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SNAP-21 PROGRAM, PHASE II

DEEP SEA RADIOISOTOPE-FUELED
THERMOELECTRIC GENERATOR
POWER SUPPLY SYSTEM

DYNAMIC LOAD CRITERIA STUDY

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BY JH/abm/jamb
DATE 5/11/71

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Report No. MMM 3691-13

AEC RESEARCH AND DEVELOPMENT REPORT

This report has been prepared under contract AT(30-1)3691
with the U. S. Atomic Energy Commission

SNAP-21 PROGRAM, PHASE II**DEEP SEA RADIOISOTOPE-FUELED
THERMOELECTRIC GENERATOR
POWER SUPPLY SYSTEM****DYNAMIC LOAD CRITERIA STUDY****MARCH 3, 1967**

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DIVISION OF CLASSIFICATION
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1.0 SUMMARY

A study was conducted to establish the magnitude of dynamic loads that the SNAP-21 deep sea radioisotope-fueled thermoelectric power system will experience during handling, transportation, implantment and recovery. In addition, the ability of the current 10-watt SNAP-21 design to withstand these dynamic loads was analyzed and a parametric analysis conducted to determine what effect increasing system dynamic capability would have on system efficiency.

It was found that shock loads during handling in terminals and during implantment could potentially be as high as 200 g's. The magnitude of these loads clearly indicated the need for shock attenuating structures for certain operations. The maximum shock loads that the system would experience in situations where shock attenuation was not practical was approximately 6 g's.

Based on the above findings, it was recommended that the existing SNAP-21 design specification be changed from the current shock requirement of 3 g's to 6 g's. The only system component requiring major redesign to achieve this higher dynamic capability is the insulation system. A method of increasing the insulation system dynamic capability by supplementing the existing radiation shield support with three tension tie rods was recommended.

Design criteria for the shock attenuation capability of the shipping container and implantment structure were also established.

2.0 INTRODUCTION

This report describes a Dynamic Loading Criteria Study conducted by the 3M Company as a part of Phase II of the SNAP-21 development program.

The purpose of this study was to determine if the existing dynamic load criteria of 3g shock and vibration per MIL-STD-810A is adequate and, if not, what dynamic load criteria should be used for design of the system.

The study consisted of an analytical investigation of the dynamic loads that the SNAP-21 Deep Sea Radioisotope-Fueled Generator will experience during its operational lifetime; determining what effect these loads will have on the system, and what design modifications or ancillary equipment will be required to insure that the system can withstand its dynamic environment.

It was necessary to conduct a specific study for this purpose because existing standards commonly used to define environmental conditions were felt to be too conservative, especially for devices weighing over 400 lbs. For many applications the conservative structural design resulting from these existing standards is not objectionable, however, for the SNAP-21 system, overall efficiency is a function of dynamic capability. Consequently, to establish the optimum combination of dynamic capability and system efficiency, it was necessary to establish what dynamic capability is actually required.

A review of published data on dynamic environment during such operations as handling, shipping, implantment, and recovery revealed that the only well documented operation was shipment by common carrier. Data on the other operations either did not exist, was in the process of being documented, or consisted mainly of opinions and estimates by individuals working in the various

fields. In view of this lack of applicable data, and the overly conservative nature of existing standards, it was decided that the only way to establish realistic dynamic load criteria would be to conduct this study.

Because the results of this study were required to establish system design criteria, it was a program pacing item and it was necessary to complete it on a compressed time schedule. Consequently, some of the areas could not be investigated in the depth desired and some simplifying assumptions had to be made. It is felt, however, that this did not compromise the results of the study and that further effort would not materially affect the conclusions.

The remainder of this report is a detailed account of the study approach, results, conclusions, and recommendations.

3.0 OBJECTIVES

The objectives of the "Dynamic Loading Criteria Study" are:

- To establish a representative dynamic environment that the SNAP-21 system will be exposed to during its normal operational life,
- To analyze the effect of this environment on the existing 10-watt system,
- To investigate the feasibility of modifying the present 10-watt system to increase its dynamic capability,
- To determine what is required to insure that the SNAP-21 system can withstand its dynamic environment, including specifying the requirements of ancillary equipment such as shipping containers or implantment structures, and
- To recommend what dynamic loading criteria should be used for the functional design of the SNAP-21 systems.

It is emphasized that the objective of this study was to investigate only "operational" dynamic environments and not to consider abnormal or catastrophic events that would be defined as accidents. Therefore, the dynamic loading criteria that will be established as a result of this investigation will only be used for the operational or functional design of components and is not to be confused with the nuclear safety design criteria. These criteria will be established by existing specifications and the system safety analysis.

4.0 STATUS OF TECHNOLOGY

To establish a sound base for the study and to avoid duplication of effort, the status of SNAP-21 10-watt system components designed during Phase I were reviewed with respect to their capability to withstand dynamic loads. In addition, a review of the literature on dynamic loading was conducted.

The capability of the 10-watt system components to withstand dynamic loads is summarized below:

- Fuel Capsule

A structural design analysis has been conducted and verified by dynamic testing. The fuel capsule was found to be capable of withstanding operational dynamic loads far in excess of the 10-watt system load criteria since its design is governed by safety considerations.

- Segmented Hold Down Ring

A design analysis found the existing spring holding force marginal for the present dynamic load criteria of 3 g's. However, this component is not considered to be a major problem since it can be redesigned using conventional engineering methods, data, and materials.

- Insulation System

- Outer Case

The results of a design analysis indicate that the case is capable of withstanding the existing dynamic load criteria (3 g's shock and vibration per MIL-STD-810A).

Neck Tube (Inner Liner)

A design analysis has been conducted which indicates that the existing monocoque neck tube can be redesigned to withstand the current dynamic load criteria at the expense of approximately 19 watts heat loss above the design goal of 30-watts. This significant increase over the design goal, and over that of early estimates made during Phase II, is the result of a more detailed analysis which considered such effects as neck tube shear deflection and the effect of increased fuel and shield weight required due to the increased heat leak. Approximately 10 watts of this increased heat leak is due to the effect of the 1.5 safety factor applied to the structural design which 3M felt necessary.

Insofar as the availability of data on dynamic loading is concerned, it was found that published data on handling, implantment, and recovery was very scarce and incomplete, while data on transportation by common carrier was well documented. An especially valuable source of data on dynamic loading during common carrier transportation was a report by the "General American Transportation Corporation" titled "Shock and Vibration Transportation Environmental Criteria".⁽¹⁾

To assist in identifying various system components as they are discussed in this report, a sectioned view of the 10-watt system is shown in Figure 1.

Figure 1. 10-Watt Preliminary Design

5.0 TECHNICAL APPROACH

The technical approach established to conduct this study was a four-step approach that closely parallels the basic study objectives. These four steps are summarized below:

- The representative dynamic environment (shock and vibration profile) that both the 10- and 20-watt systems will experience during normal operational life was defined,
- The effect of this environment on the current 10-watt design was analyzed,
- The feasibility of modifying components of the 10-watt system to withstand the environment was investigated and the magnitude of weight and efficiency penalties were assessed, and
- Design criteria were established for shipping and implantment equipment found necessary as a result of the above.

Recommendations were then made for specification changes and handling requirements based on the report conclusions. The remainder of this report consists of a description of the study effort and its conclusions.

5.1 DEFINING THE REPRESENTATIVE DYNAMIC ENVIRONMENT

The most important step in the technical approach was to establish the representative dynamic environment that the SNAP-21 system will be exposed to during its normal life cycle. This was done by tracing the system through the handling, shipping, implantment, and recovery sequence that it will experience during its anticipated operational life.

The system life cycle was divided into four phases to provide convenient work segments for simultaneous study by separate individuals. These phases are:

- Assembly and Handling
- Transportation
- Implantment
- Recovery

The dynamic environment definition for each of these phases consisted of vibration frequency and amplitude and shock magnitude.

It should be noted that the examples used in this study are not intended to be all-inclusive, but are intended to serve as representative of the situations that could occur during the operational life of the system.

As the first step in defining the representative dynamic environment, baseline configurations were established for the 10- and 20-watt systems. This was done because the dynamic environment experienced by an object during shipping and handling is a function of its size and mass. The baselines selected were the 10- and 20-watt systems with and without a shipping container. Size and weight of the systems were taken as the specification requirements while the size and weight of the shipping containers were an engineering estimate.

The baseline configurations are presented in Table 1.

5. 1. 1 ASSEMBLY AND HANDLING LOADS

Assembly and handling loads, for purposes of this study, are defined as the dynamic loads that the complete system, in the packaged and unpackaged conditions, will experience during fabrication, assembly, and handling, up until the time the packaged system is secured in a common carrier vehicle. Dynamic loads that the insulation system (when considered as a separate component) will experience during this sequence were also considered. This was done because the existing insulation system was recognized to be the most sensitive system component from a dynamic capability standpoint.

Table 1. Baseline Configurations for Dynamic Loading Criteria Study

Configuration	Size and Shape	Weight
10-Watt System	16 in. dia. x 35 in. long (hemispherical ends)	600 lbs.
10-Watt System (with container)	60 in. square x 66 in. high (box shape)	1000 lbs.
20-Watt System	18 in. dia. x 32 in. long (hemispherical ends)	800 lbs.
20-Watt System (with container)	60 in. square x 66 in. high (box shape)	1200 lbs.

Components other than the insulation system were not considered separately in this study because they are either inherently insensitive to dynamic loading or because they can be easily protected by good manufacturing and handling practices and procedures.

The entire subject of dynamic loads experienced during assembly and handling was found to be very poorly documented. Personal contact with individuals in this field⁽²⁾⁽³⁾ and reference to the literature⁽⁶⁾ revealed that the design criteria used for many devices are existing specifications, such as MIL-STD-810A, which is extremely conservative, especially for objects weighing over 400 lbs. Sandia Corporation reported that they have a current project to determine what dynamic loads are experienced by objects during terminal handling; however, this data will not be available for several months. In addition, the General American Transportation Corporation has a current NASA contract to survey and compile all available data on terminal handling environment. This data will also be available in 3-4 months.

The only documented data found on terminal handling loads was that recorded on a 9,150 pound electronic shelter that was being skidded across a warehouse floor by forklift⁽⁵⁾. The maximum shock loads recorded were 3.18g vertical and 2.55g horizontal. However, due to the large difference in weight between this device and the SNAP-21 system, it is felt that this data is not directly applicable and can only serve as a rough indicator.

In view of this lack of data, dynamic loads during system and component assembly and handling were estimated. The method used to estimate these loads was to establish the operational sequence of steps for each manufacture, assembly or handling operation. The tooling that will probably be required for each operation was also described in general. Each step of the operation was then analyzed for potential maximum dynamic loads, barring accident. The loads estimated for handling and assembly are those that may be encountered with good, though not unreasonable, control, using the equipment and tooling described.

Dynamic loads experienced by the system during handling in transportation terminals was estimated on the basis of wood and metal containers for both supervised and unsupervised handling.

The g loads for those operations were calculated by estimating velocities and weights of impacting members and then calculating the change in velocity of the system member caused by the impact. From these data an equivalent drop height "S" is calculated from the relationship $S = V^2/2g$. By reference to Harris and Crede "Shock and Vibration Handbook"⁽⁴⁾, Figure 41.1, and using rise time values from Table 41.1, the g-loadings were established.

These estimated g-loadings are maximum expected values. No attempt was made to estimate frequency of occurrence because it was expected that the loads would be infrequent, random occurrences. It therefore seemed obvious that since they can occur, the system must be designed for these loads or they must be attenuated through proper packaging or handling.

In most cases, the difference between the 10-watt and 20-watt system baseline configurations were not significant enough to affect the assembly and handling of dynamic loads. In those instances where a difference did exist, the configuration which resulted in the worst load was used to establish the dynamic criteria.

The system and component assembly and handling analysis is shown in Appendix I. The analysis is presented in the form of partial process routing sheets that delineate the sequential steps of each operation and the estimated dynamic loads. The operations pertaining to the insulation system, as a separate component, are not included in Appendix I since they were the subject of a separate analysis.

One area of handling not considered is handling from the time the system is placed aboard ship until it is ready for implantment. This would include handling and moving the container aboard ship, removing the system from its shipping container and placing it in the implantment structure. It is anticipated that the proper combination of container design, handling procedures and personnel instruction will insure that the loads imposed during these operations will not be greater than the handling and assembly loads ashore. However, it would obviously be advantageous to perform a specific study on shipboard handling when detailed information is available on the type of ship, handling equipment, and implantment structure that will be used.

