in the table are derived from Arnold, Reference I-3. All emission rates are calculated using Wyard's formulation for bremsstrahlung photon production.

Photo attenuation coefficients for the materials used in the RTG and the polynomial buildup factor used for the U-8 Mo are given in Table I-2. The polynomial buildup factor coefficients for depleted uranium were used to calculate buildup factors for the U-8 Mo shield.

The analytical model chosen for the detail shield study closely represented the actual generator assembly. Particular areas of interest include the shield contouring and copper stud on the heat source mockup.

The generator assembly was divided into two sections for analytical purposes, since the QAD program used to perform the shield study is not adapted to the investigation of flux flowing in negative directions. The generator assembly was divided into a top and bottom section. The top section used the bottom of the fuel pellet as a reference plane. The analytical models for the top and bottom sections are presented in Figures I-1 and I-2.

The shield contouring on the top section was simulated by using a segment of a right-circular cone. The slope of the cone walls was chosen to most nearly represent the shield contour. A sketch of the cone, the toroid used to describe the contoured section, and the equation for a circular cone are shown in Figure I-3. The point of intersection on the Z-axis and the angle α were chosen such that equal masses of U-8 Mo shield material lie on either side of the circular cone. The radius for the toroid was determined using the smaller shield thickness as a base and inscribing the radius tangent to both the end and side

TABLE I-1
NUCLEAR RADIATION PROPERTIES OF SrTiO_3

(Power Density = 0.85 watts/cc)

	<u>Maximum Beta Energy (Mev)</u>	<u>Average Beta Energy (Mev)</u>
Y-90	2.27	0.944
Sr-90	0.545	0.201

Photon Spectrum Due to Bremsstrahlung

<u>Photon Energy (Mev)</u>	<u>Photon Emission Rate (photons/watt . sec)</u>
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

TABLE I-2
NUCLEAR PROPERTIES OF MATERIALS
USED IN SHIELDING ANALYSIS

Uranium Dose Buildup Data*

Photon Energy (Mev)	β_0	β_1	β_2	β_3
0.25	1.0014	0.082	-0.001475	0.000543
0.50	1.0107	0.186	-0.001431	0.000488
0.80	1.0077	0.254	-0.00132	0.000418
1.10	1.0055	0.286	-0.001165	0.000349
1.40	1.0038	0.306	-0.000964	0.000290
1.70	1.0026	0.318	-0.000728	0.000242
2.00	1.00172	0.3262	-0.000466	0.000204

Mass Absorption Coefficients μ_m / ρ (cm^2 / gm)

Photon Energy (Mev)	SrTiO ₃	Hast-C	Ti	U-8Mo	Cu	SS	Min K
0.25	0.130	0.145	0.121	0.7603	0.1275	0.122	0.0
0.50	0.0848	0.0823	0.0860	0.1687	0.0820	0.0833	0.0
0.80	0.0674	0.0643	0.0684	0.0928	0.0654	0.0669	0.0
1.10	0.0583	0.0552	0.0600	0.0702	0.0563	0.0577	0.0
1.40	0.0515	0.0487	0.0530	0.0581	0.0497	0.0510	0.0
1.70	0.0468	0.0438	0.0480	0.0515	0.0441	0.0464	0.0
2.00	0.0432	0.0409	0.0442	0.0478	0.0418	0.0427	0.0

* Used to fit the dose buildup expression

$$B(\mu_0 r) = \sum_{i=0}^3 \beta_i (\mu_0 r)^i + E_i (\mu_0 r)$$

Figure I-1. SNAP-23A Shield Mockup, Top Section

Figure I-2. SNAP-23A Shield Mockup, Bottom Section

Equation for a cone with its axis parallel to the Z-axis

$$(X - X_0)^2 + (Y - Y_0)^2 - (Z - Z_0)^2 \tan^2 \alpha = 0$$

Figure I-3. Contour Mockup Using Conical Section

of the shield assembly. The approach leads to conservative dose rate predictions in the corner sections. To further contour the shield in the top and bottom corner regions three circular cones were used to describe each corner (top and bottom).

The angle between the side of the cones and the Z-axis and the point of intersection were chosen by producing a full scale drawing and overlaying the three cones on the shield assembly. The criterion used to select the point of intersection and the angle was that the volume of shield material on either side of the cone should be equal. No attempt was made to reduce the side or end shield thicknesses.

The major emphasis of the analysis was directed toward reducing the shield thickness in the contoured sections while still meeting dose rate criteria. Reduction in thickness was accomplished by decreasing the cone intercepts used to describe the section. This process was repeated until the desired dose rates were obtained for both the top and bottom shield assemblies. The new set of cones were then laid out on a full scale drawing and a best-fit toroid was constructed. The shield dimensions are shown in Figure I-4. The isodose plot for the shield dimensions was generated using the three cone approach. A list of materials and thickness used to produce the isodose plot is given in Table I-3. The dimensions given are for a thermal power of 1167 watts. The isodose plot for the shield design is shown in Figure I-5.

Figure I-4. Biological Shield, 60 Watt(e) System

**TABLE I-3
MATERIAL THICKNESS USED IN
SHIELDING ANALYTICAL MODEL**

Overall RTG Dimensions

Overall Radius = 30.45 cm

Overall Height = 69.40 cm

Fuel Capsule

SrTiO ₃ Pellets	Diameter	=	10.16 cm
	Length	=	17.16 cm
Hastelloy-C	Liner	=	0.15 cm radial
		=	0.15 cm axial
Hastelloy-C	Vessel	=	1.00 cm radial
		=	1.55 cm axial
Tantalum	Diffusion Barrier	=	0.13 cm radial
		=	0.13 cm axial

Shield

U-8Mo	Top Axial	=	7.15 cm
	Radial	=	8.00 cm
	Bottom Axial	=	8.28 cm

Copper Heat Accumulator

Cu	Radial Thickness	=	1.00 cm
	Bottom Thickness	=	0.65 cm

Copper Stud

Outer Copper Radius	=	4.25 cm
Height of Copper from Shield	=	14.34 cm
SS Accumulator Jacket Thickness	=	0.30 cm
SS Thickness on Top of Stud	=	5.40 cm

SS Accumulator Jacket	Radial Thickness	=	0.47 cm
	Axial Bottom Thickness	=	0.80 cm

Min-K

Axial Thickness		
On Top Stud	=	7.65 cm
Radial Thickness	=	13.95 cm
Bottom Axial		
Thickness	=	4.50 cm

Figure I-5. SNAP-23A Isodose Plot

REFERENCES

- I-1 "SNAP-23A Quarterly Progress Report, " May 25, 1967 through August 24, 1967, Subcontract No. 59-YJ-91605, Westinghouse-AEC Contract AT (30-1)-3800, HIT-265, August 1967.
- I-2 "SNAP-23A Quarterly Progress Report, " August 24 through November 24, 1967, HIT-281, November 1967.
- I-3 E. D. Arnold, "Handbook of Shielding Requirements and Radiation Characteristics of Isotopic Power Sources for Terrestrial, Marine, and Space Applications, " ORNL-3576, April 1964.