The only documented data that was obtained on shipboard handling was some recorded data obtained while handling aircraft jet engines aboard a carrier.⁽⁷⁾ This data is shown in Table II. An interesting aspect of this data is that it substantiates the study findings that high g loads can be experienced during transit over rough surfaces while on a steel wheeled dolly or other solid wheel vehicle. The information provided by personnel of the Naval Electronics Laboratory at San Diego, California on dynamic loads due to handling aboard ship by winch, boom, lifting arm, etc., substantiated this data. They reported loads of 2-3 g for these operations.

The problem of dynamic loads during handling aboard a submarine were also investigated. Again, documented data could not be obtained, consequently the only source of information was personal discussions with individuals experienced in this field. Several persons were contacted in the Submarine Division of BUSHIPS. Their comments are summarized as follows:

- Handling of the SNAP-21 system aboard a submarine would probably be conducted in the same manner as a torpedo; i. e. , it would be lowered with block and tackle through the hatch and then manually placed on a handling dolly for further movement aboard the submarine.
- A special handling dolly aboard the submarine would be required, or an adapter for the existing equipment.

Table II. Dockside and Shipboard Handling and Transport Shock and Vibration Measurements

Vibrations and shocks were measured during handling and storage of airplane engines aboard the USS MIDWAY. Engine containers used were manufactured by: (A) Firestone; (B) Edo; (C) Pratt and Whitney. Engines and containers were instrumented with Statham accelerometers, a 12 channel Model 104A Consolidated bridge balance, an 18 channel Model 5-114 Consolidated recording oscillograph, a type JC-1 Westinghouse vibrometer and an Askania vibrograph.

This Table summarizes the recorded shock peaks (in G) on the containers during handling.

Container Movements	Container	Shock Peaks (g)		
		Vert.	Trans.	Long.
Fork lift on flight deck crossing 1 inch diameter cable. a. Pneumatic Tires	A	0.52	----	----
	B	----	----	1.92
	C	1.05	1.03	1.10
	A	4.17	5.0	2.04
	B	----	7.13	----
	C	1.81	2.32	1.69
Lowering from flight deck into barge	A	3.59	----	0.94
	B	3.44	2.39	1.37
	C	No Records		
From pier to shipping depart- ment on dolly	A	----	6.8	4.88
	B	3.4	2.13	2.47
From shipping department to overhaul shop on trailer truck	A	----	2.93	0.55
	B	1.04	1.11	1.32
	C	0.63	0.61	1.71
From storage compartment to hangar deck	A	----	4.53	4.37
	B	2.19	1.75	3.66
	C	1.15	1.73	1.35

- Some special training will have to be provided for the crew.
- They are used to handling radioactive devices; therefore, this feature of the device would cause no problem, nor would it insure careful handling.
- The dynamic capability of torpedoes is classified; however, it is considerably above 6 g's. It was felt, however, that dynamic loads could be kept below 6 g's if the above conditions were met.

One of the major sources of dynamic loads during handling is the possibility of dropping the packaged device during terminal handling. Because the purpose of this study was to determine dynamic load criteria based on operational loads and not accidents, it was necessary to define what is an accident and what is normal handling with respect to drop height. Obviously, the best way to determine this would be to monitor a large number of operations and then, based on a statistical analysis of these data, determine what is the norm and what is random. Because this type of data was not available, a definition of normal drop height for supervised and unsupervised handling was made based on available data, discussions with individuals in this field, and on engineering judgment. A drop height of 6 inches has been commonly used as the dividing line between accident and normal; however, individuals in this field⁽³⁾ stated that in their opinion this figure was rather arbitrary and may not be the actual demarkation line. The Shock and Vibration Handbook⁽⁴⁾ pp. 41-4 states "Finally, very heavy packages weighing over 1000 lb. would be handled by heavier equipment with correspondingly more skill; any drops would be from very small heights." Based on the above, and on certain design features that will be incorporated in the shipping container, the following drop heights were established as the maximum for normal handling:

Unsupervised - 6 inch corner drop, 3 inch flat drop

Supervised - 1/2-inch corner drop, 1/4-inch flat drop

A separate dynamic load study was conducted on the insulation system, as a separate component, by the subcontractor for that component, the Linde Division of Union Carbide. The report on this effort is included as Appendix II to this report. Much of the insulation system handling will be done by fork lift, and since documented data on this subject could not be found, Linde conducted some actual tests as part of their study.

These tests were designed to simulate conditions during handling of the insulation system. In these tests, a simulated insulation system was secured to a wood pallet and the pallet was intentionally rammed by a motorized fork lift at various velocities and dropped, while on the fork lift, from various heights. The simulated insulation system was instrumented with accelerometers to provide a shock profile. Although these tests were limited in scope, the results compare favorably with the loads estimated in this report, therefore, it is felt that they substantiate the analysis technique. Data on these tests are included in Appendix II.

A summary of the estimated dynamic loads for all handling operations is shown in Table III.

It should be noted that assembly and manufacturing loads are not included in this summary. Although these loads were estimated and are shown in Appendix I, they were not used to establish design criteria since it was felt that these loads could be controlled by proper tooling and procedures. One area that may not be easy to control is removal of the system from the shipping container and placing it in the implantment structure. This operation would probably occur aboard ship. Since data on this type of operation is not available it is assumed that the loads experienced would be the same as for installing the system in the shipping container. The load for this operation as estimated in Appendix I is 4 g.

As shown in Table III, the maximum vertical shock condition is due to free fall (flat drop) of the system in a steel container with the following values:

- 3 inch fall – unsupervised (100 g)
- 1/4 inch fall – supervised (28 g)

The maximum horizontal shock condition occurs when the system in a metal container and on a metal skid is rammed by a motorized fork lift:

- Unsupervised (8 ft/sec) 200 g
- Supervised (1 ft/sec) 20 g

Table III. Estimated 'G' Loading for Various Means of Handling and Shipping of Insulation System, Fueled Insulation System and Complete System

Means of Transport and Description of Operation	System in Wood Container on Wood Pallet		System in Metal Container on Metal Skids		Remarks
	Unsupervised	Supervised	Unsupervised	Supervised	
	Acceleration g's	Acceleration g's	Acceleration g's	Acceleration g's	
1. Fork lifts - Motorized					
a. Ramming Skid	16	3	200	20	8 fps, unsupervised; 1 fps supervised Ref. 4, Fig. 41.1 Ref. Appendix B.
b. Drop to floor	6	1	25	12	1 fps unsupervised; 1/4 fps supervised Ref. 4; Fig. 41.1 Ref. Appendix B.
c. Jerking load on forks	2	1	2	1	Ref. Appendix B - because of cushioning of lift truck, container not important.
d. Transit over rough surfaces	10	2	10	2	Ref. Appendix B - because of cushioning of lift truck, container not important.
2. Fork Lifts - Manual					
a. Ramming Skid	3	1	20	3-1/2	2.5 fps fork lift speed unsupervised. 1/2 fps fork lift speed supervised (Assuming weight fork lift 1/2 weight of load. Ref. 4, Fig. 41.1.)
b. Drop to floor	6	1	25	12	Considered same values as motorized unit.

Table III. Estimated 'G' Loading for Various Means of Handling and Shipping of Insulation System, Fueled Insulation System and Complete System (Continued)

Means of Transport and Description of Operation	System in Wood Container on Wood Pallet		System in Metal Container on Metal Skids		Remarks
	Unsupervised	Supervised	Unsupervised	Supervised	
	Acceleration g's	Acceleration g's	Acceleration g's	Acceleration g's	
c. Transit over rough surfaces	11	6	58	29	1 in. vertical drop, unsupervised. 1/4 in. vertical drop, supervised. Ref. 4, Fig. 41.1.
3. Hand Truck, Rubber Wheels					
a. Ram into Solid object	7	1-1/2	70	12	3 fps ram speed unsupervised. 1/2 fps ram speed supervised. Ref. 4, Fig. 41.1.
b. Transit over rough floor	10	5	17	8	2 in. drop unsupervised; 1/2 in. drop supervised, assuming 0.008 sec. rise-time plus 0.010 wood container and 0.002 sec. metal container rise-time; Ref. 4, Fig. 41.1.
4. Hand truck, Steel Wheels					
a. Ram into solid object	7	1-1/2	70	12	Assumed same values as rubber wheel truck.
b. Transit over rough floor	15	7	40	20	2 in. drop unsupervised, 1/2 in. drop supervised, assuming 0.002 sec. rise-time plus 0.010 wood container and 0.002 sec. metal container rise-time; Ref. 4, Fig. 41.1.

Table III. Estimated 'G' Loading for Various Means of Handling and Shipping of Insulation System, Fueled Insulation System and Complete System (Continued)

Means of Transport and Description of Operation	System in Wood Container on Wood Pallet		System in Metal Container on Metal Skids		Remarks
	Unsupervised	Supervised	Unsupervised	Supervised	
	Acceleration g's	Acceleration g's	Acceleration g's	Acceleration g's	
5. Elevator	2.5	2.5	2.5	2.5	Emergency deceleration Ref. 4, Table 44.5.
6. Hoist, shop type (chain, elec. hydraulic) a. Setting down on solid floor	1-1/2	1-1/2	8	8	1/2 fps velocity both supervised and unsupervised, Ref. 4, Fig. 41.1.
7. Winch - dockside and aboard ship	3	3	3	3	Table II
8. Free fall to normal mishandling	20	6	100	28	6 in. edge drop (3 in. free fall) unsupervised 1/2 in. edge drop (1/4 in. free fall) supervised. Ref. 4, Fig. 41.1.

NOTES FOR TABLE III

1. G-loadings are dynamic loads. Static loads must be added to them where applicable for design purposes.
2. Wood pallet container rise-time: 0.010 second vertical; 0.020 second horizontal (Ref) Time measurement on Linde's test charts, Appendix I. Load magnitude checked against Ref. (4) Figure 41.1 for correlation. Metal container rise-time: 0.002 second, Ref. (4) Table 41.1.
3. Equivalent free fall distance for a known velocity was established from the formula $s = V^2/2g$.
4. Definition:
 - a. Supervised: Requires inspector to observe and control the handling methods and equipment used in load/unload operations with the transport vehicles - does not necessarily require presence of inspector during the actual transport by common carrier.
 - b. Unsupervised: No inspector on-hand to control handling.

A general comment worthy of mention is that all individuals with shipboard handling experience that were contacted emphasized that the human element was most important in safe handling. They felt strongly that proper training and supervision were just as important as proper container design, etc.

5.1.2 TRANSPORTATION LOADS

The second phase of the system life cycle, from a dynamic environment standpoint, is transportation by common carrier to its destination. Data for this portion of the report is very well documented, especially in a report titled "Shock and Vibration Transportation Environmental Criteria", prepared by the MRD Division of the General American Transportation Corporation under MFSC Contract NAS 8-11451 (Reference 1).

Data for the above report was obtained from over three hundred reports and contacts with 55 agencies and organizations active in the transportation field. Consequently, the report is quite lengthy and contains extensive graphic data.

It would be impractical to include all of this data in this report, therefore only a summary of the applicable data will be presented.

The major type of dynamic loads occurring during common carrier transportation is vibration. A graphic summary of the maximum vibration acceleration envelopes plotted as a function of frequency for aircraft, train, truck and ship transportation is shown in Figures 2, 3, 4 and 5 respectively. These graphs are excerpts from Reference (1). Table IV is a summary of the maximum vibration environment that can be expected. It is emphasized that the data shown is the response of the cargo floor; i. e., it is the input to the cargo. No attempt was made to define transfer functions or cargo response relationships. The actual loads experienced by the cargo due to these inputs will be a function of the cargo mass, since the heavier the cargo is, the more energy is required to drive it. Thus for relatively heavy cargo, such as the packaged SNAP-21 systems, the actual package response will be something less than the input. The degree of attenuation will be a function of the available energy in the driving medium, thus for a relatively light, flexible vehicle, such as a helicopter, the degree of attenuation would be greater than for a ship.

Figure 2. Aircraft Acceleration Envelopes - Composite (Propeller, Helicopter, Jet)

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Figure 3. Railroad Acceleration Envelopes - Directional Composites (Vertical, Lateral, Fore/Aft)

Figure 4. Truck Acceleration Envelopes - Road Condition Composites (Paved and Rough Roads)

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Figure 5. Ship Acceleration Envelopes - Composites (Slam and Emergency Maneuvers; Maximum Vibration)

Table IV. Transportation Vibration Summary

Transportation Mode	Acceleration Magnitudes (G's) at Indicated Frequencies		
	5 cps	50 cps	Highest Peak at N cps
<u>RAILROAD</u> 1) over-the-road	2.5	1.5	5.0 at 100
<u>AIRCRAFT</u> 1) propeller 2) jet 3) helicopter	unavailable unavailable 0.1	1.0 0.1 1.5	6.0 at 400 2.5 at 800 7.0 at 400
<u>SHIP</u> 1) maximum vibration 2) slam and emergency	0.09 2.0	0.5 3.0	0.9 at 12 3.0 at 2.5 and 50
<u>TRUCK</u> 1) paved road 2) rough road	0.3 1.7	0.2 0.3	0.35 at 4 2.0 at 10

NOTE: Data shown includes only response of the cargo floor (i.e., input to cargo) when cargo is relatively heavy, loads will be reduced.

"The severest shock and vibration environment associated with transport vehicles occurs during railroad car coupling or humping operation"⁽¹⁾. In view of this, the dynamic loads due to humping were investigated separately. Reference (1) described the shock and vibration environment for railroad coupling and humping operations as shock spectra. A shock spectrum is defined as the maximum acceleration of a series of single degree of freedom systems resulting from a specific shock excitation. The shock spectra for coupling at 8 mph is shown in Figures 6 and 7 for the vertical and fore/aft directions respectively. The data at 8 mph was selected because data on the distribution of coupling speeds shows that this will encompass 90% of all occurrences. This is shown in Figure 8 which is reproduced from Reference (1).

As shown in Figures 6 and 7, the dynamic response of a system is greatly affected by the natural frequency of the system and, to a lesser extent, by the percent of damping in the system. As shown in Figures 6 and 7, the dynamic loads can be kept below 8 g if the system natural frequency is limited to 5 cps and is 5.0 percent damped.

It was originally intended to investigate the transportation dynamic loads on a statistical basis, however, the paucity of data prevented this technique. The technique used in Reference (1), and which was used for this report, of enveloping all data lends itself to a conservative environmental definition with a very high (99 percent) statistical confidence level⁽⁸⁾. In view of the relatively low level of dynamic loads determined by this approach, as compared to other phases of the system life cycle, and the desired high reliability of the system, it is felt that these data should be used as design criteria.

The problem of dynamic loads during transportation was also investigated by the Battelle Memorial Institute as part of this study. The results of their effort are shown in Appendix IV. Although some of their data is slightly different from that shown above, it is felt that this is due to different references and assumptions.

5.1.3 IMPLANTMENT LOADS

The lack of documented data on dynamic loads during ocean implantment was quite evident after contact with numerous agencies involved in oceanographic work and actual ocean implantment (Appendix III). The most common response

Figure 6. Railroad Coupling Shock Spectrum - Standard Draft Gear (8.0 mps, Vertical)

Figure 7. Railroad Coupling Shock Spectrum - Standard Draft Gear (8.0 mph, Fore/Aft)

Figure 8. Railroad - Distribution of Coupling Speeds

of these sources, with respect to payload dynamic environment during implantment, was that they routinely overdesign their components so that dynamic loading will not be a problem. This is a logical and expedient solution when weight or efficiency are not design considerations, however, it is not practical for the SNAP-21 system.

A report by the Naval Civil Engineering Laboratory (Reference 9) compared actual cable dynamic load data recorded during the implantment of a Submersible Test Unit (STU) with analytical data. These data showed fair comparison between actual and analytical data; however, at certain depths the actual stresses recorded exceeded those calculated by a factor of 2. The STU is a large steel framework with an in-water weight of approximately 6000 lbs.⁽¹⁰⁾ The measured data for this unit being implanted in 6-10 foot swells with periods of 10-12 seconds revealed dynamic cable loads up to 2 g's. Because of the large differences in size and configuration between the STU and the SNAP-21 systems, it is felt that this data is not directly applicable to this study.

The most applicable data that was found on dynamic loads experienced by a payload while underwater and connected to a surface vessel was obtained by private communication with individuals at the Scripps Institute of Oceanography. They reported that during a bottom mapping project carried out over the past six years, they have measured actual dynamic loads experienced by a deep-tow echo finder while it was being towed by a 100 foot long converted harbor tug. During these instrumented tests, an accelerometer was mounted on the end of the boom supporting the cable and another in the echo finder. In this way the tow ship dynamics could be correlated with the payload dynamics. Details of these data are as follows:

- Echo Finder Size – 18 inch dia. x 40 inches long
Weight – 1200-1500 lbs.
- Towing Attitude – Cable attached at midpoint of 40 inch length
- Towing Speed – 2 knots
- Cable Size and Material – 3/8 inch dia. steel (10,000 lbs. test)
- Length of Cable – 5,000 ft. to 30,000 ft.

- Depth of Echo Finder – 5,000 to 15,000 ft.
- Sea Conditions – Varied, ranging from calm to Sea States 5-6.
- Maximum Accelerations – On boom = 0.75 g
On payload = 1/10 of boom loads.

This data indicates that dynamic loads on a submerged payload attached by cable to a surface vessel are quite low. It is felt that these low dynamic loads are due in part to the fact that the payload was being towed which would create a definite catenary in the support cable. This catenary would serve to damp out any cyclic loads on the system and thus prevent the possibility of resonance when the natural frequency of the cable – payload system coincides with the motion of the support vessel.

Although the above data is partially related to the general problem of implantment dynamics, it was felt to be inadequate for the purposes of this study. In view of this, the Battelle Memorial Institute (BMI) was contracted to analytically study the problem of dynamic loads during implantment of the SNAP-21 systems.

In performing this study, BMI utilized a two-dimensional ship motion computer program to determine the ship response to various sea states. This data was then used to analyze payload dynamic loads caused by resonance between the ship and the spring mass system consisting of the SNAP-21 system and its implantment cable. Also considered were shock loads due to water impact if the SNAP-21 were dropped into the water or if it were struck by a large wave while it is suspended over the ship's side.

The limited time available for this portion of the study (approximately two weeks) necessitated several approximations as to the type of ship that would be used and the effect that an implantment structure for the system would have on the system dynamics during implantment. In these cases, assumptions were made to produce a conservative estimate of the dynamic loads, i.e., more severe than would actually be required. For this reason, the predicted loads are to be considered preliminary in the sense that a detailed analysis of the implantment operation should be conducted once the specific vessel, mission, implantment site, etc., are known.

Results of the BMI study are presented in Appendix IV. A summary of the maximum loads determined in Appendix IV are shown in Table V. In addition to the loads shown in Table V, the problem of bottom impact must be considered.

Table V. Implantment Loads Summary

<u>Shock</u>	
Horizontal (normal to system axis)	141 g's due to wave "slap" (5 ft. wave) 20-Watt 101 g's due to wave "slap" (5 ft. wave) 10-Watt
Vertical (in-line with system axis)	123 g's due to free fall into water from 20 ft. height 73 g's for 10 foot drop into water
<u>Vibration</u>	
0.68 g's at 0.215 cps (occurs at resonance when cable length is 380 feet and wave height is 5 feet, Sea State of approximately 5).	

NOTE: Shock load calculations did not include the effect of an implantment structure.

From Appendix IV it is shown that underwater velocities due to implantment ship motion can be as high as 16 ft/sec. The total vertical velocity will then be the total of this dynamic component plus the constant lowering velocity. If it is assumed that the system impacts on a rigid bottom (rock) the impact loads experienced by the system will be a function of the in-water weight of the system, the impact velocity and the spring rate of the implantment structure. Figure 9 shows the relationship of these parameters and thus forms the design criteria for the implantment structure.

The implantment ship used as a model for this analysis was a barge with dimensions as shown below:

Length	100 ft.
Beam	30 ft.
Draft	5 ft.
Displacement	960,000 lbs.

Figure 9. Deflection vs Bottom Impact Velocity for Maximum De-acceleration of 6 g's

This configuration was selected because time did not permit considering a wide range of vessels, and, from previous work, BMI felt that it would provide conservative results.

The computer program used has been proven by BMI during similar analysis of underwater pipe laying operations. The program analytically determines the response of the barge to various Sea State conditions. For this analysis, a Sea State of 5 was the maximum condition considered because it was felt that implantment would not be considered in more severe seas.

As shown in Table V, the acceleration loads due to impact with the water appear quite high. These data were discussed with BMI and they expressed confidence in their analysis results, however, they did not have any actual test data to verify it. The results were based completely on analytical data obtained from an integration program that considered effects of payload shape, water compressibility, etc. Further discussion on this subject with personnel at David Taylor Model Basin tended to support the results since they had related data obtained during ship "slamming" experiments that was in reasonably close agreement.

The maximum vertical velocity and resultant g loads shown in Table V for a 20 ft. free fall to the water could be defined as an accident and therefore would not be considered normal. There are, however, a combination of vertical velocity components that will occur during normal operation and, if these velocities happen to occur in a cumulative manner, significant acceleration loads could occur. The velocity components that will contribute to this total are:

- a. The vertical component due to ship rolling
- b. The vertical motion of the waves, and
- c. The lowering speed of the cable.

In an attempt to determine what representative vertical velocities can be attained due to these conditions, an analysis was performed that considered various implantment ship sizes and ship response to sea conditions. This analysis is shown in Appendix V.

For the extremes of ship sizes and responses considered in this analysis, total vertical velocities considering all contributing factors could range from 9.3 to 26.9 ft/sec. This will result in vertical accelerations of 20 and 90 g's respectively, using BMI's water impact data. For these same extreme conditions, horizontal impact velocities with the ships' side would range from 6.75 to 10.8 ft/sec. Acceleration loads of two metal objects impacting at this velocity would be extremely high and thus the need for a shock attenuating structure is clearly indicated. It is interesting to note that from Appendix V certain combinations of ship size and boom rigging can be specified that will prevent any contact of the suspended system and the ships side. This would be a very desirable condition from a system protection standpoint and should be considered as a possibility for implantment.

5.1.4 RECOVERY LOADS

Actual data on dynamic loads encountered during recovery was also very sparse. Consequently, the primary source of information was personal communication with individuals involved in this work. In general, most persons contacted felt that recovery loads would be similar to implantment loads except for the loads that could be encountered during break-away from the bottom. The only actual data on this subject was obtained from personnel at NCEL who are currently conducting tests in San Francisco Bay to determine break-away coefficients for various shapes that are buried in the mud. They had not completed the tests at the time of our contact; however, as an example they reported a required break-away force of 5,000 to 7,000 lbs. for a six foot diameter sphere. In conducting these tests they pull very slowly to simulate plastic deformation of the bottom. Consequently, if this method were used, the dynamic loads experienced by the payload would be negligible.

The usual method of engaging an implanted object for recovery is to remotely release a submerged buoy that is attached to the payload recovery cable. These techniques are described in Ref. (10). It is usual to have redundant methods of engaging the payload recovery cable with the least preferred method being grappling.

To obtain an estimate of the dynamic loads that a payload would experience during grappling a simplified analysis was conducted. For this analysis it was assumed that the recovery ship would drag for the payload using a polypropylene/nylon cable at drag speeds from 1 to 7 knots. It was also assumed that a slip clutch on the recovery ship could be set so that the maximum load on the recovery cable could be limited to 2,000 lbs. above the tow drag force. It was then assumed that the payload would accelerate uniformly from rest to the recovery ship speed in the distance corresponding to the strain in the cable ($S = \frac{PL}{AE}$). A cable strength of 5,000 lbs. was used and both steel and plastic cables were considered.

A plot of payload acceleration vs. recovery ship speed for various cable lengths is shown in Figure 10. As shown in Figure 10, payload dynamic loads due to grappling can be kept below 5 g's so long as recovery ship speeds are kept below 5 knots and the cable length is greater than 500 ft.

5.1.5 DYNAMIC ENVIRONMENT SUMMARY

A summary of the maximum dynamic environment for each phase of the systems operational life is shown in Table VI. As shown, the source of maximum vibration loads is transportation, while handling and implantment impose the greatest shock loads. It should be noted that the loads shown in Table VI are loads applied to the complete system, including its shipping container or implantment structure. The magnitude of these loads clearly indicates the need for shock attenuating equipment for some operations.

5.2 DETERMINE THE DYNAMIC CAPABILITY OF THE EXISTING 10-WATT SYSTEM

The second step in the technical approach was to perform a dynamic analysis of the existing 10-watt system to determine its dynamic capability. Because only two of the major components, the thermoelectric generator and insulation system, are sensitive to dynamic loading, the analysis effort was concentrated on them. As mentioned previously in Section 4.0, Status of Technology, the segmented hold down ring design is also sensitive to dynamic loading; however, it was not considered in this study because the anticipated design changes required to increase its dynamic capability are straightforward.

Figure 10. Payload Acceleration Due to Recovery by Grappling

Table VI. Summary of Maximum Dynamic Environment

Operation	How Encountered	Maximum Shock	Maximum Vibration
Handling	Vertical 3 in. drop of system in steel container	100 g's vertical	----
	Ramming container with fork lift at 8 ft/sec. (unsupervised)	200 g's horizontal	----
Transportation	Ship transport during slam and emergency maneuvers (intermittent)	----	3 g's at 2.5 and 50 cps
	Helicopter (steady)	----	7 g's at 400 cps
	Truck (rough road)	----	2.0 g's at 10 cps
Implantment	Wave slap	141 g's (horiz.)	----
	Free fall to water (20 ft)	123 g's (vert.)	----
Recovery	Grappling	5 g's	----

Dynamic analysis of the thermoelectric generator is a complicated problem because some of the variables, such as friction and damping due to MIN-K insulation, cannot be predicted exactly and therefore, require judgment on the part of the analyst. Because of this, independent analyses were conducted by three individuals in order to verify their assumptions and judgment. Two of these analyses were conducted by consultants and one by the 3M Company. The three analyses are shown in Appendix VI, VII, and VIII. Although variations in approach and assumptions existed between the investigators, their conclusions were generally in accord. A summary of the generator dynamic capability as determined by these investigators is shown in Table VII. This table presents the most conservative of the results.

The remaining critical component, from a dynamic capability standpoint, is the insulation system. This component is being developed by the Linde Division of Union Carbide. They also did the Phase I development, consequently they were directed to analyze this component. This analysis is shown in Appendix II. The dynamic capability of the insulation system shown in Table VII is for the system as it was redesigned during Phase II. The data for the insulation system, as it was designed and fabricated during Phase I, is not presented since the dynamic capability of the Phase I configuration was found to be completely inadequate.

As shown in Table VII, the most critical component from a dynamic capability standpoint is the insulation system. Also, it is obvious that all components, except the insulation system, are capable of shock loads up to 8 g's with a safety factor of 1.5.

5.3 INVESTIGATE FEASIBILITY OF MODIFYING CURRENT 10-WATT SYSTEM TO INCREASE DYNAMIC CAPABILITY

A review of the dynamic analysis results shown in Table VII reveals that the component requiring dynamic capability improvement is the insulation system. Since the critical structural element of the insulation system is the shield support, several methods of improving the systems capability by strengthening and improving the supports were investigated.

Table VII. Dynamic Analysis Summary

Existing 10-Watt Design

Component	Design Requirement		Dynamic Capability (shock)	Factor of Safety (shock)	Natural Frequency (cps)	Remarks
	Shock	Vibration				
<u>Thermoelectric Generator</u>						Thermopile capability based on maintaining critical interfaces. Capability of the thermoelectric legs to withstand internal stress based on empirical data.
Outer Case	3 g's 6 ms	4.6 g's 50 cps	80 g's vert. 120 g's later.	26 40	3100 axial 1650 lateral	
Thermopile	3 g's 6 ms	4.6 g's 50 cps	12 g's vert. 80 g's lat.	4 26	70	
<u>Insulation System</u>						
Neck Tube	3 g's 6 ms	4.6 g's 50 cps	4.6 g's	1.5	167	

NOTE: Since the natural frequencies of the components are higher than expected environmental frequencies, resonance will not occur. Consequently if the system is designed to withstand the shock environment, vibration is not considered to be a mode of failure.

As a part of their effort, Linde conducted a design study to analytically evaluate various methods of shield support and how these methods affect system performance. The methods of support considered by Linde were: a tapered cantilever neck tube, neck tube plus three tension tie rod supports, and ceramic compression rods. Details of this effort are presented in Appendix II.

In addition to the new support methods considered by Linde, the possibility of using a rigid web-and-foil insulation material developed by BMI was considered. This rigid insulation block, as it would be used in the SNAP-21 insulation system, would serve as a supplementary shield support in conjunction with the Linde multi-foil super-insulation system. A schematic of the three BMI configurations considered is shown in Figure 11.

The effect of the various support methods on insulation system performance is shown in Figure 12. Since increasing the dynamic capability of the insulation system causes a decrease in its thermal efficiency, the efficiency of the overall system will also be affected. The effect of increased dynamic capability on system efficiency and weight is shown in Figures 13 and 14.

A review of the above data reveals that the insulation system can be modified by the addition of three tension tie rods to be capable of 6 g's and still maintain the same system efficiency as the current design will provide at the original dynamic requirement of 3 g's. This method of modifying the support structure is considered to be the most feasible from the standpoint of development risk. In addition, improving the system capability to this level will not require major modification of any other components.

In view of these data, and the magnitude of dynamic loads estimated for handling and implantment, it appears that increasing the system's dynamic capability to 6 g's is feasible and will meet the dynamic load requirements. This load level was used to determine design criteria for the shipping container and implantment structure in the following paragraph.

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Figure 11. Schematic of Shield Support Configurations Considered Utilizing BMI Web and Foil Insulation

Figure 12. Effect of g-Loading on Total Insulation System Heat Leak for Different Support Systems (Includes 1.5 Safety Factor)

Figure 13. SNAP-21 10-Watt System Efficiency vs Dynamic Capability for Various Insulation System Shield Supports

Figure 14. SNAP-21 10-Watt System Weight Increase vs System Dynamic Capability for Various Insulation System Shield Supports (Safety Factor of 1.5 Included)

5.4 DESIGN CRITERIA FOR SHIPPING AND IMPLANTMENT STRUCTURES

Based on the anticipated dynamic environment and a system capability of 6 g's, the design criteria for shipping and implantment containers or structures that will be required to attenuate the anticipated dynamic loads to levels that the system can withstand were established. These criteria are listed below.

5.4.1 SHIPPING CONTAINER CRITERIA

The shipping container requirements are as follows. The shipping container must:

- Attenuate shock due to 3 inch vertical drop and 6 inch corner drop to 6 g's on suspended system.
- Have a natural frequency less than 5 cps.
- Incorporate the following features in addition to normal AEC and ICC shipping container requirements:
 - a. Minimum size
 - b. Retractable pneumatic wheels to provide mobility in terminals, cargo holds, etc.
 - c. Built-in fixtures for fork-lift and winch handling to insure proper attitude during handling.
 - d. Be of metallic construction (reusable).
 - e. Have built-in attachment points for hold down cables, etc.

A preliminary design analysis of the suspension system required to provide this shock attenuation is shown in Appendix IX. This analysis shows that a combination of four rubber shear mounts with spring rates of 375 lbs/in. each will provide the necessary attenuation. The natural frequency of this system will be below 6 cps. These mounts are commercially available and in wide use.

Some auxiliary damping in addition to the inherent damping of the rubber mounts may be required to prevent any possibility of resonance.

5.4.2 IMPLANTMENT STRUCTURE CRITERIA

An implantment structure will be required to provide shock attenuation in two different situations. One is the "above surface" situation where the "wave slap" loads of up to 143 g's horizontal and the vertical acceleration loads of up to 90 g's must be attenuated to 6 g's. In addition, the possibility of impact with the ship at velocities up to 10 ft/sec must be attenuated to 6 g's. In the "below surface" situation the structure must attenuate bottom impact loads due to impact velocities of up to 20 ft/sec.

The water impact loads can best be attenuated by changing the effective shape of the device by artificially increasing its length, decreasing its water entry angle, or incorporating baffle or swash plates. For the ship impact problem, a spring device or shock absorbing cushion, such as a pneumatic or plastic foam device, can be used to attenuate the 10 ft/sec impact to 6 g's. A spring rate of approximately 300 lbs/in. would provide the required protection.

The second situation that must be considered is impact with a hard bottom. Bottom impact will occur and, even if lowering speeds were reduced to 1 ft/sec or less, some cushioning will be required. This could be achieved with a spring device or plastic deformation of a crushable energy absorbing substance. Other methods that should be considered are hydraulic cushioning or snubbing devices that would force trapped sea water out of an orifice at a controlled rate. Exact design requirements for this condition can be defined once the maximum lowering velocities are established. A curve of the spring rate and deflection required has already been shown in Figure 9.

The implantment structure design should be such that it does not promote marine growth or fouling that would impede heat transfer.

6.0 CONCLUSIONS

The conclusions that can be drawn about dynamic loading of the SNAP-21 system are presented below:

- The system requires external protection during shipment and implantment. This requirement is independent of dynamic load considerations because a highly stressed member (pressure vessel) is the outer-most component and an accidental dent, nick, scratch, etc., could cause a serious stress concentration point and subsequent failure.
- Dynamic loads imposed on the system during handling and implantment can be as high as 200 g's.
- It is not feasible to redesign the system to be capable of withstanding all potential dynamic loads, therefore, a means of shock attenuation must be provided for terminal handling, shipping, and surface implantment.
- The maximum dynamic loads that the system will be exposed to in situations where shock attenuation is not feasible are as follows:
 - a. Removing system from assembly jig and installing it in an implantment structure aboard ship (assumed to be identical with an equivalent operation ashore from dynamic load standpoint) 6-1/2 g's
 - b. Installing system in shipping container and removing it aboard ship. 4 g's

- c. Securing the system to the stern of a ship and then
 executing emergency maneuvers. (This could occur
 during an implantment attempt that was aborted due
 to weather, etc.) 3 g's

- The only component requiring major modification to upgrade the
 system dynamic capability to 6 g's is the insulation system. The
 insulation system dynamic capability can be increased to 6 g's
 by the addition of three tension rods to the shield support system.
 This increased capability can be obtained at the expense of an
 additional heat leak of approximately 17 watts over the original
 design goal.

- It is technically feasible to increase the system dynamic
 capability to 6 g's and still maintain the system efficiency originally
 specified for 3 g's by the use of ceramic compression rods
 in the insulation system to support the shield, however, it
 would increase the insulation system complexity.

- By incorporating system design modifications to increase
 the system dynamic capability to 6 g's, and by the use of
 shipping and implantment structures, the SNAP-21 system
 can withstand its anticipated dynamic environment.

- Adequate instruction, training and procedures must be pro-
 vided for personnel handling the system to minimize the
 possibility of accidental damage.

7.0 RECOMMENDATIONS

Recommendations that can be made as a result of this study are as follows:

7.1 HANDLING AND ASSEMBLY

- Handling of the system, packaged or unpackaged, must be supervised at all times.
- Steel rimmed or solid wheel transportation vehicles should not be used.
- A specific study of the problem of handling aboard ship should be conducted once the specific type of ship and available handling gear is defined.

7.2 TRANSPORTATION

It is recommended that the system not be shipped by rail. This recommendation is made even though the modified system, in its shipping container, could theoretically withstand the maximum dynamic loads of 8 g's during humping or coupling (design requirement of 6 g x 1.5 safety factor = 9 g). The reason for this recommendation is the difficulty of providing adequate supervision during shipment. This is especially important from a safety hazard consideration, since the system could spend long periods of time in switching yards, etc., and could be in close proximity to hazardous cargoes such as gasoline, explosives, etc., which in the event of an accident could create a nuclear safety hazard.

7.3 IMPLANTMENT

- Braided, non-rotating polypropylene cable should be used as the implantment cable.
- The cable must be "sized" for the specific payload weight so that the effective stiffness can be kept low.
- If possible, implantment gear should be designed to preclude the possibility of banging against the side of the ship.
- Implantment should not be attempted in sea states above 5.

7.4 SYSTEM DESIGN

- The system specification should be changed with respect to dynamic environment to a shock requirement of 6 g's in 6 ms and a vibration environment in accordance with MIL-STD-810A, except that maximum acceleration shall be limited to 3 g's.
- The insulation system should be modified by incorporating three tension tie rods as a supplement to the neck tube for structural support of the biological shield.

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APPENDIX I

ANALYSIS OF DYNAMIC LOADS ON SNAP-21 COMPONENTS AND SYSTEM DURING HANDLING AND ASSEMBLY

Table I-1. Assembly of Insulation System, Generator Mounting Plate and Centering Ring Sub-Assembly into Body of Pressure Vessel and Packaging Assembly. (Metal Container Assumed)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Load
10	Obtain Pressure Vessel Body from stock. It will be in wood shipping crate.	Fork lift truck	Insignificant
20	Remove cover and sides of crate and padding as needed to allow vertical lift from crate.	Crowbar and hammer	Insignificant
30	Attach Lifting Bar to Pressure Vessel Body.	Lifting bar and screws - small hand wrenches	Insignificant
40	Lift Pressure Vessel Body from container and remove container. Raise high enough to accept Assembly fixture OPR. 50.	Hydraulic hoist with cable. Portable	Insignificant
50	Place Assembly Fixture under Pressure Vessel Body and slowly lower Pressure Vessel Body into place.	Assembly Fixture - Holds Pressure Vessel Body in vertical position and supports it with springs or cushioning to be developed later but commensurate with dynamic environment herein described. Fixture is portable on 4 wheels.	Insignificant
60	Clamp or bolt Pressure Vessel Body into place as indicated by design to be developed later.	Small hand tools	Insignificant
70	Remove Cable, hoist and lifting bar.	Small hand tools	Insignificant

Table I-1. Assembly of Insulation System, Generator Mounting Plate and Centering Ring Sub-Assembly into Body of Pressure Vessel and Packaging Assembly. (Metal Container Assumed)
(Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Load
80	Obtain packaged Insulation System from stock.	Fork lift truck	3 g's - (same values as terminal handling, supervised)
90	Remove cover from container. Remove container insulating padding or springs as needed.	Small hand wrenches	Insignificant
110	Attach hydraulic hoist cable to shipping plate and put tension in cable until weight of sub-assembly is supported by cable. Lock hoist in this position.	Hydraulic hoist with cable. Portable.	Moving hoist into container at speed 1 f. p. s. 800 lb. hoist - 400 lb. container 0.020 second horizontal time-rise. 1-1/2 g's.
120	Remove container insulation pad or springs as required for lifting out sub-assembly.	Small hand wrenches	Insignificant
130	Raise sub-assembly with hydraulic hoist to clear container. Manipulate by hand. Pull container out from under sub-assembly.	Sub-assembly supported by mounting plate which is bolted to the radiation plug mounting holes.	Sub-assembly impacting against container - Velocity 1 f. p. s. 100 lb. container - 300 lb. System 0.020 second horizontal time-rise. 1/2 g.
140	Place Pressure Vessel Body and its assembly fixture (OPR. 80) under the Insulation Envelope Configuration.	--	Insignificant

Table I-1. Assembly of Insulation System, Generator Mounting Plate and Centering Ring Sub-Assembly into Body of Pressure Vessel and Packaging Assembly. (Metal Container Assumed)
(Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Load
150	Slowly lower Insulation Envelope Configuration into Pressure Vessel Body. Manipulate by hand.	Small hand tools	Insulation Envelope bumping Pressure Vessel on entry and on seating on bottom 0.5 f.p.s. maximum velocity. 300 lbs. system - 200 lbs. Body 0.002 rise-time. 7-1/2 g's.
160	Remove Insulation System shipping plate.	Small hand tools	Insignificant
170	Place necktube guard into necktube.	Necktube guard - to be developed.	Insignificant
180	Obtain from stock Micro Quartz insulating pad.	--	--
190	Place Micro Quartz insulating pad in place on Insulation System.	None	Insignificant
200	Obtain generator Mounting Plate and Centering Ring sub-assembly from stock.	None	None
210	Attach lifting bar to Generator Mounting Plate and Centering Ring sub-assembly.	Lifting bar for Generator Mounting Plate and Centering Ring.	Insignificant
220	Attach hydraulic hoist cable to lifting bar and raise.	Hydraulic hoist with cable-portable.	Insignificant

Table I-1. Assembly of Insulation System, Generator Mounting Plate and Centering Ring Sub-Assembly into Body of Pressure Vessel and Packaging Assembly. (Metal Container Assumed)
(Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Load
230	Center Generator Mounting Plate over assembly of Insulation System and Pressure Vessel Body. Slowly lower into body until ring contacts Insulate envelope. Manipulate by hand to assure smooth even entry.	None	Generator Mounting Plate Assembly impacting Pressure Vessel on entry, seating on insulations system 1/2 f.p.s. 50 lbs. - generator mounting plate sub-assembly 500 lbs. Pressure Vessel assembly. Rise-time 0.002 seconds. 1-3/4 g's.
240	Remove cable, hydraulic hoist Generator Mounting Plate lifting bar.	Small hand tools.	Insignificant
250	Obtain Shipping Plate Assembly from stock.	Shipping plate assembly to be a part of the shipping container. It will bolt to the outer ring of holes of the Pressure Vessel Body and form a hold down for the Generator Mounting Ring and Neck Tube Guard. It will have a removable inner plate for fueling the system and will form a dust and dirt barrier. It will have hoisting provisions.	None
260	Place two Pilot Screws into flange of Pressure Vessel Body.	Pilot Screws to be developed.	Insignificant

Table I-1. Assembly of Insulation System, Generator Mounting Plate and Centering Ring Sub-Assembly into Body of Pressure Vessel and Packaging Assembly. (Metal Container Assumed)
(Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
270	Raise Shipping Plate by hoist. Place in position over the pilot screws onto the Generator Mounting Plate.	Hydraulic hoist with cable.	Shipping plate striking screws and setting down impact on generator mounting plate. Rise-time 0.002 second. 1-3/4 g's. Moving hoist into assembly fixture 1 f.p.s. velocity, 800 lbs. each, 0.002 second rise-time. 12 g's.
290	Fasten shipping plate with flange mounting screws. Torque to be specified when design finalized.	Shipping screws	Insignificant
310	Put tension into cable until weight of assembly is supported by the cable.	None	Insignificant
320	Remove springs or cushioning attaching to Pressure Vessel Body.	Small hand tools	Insignificant
330	Raise assembly until it is above the assembly jig. Remove assembly jig.	None	Assembly impacting assembly jig 1 f.p.s. velocity, 500 lbs. to 200 lbs. 0.002 second rise-time. 8-1/2 g's.

Table I-1. Assembly of Insulation System, Generator Mounting Plate and Centering Ring Sub-Assembly into Body of Pressure Vessel and Packaging Assembly. (Metal Container Assumed)
(Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
340	Place shipping container beneath assembly. If more vertical clearance is required, raise assembly accordingly.	Shipping Container - to be designed to be commensurate with the dynamic loads. Actual putting into container will be outlined in detail after container design is completed. Container to incorporate heat dissipation means.	Insignificant
350	Lower assembly into position.	None	Assembly striking sides of shipping container - maximum velocity 1/2 f.p.s. 600 lbs/400 lbs-10-watt - 8 g's. 800 lbs/400 lbs-12-watt - 6-1/2 g's. 0.002 second rise-time.
360	Assemble supporting members or cushions as required by design of container. Complete containment as required.	Small hand tool	Insignificant
370	Remove cable and hoist.	None	Hoist striking shipping container 1 f.p.s. velocity 0.002 rise-time (Steel container) 10-watt system - 9 g's.
380	Place in storage	Fork lift truck	Same values as supervised shipping. For metal container: 29 g's.

Table I-2. Fueling of Insulation System by Oak Ridge National Laboratory (ORNL)

Note 1. It is assumed that handling by ORNL of the container with the Pressure Vessel Body, Insulation System, Generator Mounting Plate and Centering Ring Assembly will impose no worse dynamic load on the packaged system than those imposed through shipping under supervised conditions.

In addition to the above described assembly, the following components will be furnished to ORNL for the fueling operation:

Fuel Capsule
Fuel Capsule Cap

Radiation Shield Plug with handling bar
Screws, Radiation Shield Plug

Barrier Assembly
Heat Sink (to be designed)
Screws, Heat Sink

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
10	Place container in fueling position. Brace as required.	Materials handling equipment.	See note above.
20	Open container top to expose inner shipping plate. Remove parts of container required for fueling.	Hand tools	Insignificant
30	Remove inner shipping plate.	Hand tools	Insignificant
40	Layout parts for remote handling.	Special holding jigs, fixtures, table, etc.	Insignificant

Table I-2. Fueling of Insulation System by Oak Ridge National Laboratory (ORNL) (Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
50	Remove Neck Tube Guard.	Materials handling equipment	Insignificant
60	Place shielded and chilled fuel capsule in position for remote removal of capsule.	Materials handling equipment	None
(The following operations will be performed by remote control.)			
70	Remove cover of fuel capsule shield.	None	None
80	Change manipulator tool to fuel capsule tool.	Fuel capsule tool designed to hold fuel capsule with a claw type grip; have a sliding guide to pilot into top of neck tube and control sideward movement of fuel capsule within the confines of the tube without impacting sides of tube; and incorporate a bottoming spring loaded compliance.	None
90	Pick up fuel capsule.	Change design of Fuel Capsule Cap to provide circular recess with back out for positive gripping and tapered lead in edge.	Insignificant
100	Place fuel capsule into insulation system 2 fpm max. velocity.	None	Insignificant
110	Release fuel capsule and retract manipulator arm.	None	Insignificant

Table I-2. Fueling of Insulation System by Oak Ridge National Laboratory (ORNL) (Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
120	Change manipulator tool to Radiation Plug tool.	Radiation Plug tools have basic features described for fuel capsule tool Opr. No. 80 plus a rotational capability to line up screw holes with tapped holes.	None
130	Pick up Radiation Plug with Screws.	Radiation Plug to have 6 Mounting Screws set in place in the holes. Slight design change required for external pickup.	Insignificant
140	Place Radiation Plug into Neck Tube. Max. velocity 2 fpm.	None	Insignificant
150	Release Radiation Plug and retract manipulator arm.	None	Insignificant
160	Change manipulator arm tool to standard parallel jaw hands.	None	None
170	Pick up Radiation Plug screw wrench.	Wrench to be developed.	None
180	Tighten screws for Radiation Plug.	Wrist rotation torque previously set.	Insignificant
190	Place Heat Sink into Position.	Heat Sink device - to be developed later.	Insignificant - Assuming that intimate contact with radiation plug is not required.
200	Decontaminate system.	None	Insignificant

Table I-2. Fueling of Insulation System by Oak Ridge National Laboratory (ORNL) (Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
210	Install Heat Sink Screws tightened to specific torque.	Hand Tools	Insignificant
220	Install inner shipping container plate.	Hand tools	Insignificant
230	Replace parts removed under Opr. No. 20.	Hand tools	Insignificant
240	Transport container to shipping dock.	Materials handling equipment	See Note 1

Table I-3. Final Assembly and Packaging of System

Note 1. It is assumed that the in-plant handling of the packaged system will be of the same level as terminal 'supervised' operations. It is also assumed that metal shipping containers are used.

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
10	Obtain Pressure Vessel Body fueled assembly from stock.	Fork Lift Truck	See Note 1 (20 g's)
	Obtain Pressure Vessel Cover assembly, including converter receptacle, from stock.	None	Insignificant
20	Obtain the following from stock as required: Generator Generator Mounting Screws Generator Mounting Plate and Centering Ring Screws MIN-K Insulation Pressure Vessel O-Rings (2) Pressure Vessel Cover Screws System Shipping Container Radiation Disc Radiation Disc Screws	Hand Carry	Insignificant
30	Pressure Vessel Body - Open container. Attach hydraulic hoist cable to shipping plate and put tension on cable until weight of assembly is supported by cable.	Hydraulic Hoist - Portable Small hand tools	Running hoist into container -- 1 f. p. s. velocity. 800 lb. hoist, 800 lb. container, 0.002 second rise-time. 18 g's.

Table I-3. Final Assembly and Packaging of System (Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
40	Disassemble supporting members or cushions as required by design of container.	Small hand tools.	Insignificant
50	Lift Pressure Vessel Body clear of container and remove container. Raise body high enough to accept assembly jig.	None	Assembly striking sides of shipping container -- maximum velocity 1/2 f.p.s. 600 lb. system/200 lb. container, 0.002 second rise-time. 3 g's.
60	Place Assembly Jig into position to accept Pressure Vessel Body assembly.	Assembly Jig same as used for assembly of insulation system and hold down ring into Pressure Vessel Body.	Insignificant
70	Lower Pressure Vessel Body assembly into position for securing to Assembly Jig and secure it to jig.	Small hand tools	Insignificant -- assuming some cushioning built into jig.
80	Release cable from shipping plate and remove hoist.	None	Pushing hoist into Assembly Jig -- 1 f.p.s. velocity. 600 lb. Pressure Vessel, 800 lb. hoist, 0.002 second rise-time, 10 g's.
90	Remove inner shipping plate from top of Pressure Vessel Body assembly.	Small hand tools	Insignificant

Table I-3. Final Assembly and Packaging of System (Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
100	Remove Heat Sink from system.	Small hand tools	Insignificant
110	Place Radiation Disc in position. 1 f. p. s. velocity	Holder for outer edge -- spring loaded.	Insignificant
120	Insert Radiation Disc Screws and tighten to specified torque.	Torque Wrench	Insignificant
140	Place Generator in place by hand. 1/4 f. p. s. maximum velocity.	None	Insignificant
150	Bolt Generator in place. Torque to be specified later.	Small hand tools	Insignificant
160	Connect Generator output terminals to load if later developments indicate need.	Electrical load device if required.	Insignificant
170	Pour MIN-K insulation into cavity surrounding Generator. Fill cavity.	Funnel	Insignificant
180	Remove outer shipping plate from top of Pressure Vessel Body assembly.	Small hand tools	Insignificant
190	Remove Pressure Vessel Cover assembly from container.	Hand tools	Insignificant
200	Remove necessary screws from Receptacle Flange and attach lifting bar.	Lifting Bar to be designed.	Insignificant

Table I-3. Final Assembly and Packaging of System (Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
210	Raise Pressure Vessel Cover assembly and position over Assembly Jig and Pressure Vessel Body so that wires from converter can be soldered to Generator.	Hydraulic Hoist -- portable.	Same as Opr. No. 80
220	Make solder connection of wires to Generator. Disconnect load first if added in Opr. No. 120.	Soldering gun and soft solder.	Insignificant
230	Place O-Rings (2) into O-Ring grooves.	None	Insignificant
240	Lower Pressure Vessel Cover onto Pressure Vessel Body.	None	Pressure Vessel Cover striking Pressure Vessel Body or Generator Mounting Flange at velocity 2 f.p.s. 50 lbs. - 600 lbs. 0.002 second rise-time. 3-1/2 g's.
250	Hand tighten all flange mounting screws - Pressure Cover.	None	Insignificant
260	Tighten all screws in Pressure Vessel flange in pattern, sequence and manner to be prescribed later.	Socket wrench	Insignificant
270	Put tension in hoist cable to support weight of system.	None	Insignificant
280	Disassemble assembly jigs, supporting members or cushions as needed to lift clear.	Small hand tools	Insignificant

Table I-3. Final Assembly and Packaging of System (Continued)

Opr. No.	Description	Tooling Required	Probable Cause and Estimate of Dynamic Loads
290	Raise final assembly clear of assembly jig and remove jig.	None	Assembly impacting against assembly jig on being raised - 1 f.p.s. velocity. 800 lbs. to 200 lbs. 0.002 second rise-time. 6-1/2 g's.
300	Raise assembly high enough to clear shipping container. Place shipping container under assembly.	Fork lift truck to move shipping container.	Insignificant
310	Lower assembly into position in shipping container.	None	Bump side of Pressure Vessel against container -- velocity 1 f.p.s. 800 lbs. to 200 lbs. 0.002 second rise-time. 4 g's.
320	Assemble supporting members to cushions as required by design of container.	Small hand tools	Insignificant
330	Remove cable and hoist.	None	Hoist striking shipping container - velocity 1 f.p.s. 800 lbs. - 100 lbs; 0.002 second rise-time. 11 g's.
340	Remove lifting bar and replace Flange Receptacle Cover Screws.	Small hand tools	Insignificant
350	Plug in electrical load if required.	Load to be determined.	Insignificant
360	Complete containment.	Small hand tools	Insignificant
370	Place in storage.	Fork lift truck	Same as Oppr. No. 10

APPENDIX II
HIGH TEMPERATURE VACUUM INSULATION SYSTEM
DYNAMIC CRITERIA STUDY

UNION CARBIDE CORPORATION

Linde Division

Engineering Department

Tonawanda, New York

HIGH TEMPERATURE

VACUUM INSULATION SYSTEM

Letter Subcontract Number

AT(30-1)-3691-5511

TOPICAL REPORT

DYNAMIC CRITERIA STUDY

TASK IA

for

3M Company

Electrical Products Division

St. Paul, Minnesota

January 31, 1967

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1.0 INTRODUCTION

At the request of 3M Company a study was undertaken by the Trade Division of Union Carbide Corporation to determine: 1) the system heat loss resulting from increased dynamic loading on the design of the HEVJS unit considering various support concepts, and 2) the magnitude of the g-loading to which the unit will be subjected during shipping and handling. A parametric study was conducted to determine how the increased g-loading will affect the system heat leak for a number of proposed support concepts to include neck tube support, neck tube with three tension rod supports, neck tube with three compression rod supports, bellows with six compression rod supports, bellows with six tension rod supports, bellows with stacked disc compression supports, and bellows with compressed multi-layer insulation supports.

An additional study was performed to determine the allowable input g-loading to the unit during shipping for the various support concepts for the specific case where the unit is directly supported by rigid attachment to the nuclear shield.

An investigation of the g-loading to which the unit will be subjected during shipping and handling was conducted to include 1) a literature survey on the dynamic loads normally encountered by similar items during shipping and handling, 2) preparation of a flow plan for fabrication and testing of the unit which indicated estimated g-loadings encountered during each operation, and 3) experimental testing conducted with fork-lift trucks to determine dynamic loading encountered during fork-lift truck operations since a large portion of the unit handling is expected to be accomplished by fork-lift trucks.

2.0 PARAMETRIC STUDY

2.1 Heat Loss Determination for Proposed Support Systems

A parametric study was conducted to determine how increased unit g-loading will affect the total system heat leak for a number of proposed support systems to include: neck tube support, neck tube with three tension rod supports, neck tube with three compression rod supports, bellows with six compression rod supports, bellows with six tension rod supports, bellows with stacked disc compression rod supports, and bellows with compressed multi-layer insulation supports.

The support systems were analyzed in the range of 3g to 20g and/or a maximum of 70 watts total system heat leak. Figure 1 is a graphical presentation showing the effect of g-loading on total system heat leak for the different support systems. A schematic diagram of each of the support systems is shown on Figure 2. The reason only the first two support systems are shown on Figure 1 is that the other three systems were found not suitable for this application because of geometry considerations or insufficient property data to permit an evaluation. The support systems were designed for both shock and vibration loadings at 50 cps. The effect of increased fuel loading on shield weight for

FIGURE 1

FIGURE 2

SCHEMATIC DIAGRAM OF DIFFERENT SUPPORT SYSTEMS

Neck Tube

Neck Tube Plus
Three Tension Rods

Neck Tube Plus
Three Compression Rods

Bellows With Six
Tension or Compression Rods

Bellows With Six
Stacked Disc Compression
Supports

Bellows With Compressed
Multi-Layer Insulation

Note: Tension and Compression Rods are located 120° apart.

systems with a heat loss greater than 30 watts has been factored into the structural strength of the different support systems as per 3M Company letter of November 22, 1966. For this reason the total system heat leak for the neck tube and the neck tube with three tension rod support systems at 3g dynamic loading conditions is slightly higher than reported previously in Report No. 3.

The following simplifying assumptions were made for the parametric study.

1. The amplification factor due to shock was assumed to be between 1.2 and 1.3 for the support systems investigated. The amplification factor due to shock for a cantilever beam with a weight on its end ranges from 0.85 to 1.3 as presented in Appendix B, Reference 5, Figure 5.9, page 117. By using the 1.2 or 1.3 amplification factor for this parametric study, the calculations are on the conservative side.

2. The end temperatures of the tension and compression rods were assumed to be the same as the end temperatures of the neck tube. By making this assumption, it was not necessary to solve for the temperature distribution in the neck tube and in the tension or compression rods separately. The end temperatures of the neck tube and support rods were 1325°F and 150°F.

2.1.1 Bellows with Six Compression Rods Support System

The support system utilizing the bellows (Hastelloy-X) with six compression rods (zirconium dioxide) offers the best thermal performance among the support systems investigated. The compression rods were designed for 1.5 safety factor based on the compressive yield strength and were checked for buckling in compression. The compression rods will be installed approximately 45 degrees to the axial centerline to support the shield both in the hot and cold conditions. It was found that the highest stress in the compression rods occurred when the unit was in the horizontal position. The amplification factors due to vibration at 50 cps for different g-loadings were calculated for the unit being in the horizontal position. The amplification factor due to vibration at 50 cps ranged from 1.0 to 1.2 depending on the g-loading. The amplification factor due to shock loading was assumed to be 1.2 for all input g-loadings. The 0.008 inch wall thickness of the bellows was determined based on minimum manufacturing capabilities. The stresses in the bellows due to the 2 atm. outside pressure were calculated and were found safe; calculations for 10g dynamic loading are presented in Part IV of Appendix A. Results of the calculations for 3g, 10g and 20g dynamic loading are summarized in Table I.

2.1.2 Neck Tube with Three Compression Rods Support System

The support system utilizing a straight or tapered neck tube (Hastelloy-X) and three compression rods (zirconium dioxide) offers the second best thermal performance among the systems investigated. In this support system the neck tube carries about 40 per cent of the total load. The balance of the load is carried by the three compression rods. The neck tube wall thickness of 0.0096 inch at 3g dynamic loading was selected

TABLE I

RESULTS OF PARAMETRIC STUDY FOR BELLOWS WITH
SIX COMPRESSION ROD SUPPORT SYSTEM

G-Loading	<u>3</u>	<u>10</u>	<u>20</u>
Natural Frequency rad/sec	755	1405	1940
Amplification Factor Due To Vibration At 50 cps	1.2	1.03	1.0
Amplification Factor Due To Shock Loading (.85 - 1.3)	1.2	1.2	1.2
Angle Between Compression Rods And Axial Center Line (Degrees)	45	45	45
Allowable Stress In Compression Rods For 1.5 Safety Factor (psi)	16,666	16,666	16,666
Maximum Applied Stress In Compression Rods (psi)	14,910	14,415	15,032
Maximum Load Carried By Compression Rod (Lbs)	560	1866	3733
Bellows ¹ Wall Thickness (Inches)	.008	.008	.008
Compression Rod Diameter ² Required For 1.5 Safety Factor (Inches)	7/32	13/32	9/16
Heat Leak Through Bellows (Conduction Plus Radiation) (Watts)	6.9	6.9	6.9
Heat Leak Through Six Compression Rods (Watts)	2.2	7.4	14.3
Heat Leak Through Bellows Plus Six Compression Rods (Watts)	9.1	14.4	21.2
Heat Leak Through Insulation (Watts)	9	9	9
Total System Heat Leak (Watts)	18.1	23.4	30.2

1) Material - Hastelloy-X

2) Material - Zirconium Dioxide

based on minimum manufacturing capabilities. The neck tube at 3g dynamic loading had a safety factor higher than 1.5 in shear buckling. Since the compression rods were designed for a 1.5 safety factor in compression, the mode of failure shifted from the neck tube in shear buckling to the compression rods in compression. This is an advantage of this system at 3g dynamic loading since the strength of the compression rods in compression can be determined with a greater reliability than the strength of the neck tube in shear buckling. At 10g dynamic loading the straight neck tube with 0.0095 inch wall thickness determined for the 3g loading had to be changed to a tapered neck tube with 0.013 inch and 0.017 inch wall thicknesses at top and bottom, respectively, in order to design the neck tube for a 1.5 safety factor in shear buckling. Based on similar calculations at 5g dynamic loading for the neck tube and three tension rod support system, over 5g dynamic loading the neck tube is designed in shear buckling. In this support system the compression rods will be installed about 73 degrees to the axial centerline to support the shield both in the hot and cold conditions. The amplification factor due to vibration at 50 cps was not calculated for this system because similar calculations for the neck tube with three tension rod supports indicated that the amplification factor will be near unity. The amplification factor due to shock loading was assumed to be 1.2. Calculations for 10g dynamic loading are presented in Part III of Appendix A. Results of the calculations for 3g, 10g, 15g and 20g dynamic loading are summarized in Table II.

2.1.3 Neck Tube with Three Tension Rods Support System

The support system incorporating a straight or tapered neck tube (Hastelloy-X) and three tension rods (Hastelloy-X) was found thermally less efficient than the support system incorporating the neck tube and three compression rods. The thermal conductivity to yield strength ratio is about six times larger for Hastelloy-X than for zirconium dioxide. The straight neck tube with 0.0095-inch wall thickness was used for the 3g and 5g design cases. The 0.0095-inch wall thickness was selected based on minimum manufacturing capabilities. With dynamic loading over 5g the neck tube is designed for a 1.5 safety factor in shear buckling. The tension rods in this support system will be installed at 60 degrees to the axial centerline to support the load in both the cold and hot conditions. In this support system the neck tube carries about 40 per cent of the total load, and the balance of the load is carried by the tension rods. The amplification factor due to vibration at 50 cps for the different g-loading ranged from 1.06 to 1.10. The amplification factor due to shock was assumed the maximum 1.3 for all g-loadings. Calculations for 5g dynamic loading are presented in Part II of the Appendix A. Results of the calculations at 3g, 5g and 10g dynamic loading are presented in Table III.

2.1.4 Neck Tube Support System

The support system using only a tapered neck tube (Hastelloy-X) has the highest heat leak among the systems investigated. The neck tube was designed for a 1.5 safety factor for buckling in combined bending and internal pressure at the cold end and combined transverse shear and internal pressure at the hot end. The amplification factor due to vibration at 50 cps for this support system ranged from 1.4 to 1.8, depending on

TABLE II

RESULTS OF PARAMETRIC STUDY FOR NECK TUBE AND THREE
COMPRESSION ROD SUPPORT SYSTEM

G-Loading	<u>3</u>	<u>10</u>	<u>15</u>	<u>20</u>
Fuel Load Increase (Watts)	0	12	28	45
Increased Shield Weight (Lbs)	264 (Not Increased)	268	274	279
Amplification Factor Due To Shock Loading (.85 - 1.3)	1.2	1.2	1.2	1.2
Angle Between Compression Rods And Axial Center Line (Degrees)	73.3	73.3	73.3	73.3
Top And Bottom Neck Tube Wall Thickness Required For 1.5 Safety Factor (Inches)	T .0095 B .0095	T .013 B .017	T .019 B .026	T .026 B .035
Allowable Stress In The Compression Rod For 1.5 Safety Factor (psi)	16,666	16,666	16,666	16,666
Maximum Applied Stress In The Compression Rod (psi)	12,180	12,295	13,365	13,435
Maximum Load Carried By The Compression Rod (Lbs)	457	1,592	2,460	3,337
Compression Rod Diameter Required For 1.5 Safety Factor (Inches)	7/32	13/32	31/64	9/16
Heat Leak Through Neck Tube ¹ (Watts)	17.0 - 18.3	26.9 - 28.3	40.5 - 41.8	55.0 - 56.3
Heat Leak Through Three Compression Rods ² (Watts)	1.4	4.9	6.9	9.3
Heat Leak Through Neck Tube Plus Three Compression Rods (Watts)	18.4 - 19.7	31.8 - 33.2	47.4 - 48.7	64.3 - 65.6
Heat Leak Through Insulation (Watts)	9	9	9	9
Total System Heat Leak (Watts)	27.4 - 28.7	40.8 - 42.2	56.4 - 57.7	73.3 - 74.6

1) 1.5 mil variation on outside diameter is allowed for manufacturing tolerances
(Material, Hastelloy-X)

2) Material, Zirconium Dioxide

TABLE III

RESULTS OF PARAMETRIC STUDY FOR NECK TUBE WITH
THREE TENSION ROD SUPPORT SYSTEM

G-Loading	<u>3</u>	<u>5</u>	<u>10</u>
Fuel Load Increase (Watts)	6	13	40
Weight of Shield Plus Fuel Capsule (Lbs)	266	269	278
The Angle Between The Tension Rods and The Axial Center Line (Degrees)	60	60	60
Top and Bottom Neck Tube Wall Thickness Required For 1.5 Safety Factor (Inches)	T .0095 B .0095	T .0095 B .0095	T .013 B .017
Natural Frequency rad/sec	1006	1130	1226
Deflection of The Shield At The Center of Gravity (Inches)	1.1×10^{-3}	1.52×10^{-3}	2.57×10^{-3}
Amplification Factor Due To Vibration At 50 cps	1.1	1.07	1.06
Amplification Factor Due To Shock	1.3	1.3	1.3
Tension Rod Diameter Required For 1.5 Safety Factor (Inches)	11/64	15/64	11/32
Maximum Load Carried By The Tension Rod (Lbs)	453	809	1704
Allowable Stress In The Tension Rod (psi)	19,958	19,958	19,958
Maximum Applied Stress In The Tension Rod (psi)	19,545	18,770	18,371
Heat Leak Through Neck Tube ¹ (Watts)	17.0 - 18.4	17.0 - 18.4	26.9 - 28.3

TABLE III(Cont'd)

RESULTS OF PARAMETRIC STUDY FOR NECK TUBE WITH
THREE TENSION ROD SUPPORT SYSTEM

G-Loading	<u>3</u>	<u>5</u>	<u>10</u>
Heat Leak Through Three Tension Rods ² (Watts)	7.9	15.0	32.3
Heat Leak Through Neck Tube Plus Three Tension Rods (Watts)	24.9 - 26.3	32.0 - 33.4	59.2 - 60.6
Heat Leak Through Insulation (Watts)	9	9	9
Total System Heat Leak (Watts)	33.9 - 35.3	41.0 - 42.4	68.2 - 69.6

- 1) 1.5 mil variation on outside diameter is allowed for manufacturing tolerances
(Material - Hastelloy-X)
- 2) Material - Hastelloy-X

the g loading. Since the maximum amplification factor due to shock was 1.3, this system was designed for the higher amplification factors obtained due to vibration. Calculations for 5g dynamic loading conditions are presented in Part I of Appendix A. Results of the calculations at 3g, 5g and 8g are summarized in Table IV.

2.1.5 Bellows with Six Tension Rods Support System

The support system which uses bellows with six tension rods was found not capable of supporting the shield in both the hot and cold conditions due to the large thermal expansion of the Hastelloy-X tension rods compared to the thermal expansion of the shield. The thermal expansion coefficient of the tension rods (Hastelloy-X) is about twice that for the compression rods (zirconium dioxide). Calculations for this system are presented in Part V of Appendix A.

2.1.6 Bellows with Six Stacked Disk Compression Rods Support System

The support system which uses the bellows with six stacked disc compression supports is not able to support the shield in both the hot and cold conditions for the same reason as was explained for the bellows and six tension rod system. The thermal conductivity of stacked disc supports operating at high temperatures is not known. It is expected that at high operating temperatures under high compressive loads in vacuum the metal discs would partially or completely weld together, and therefore the thermal conductivity of the stacked disc support would approach the conductivity of the bulk material. Coating of the metal discs with a ceramic material may possibly avoid the potential welding problems.

2.1.7 Bellows with Compressed Multi-Layer Insulation Support System

The support system utilizing the bellows with compressed multi-layer insulation is not recommended for this application. Calculations based on the static load material properties show that this support system will withstand these static loads; however, experimental data are not available on the load-bearing capacity of multi-layer compressed insulation under dynamic loading conditions. Calculations for this support system are presented in Part VI of Appendix A.

2.2 Allowable System "g" Loading with a Directly Supported Nuclear Radiation Shield

Calculations were performed to determine the allowable "g" loading on the HTVIS system during handling or shipping when the nuclear radiation shield is rigidly supported and directly receives the dynamic loads. This study showed that, with direct support of the nuclear shield, dynamic loads significantly higher than the design loading (based on loading of the enclosure) can be safely applied to the unit. The greater loading capacity resulting from directly loading the nuclear shield derives from the fact that the unit support system must only support the relatively lightweight enclosure shell instead of the heavy nuclear radiation shield. Figure 3 is a graphical presentation of the results of this shield as applied to the four potential support concepts. Calculations are presented in Part VII of Appendix A.

TABLE IV

RESULTS OF PARAMETRIC STUDY FOR NECK TUBE

SUPPORT SYSTEM

G-Loading	<u>3</u>	<u>5</u>	<u>8</u>
Fuel Load Increase (Watts)	19	39	59
Weight Of Shield Plus Fuel Capsule (Lbs)	271	280	285
Top and Bottom Neck Tube ¹ Wall Thickness Required For 1.5 Safety Factor (Inches)	T .024 B .020	T .033 B .030	T .045 B .044
Natural Frequency rad/sec	461	509	553
Deflection Of The Shield At The Center Of Gravity (Inches)	.54 x 10 ⁻³	.71 x 10 ⁻³	.92 x 10 ⁻³
Amplification Factor Due To Vibration At 50 cps	1.86	1.60	1.47
Heat Leak Through Neck Tube ² (Watts)	39.6 - 40.9	56.8 - 58.2	80.5 - 81.9
Heat Leak Through Insulation (Watts)	9	9	9
Total System Heat Leak (Watts)	48.6 - 49.9	65.8 - 67.2	89.5 - 90.9

1) Hastelloy-X

2) 1.5 mil variation on the outside diameter is allowed for manufacturing tolerances

FIGURE 3

One potential disadvantage of directly supporting the radiation shield by the shipping container is that the unit must be shipped cold and then heated at the test location.

3.0 DETERMINATION OF EXPECTED DYNAMIC LOADING DURING UNIT TRANSPORTATION AND HANDLING

3.1 Literature Survey to Determine Expected Dynamic Loading

In order to select a HTVIS unit support system for further analysis, the magnitude of the expected loadings which are normally received during shipping and handling must be determined. Two options exist for shipping or handling a unit: 1) the unit can be shipped in a shipping container which will reduce the loads experienced by the unit, or 2) the unit can be shipped without a shipping container and be designed to withstand the shipping and handling dynamic loads. In either case an estimate of the dynamic loads produced during shipping and handling must be made in order to either design a unit or a shipping container.

Dynamic loads produced by the four major modes of transportation (air, truck, ship and rail) are well documented in the literature. (See references 1 and 2.) A great deal of experimental data is presented in these references pertaining to the environmental loads transmitted to the cargo areas for each mode of transportation. Less well defined is the environmental load that can be expected while handling a unit either prior to or subsequent to the actual transport. Categories to be considered under handling include fork-lift truck, hand truck, and shop hoist or winch. Literature sources which present experimental data for these handling conditions could not be located. Because it is expected that much of the handling will be by fork-lift truck as shown in Section 3.2, fork-lift truck tests to determine the expected loadings to a unit were performed and the results are presented in Section 3.3.

Linde and 3M personnel worked together using the available literature and fork-lift truck test data to estimate the dynamic environmental loads that the system would experience. The expected system loads were estimated considering both a wood shipping platform and a metal shipping platform with both supervised and unsupervised shipping and handling. Supervised handling requires an inspector on hand to observe and control the handling methods and equipment used in load and unload operations with the transport vehicles. The presence of an inspector is not required during actual transport from one location to another. For unsupervised handling an inspector is not required.

The estimated g loadings for transport and handling of the HTVIS are presented in Table V. The values presented in this table are estimated to be maximum values. The frequency of occurrence of these maximum values could not be determined because of the lack of correlating test data. The assumptions used in the preparation of Table V are as follows:

1. "g" loadings are dynamic loads. Static loads must be added where applicable for design purposes.

TABLE V

ESTIMATED "G" LOADING FOR SHIPPING AND HANDLING OF INSULATION SYSTEM,
FUELED INSULATION SYSTEM AND COMPLETE SYSTEM

- 14 -

Means of Transport & Description of Load	System in Wood		System in Metal		
	Container on Wood Pallet		Container on Metal Skids		
	Unsupervised	Supervised	Unsupervised	Supervised	
1) Air Shipment					
a) Jet	2.5 @ 800 cps	Same	Same	Same	Ref. 1, Fig. 1
	0.1 @ 50 cps				
b) Prop	6.0 @ 400 cps				
	1.0 @ 50 cps				
2) Truck					
a) Smooth Road	0.2 @ 50 cps	Same	Same	Same	Ref. 1, Fig. 24
	0.35 @ 4 cps				
	0.3 @ 50 cps				
	2.0 @ 10 cps				
3) Ship					
a) Rough Sea	0.5 @ 50 cps	Same	Same	Same	Ref. 1, Fig. 20,21,22,23
	0.9 @ 12 cps				
b) Slam & Emergency	3.0 @ 2.5 cps				
	3.0 @ 50 cps				
4) Fork Lifts - Motorized					
a) Ramming Skid	16	3	200	20	8 fps, unsupervised; 1 fps supervised Ref 2, Fig 41.1: Section 3.3.
b) Drop to Floor	6	1	25	12	1 fps unsupervised; 1/4 fps supervised Ref 2, Fig 41.1: Section 3.3.
c) Jerking Load on Forks	2	1	2	1	Because of cushioning of lift truck, container not important.
d) Transit Over Rough Surfaces	10	2	10	2	Because of cushioning of lift truck, container not important.
5) Fork Lifts - Manual					
a) Ramming Skid	3	1	20	3-1/2	2.5 fps fork lift speed unsupervised 1/2 fps fork lift speed supervised Considering weight fork lift 1/2 weight of load. Ref 2, Fig 41.1

TABLE V (cont'd)

Means of Transport & Description of Load	System in Wood Container on Wood Pallet		System in Metal Container on Metal Skids		
	Unsupervised	Supervised	Unsupervised	Supervised	
b) Drop to Floor	6	1	25	12	Considered same values as motorized unit.
c) Transit over Rough Surfaces	11	6	58	29	1" vertical drop, unsupervised 1/4" vertical drop, supervised Ref 2, Fig 41.1
6) Hand Truck, Rubber Wheels					
a) Ram into solid object	7	1-1/2	70	12	3 fps ram speed unsupervised 1/2 fps ram speed supervised Ref 2, Fig 41.1
b) Transit over Rough Floor	10	5	17	8	2" drop unsupervised; 1/2" drop supervised, assuming .008 time rise plus .010 wood container & .002 metal container time rise; Ref 2, Fig 41.1
7) Hand Truck, Steel Wheels					
a) Ram into Solid Object	7	1-1/2	70	12	Assumed same values as rubber wheel truck
b) Transit over Rough Floor	15	7	40	20	2" drop unsupervised, 1/2" drop supervised, assuming .002 time rise plus .010 wood container & .002 metal container time rise; Ref 2, Fig 41.1
8) Elevator	2.5	2.5	2.5	2.5	Emergency deceleration Ref 2, Table 44.5
9) Hoist, shop type (chain, elec. hydraulic)					
a) Setting down on solid floor	1-1/2	1-1/2	8	8	1/2 fps velocity both supervised & unsupervised. Ref 2, Fig 41.1
10) Winch - dockside	?	?	?	?	No values assigned due to lack of available data and knowledge.
11) Free fall due to normal handling	20	6	100	28	6" edge drop (3" free fall) unsupervised; 1/2" edge drop (1/4" free fall) supervised. Ref 2, Fig 41.1

2. The system platform receives same g loading as transport carrier cargo floor.

3. The system has no horizontal movement when on truck cargo floor. If system is free to move, the g loading would be 70 per cent the value of an equivalent distance free fall (Reference 2, pp. 44-49).

4 Other items on the cargo floor are not free to impact the system.

5. The wooden platform rise-time is 0.010 second vertical and 0.020 second horizontal (Reference time measurement on fork-lift truck test charts). Load magnitude checked against Reference 2, Figure 41.1 for correlation. The metal platform rise time is 0.002 second, Reference 2, Table 41.1.

6. The equivalent free fall distance for a known velocity was established from the relationship $\text{distance} = (\text{velocity})^2 / 2g$.

Transportation by rail was not considered because of the extremely high g loads experienced when coupling standard railroad cars.

From Table V it can be seen that supervised transportation and shipment with the system or a wooden platform provides the lowest dynamic loading to the system with a maximum of only 7g predicted. During unsupervised transport and handling with the system on the wooden platform, the maximum predicted g load is 20g. The maximum supervised metal platform shipping and handling g load is estimated to be 29g while the unsupervised maximum g load for the metal platform is predicted to be 100g. All the g loads experienced by the unit can be reduced by using shock mitigation methods in a shipping container, if necessary.

A survey of the design g loadings used for standard Linde commercial cryogenic fluid containers indicated that these containers are generally designed to withstand in the order of 10g in the horizontal direction and 15g in the vertical direction. These containers employ a shock mitigation device on the base which allows the input g loadings to be higher than the container design loadings.

3.2 Estimation of HTVIS Unit "g" Loadings During Fabrication, Testing and Packaging

To determine the expected g loadings during unit fabrication, testing and packaging, a flow plan (Table VI) was prepared to show each operation on the unit, the tooling required, and the estimated dynamic load encountered. The dynamic load values of Table V were extensively used in estimating the loads for each operation on Table VI. The maximum g loading estimated was 7.5 g for the operation of picking up the unit with a chain hoist and setting the unit on the test stand. The loads produced during these operations can, however, be reduced by use of cushioning devices.

TABLE VI

FABRICATION TESTING AND PACKAGING FLOW PLAN FOR HTVIS

<u>OPR. NO.</u>	<u>DESCRIPTION</u>	<u>TOOLING REQUIRED</u>	<u>PROBABLE CAUSE AND ESTIMATE OF DYNAMIC LOAD</u>
Operations prior to welding inner tube to outer vacuum enclosure are not considered to present any problems which obvious handling and fabrication care will not take care of. Therefore, operations start with insulated radiation shield.			
100	Transport to welding shop Place on floor	Mounted on wrapping stand (wheeled). Fork lift truck.	Insignificant - neck tube supports no weight.
110	Place on support pad in up- right position with chain hoist.	Wooden contoured support pad, chain hoist attached to bolted support member.	Insignificant - neck tube supports no weight.
120	Lower top head onto neck tube to base support.	Hand place	Insignificant
130	Shim base support to correct alignment.	Hand, shim stock	Insignificant
140	Tack weld top head to inner liner per specs.	Heliarc welder	Insignificant
150	Chain hoist into vertical rotating chuck.	Vertical chuck	Same as OPR. No. 300
160	Weld per specification.	Heliarc welder	Insignificant
170	Chain hoist from chuck to wrapping stand in vertical position.	Wrapping stand per OPR No. 100	Same as OPR No. 300
180	Tilt to horizontal position.	---	Insignificant
190	Roll wrapping stand to engage shield support into horizontal chuck.	---	Insignificant

TABLE VI (Cont'd)

OPR. NO.	DESCRIPTION	TOOLING REQUIRED	PROBABLE CAUSE AND ESTIMATE OF DYNAMIC LOAD
200	Slide on lower vacuum enclosure and strap in position	Hand operation. Metal strap to be designed.	Insignificant
210	Tack weld per spec	Heliarc welder	Insignificant
220	Automatically weld seam	Heliarc welder	Insignificant
230	Roll out of chuck and tilt to vertical position	---	Insignificant
240	Pick up with chain hoist by rotating shaft	Chain hoist	Same as OPR No. 300
250	Set on pallet 1/4 fps maximum rate	Wooden contoured pallet sitting on rubber tire hand cart	Impact of unit on pallet 1-1/2 g's. Rise time to maximum acceleration of .018 s.
260	Install neck tube guard	Neck tube guard - to be designed	Insignificant
270	Remove rotating shaft bolts and shaft	Small hand tools	Insignificant
280	Transport unit to test area a. Hand truck into elevator b. Elevator down c. Fork-lift truck from elevator to test area dock Set onto hand truck d. Hand truck into test lab	Hand truck Elevator Fork lift Hand truck	1-1/2 g's 2-1/2 g's 3 g's Values same as supervised terminal handling loads Table V 1-1/2 g's
290	Attach inverting fixture	Inverting fixture - to be designed - small hand tools	Insignificant
300	Attach chain hoist to inverting fixture and raise 1/4 fps maximum velocity	---	Rise time to maximum acceleration .002's 7-1/2 g's. (Could be attenuated with spring on chain or fixture cushioning)

TABLE VI (Cont'd)
TOOLING REQUIRED

PROBABLE CAUSE AND ESTIMATE
OF DYNAMIC LOAD

<u>OPR. NO.</u>	<u>DESCRIPTION</u>	<u>TOOLING REQUIRED</u>	<u>PROBABLE CAUSE AND ESTIMATE OF DYNAMIC LOAD</u>
310	Rotate to upside down position	--	Insignificant
320	Place test stand under unit and lower unit onto stand 1/4 f.p.s maximum velocity	Test stand-to be developed (contoured table)	Impact with stand 7 1/2 g's. (could be attenuated with pad or spring effect)
330	Attach vacuum pump hoses	Small hand tools	Insignificant
340	Evacuate, leak check, getter fill, and temporary seal off.	Hand tools, test equipment	Insignificant
350	Remove vacuum hoses	Hand tools	Insignificant
360	Pick up by chain hoist	Chain hoist	Same as OPR. NO 300
370	Rotate to right side up position	--	Insignificant
380	Set on pallet by chain hoist	Same as OPR. NO. 250	Same as OPR. NO. 250
390	Transport unit to welding shop	Reverse sequence OPR. NO. 280	Same as OPR. NO. 280
400	Pick up with chain hoist and rotate to upside down position	--	Same as OPR. NO. 300
410	Set on test stand (same as OPR. NO. 310)	Same as OPR. NO. 320	Same as OPR. No. 320
420	Perform final seal off weld	Helic arc welder	Insignificant
430	Pick up with chain hoist and rotate to right side up position	--	Same as OPR. NO. 300
440	Set on pallet	Same as OPR. NO. 250	Same as OPR. NO. 250

TABLE VI (Cont'd)

<u>OPR. NO.</u>	<u>DESCRIPTION</u>	<u>TOOLING REQUIRED</u>	<u>PROBABLE CAUSE AND ESTIMATE OF DYNAMIC LOAD</u>
450	Transport unit to test area (Same as OPR. NO. 280)	Same as OPR. NO. 280	Same as OPR. NO. 280
460	Attach chain hoist to inverting fixture and raise unit 1/4 fps maximum velocity	--	Same as OPR. NO. 300
470	Remove hand cart place pallet on floor and lower unit into pallet 1/4 fps velocity maximum	--	Impact of unit on pallet 2 1/2 g's Rise time to maximum acceleration .02 seconds.
480	Remove inverting fixture and neck tube guard	Small hand tools	Insignificant
490	Insert heater into neck tube and attach instruments	Small hand tools	Insignificant
500	Run heat leak tests per specifications	--	Insignificant
510	Remove instruments and heater	Small hand tools	Insignificant
520	Attach shipping plate to radiation plug screw holes of neck tube	Shipping plate to be designed to incorporate lifting means	Insignificant
530	Raise unit with chain hoist	Chain hoist	Same as OPR. NO. 300
540	Lower unit into shipping cont- ainer. Secure into container as design indicates close container	Shipping container - to be designed, small hand tools.	Insignificant because of support means.

TABLE VI (Cont'd)

If unit is a prototype or qualification unit, it will be shipped to Dynamic Test Site and Steps 550-650 will be required. Insufficient information is available for the assignment of load values at this time.

If the unit is a production unit it will be shipped to 3M Company.

In either case the shipping and handling values assigned for terminal operations will apply for all movements of the packaged item. In plant transport of the packaged system can be assumed to be of the supervised level.

Operations after unit in shipping container is ready for opening up by Dynamic Test Site:

<u>OPR. NO.</u>	<u>DESCRIPTION</u>	<u>TOOLING REQUIRED</u>	<u>PROBABLE CAUSE AND ESTIMATE OF DYNAMIC LOAD</u>
550	Open container. Remove securing members as indicated by container design	Small hand tools	Insignificant
560	Raise out of container. Remove container.		
570	Place in dynamic test fixture. Remove shipping plate		
580	Mount fixture into dynamic load machine. Place heater in neck tube and instrument unit		
590	Conduct tests per outline.		
600	Remove instruments, heater. Remove fixture from dynamic load machine		
610	Reinstall shipping plate		
620	Attach chain hoist and remove from test fixture.		
630	Lower into shipping container		
640	Secure into container. Close container		
650	Transport to loading docks.		

TABLE VI (Cont'd)

Return shipment to Linde Division by common carrier. Terminal load values apply for return shipment handling. After receipt of unit by Linde the following steps are taken:

<u>OPR. NO.</u>	<u>DESCRIPTION</u>	<u>TOOLING REQUIRED</u>	<u>PROBABLE CAUSE AND ESTIMATE OF DYNAMIC LOAD</u>
660	Transport packaged unit to test area	Same as OPR No. 280	Same as OPR No. 280
670	Open container. Remove holding members. Raise unit with chain hoist attached to shipping plate	Small hand tools chain hoist	Same as OPR No. 300
680	Place on container pallet	Pallet from OPR No. 250	Same as OPR No. 250
690	Remove shipping plate	Small hand tools	Insignificant
700	Remove neck tube guard	Small hand tools	Insignificant

Operations Numbers 490 to 540 are repeated here.

3.3 HTVIS Unit Handling On a Fork-Lift Truck, Experimental Tests

As shown on Table VI, it is projected that there will be considerable handling of a HTVIS unit with a fork-lift truck during the fabrication, testing and shipping. Because no fork-lift truck handling data could be found in the literature, fork-lift truck tests were performed to determine the magnitude of dynamic loads that are produced during handling. To conduct these tests a mock-up HTVIS unit was fabricated to simulate both 300-pound and 500-pound units. This unit was mounted on a standard wood pallet, instrumented with accelerometers and tested using a commercial fork-lift truck. Dynamic loadings were determined for handling in the following modes:

1. Ram pallet with fork-lift truck.
2. Jerk unit up from floor (pallet on truck forks).
3. Stop unit on forks.
4. Drop unit on floor (pallet on truck forks).
5. Traverse obstacles on the floor at normal speeds (pallet on truck forks).

These tests and results should not be considered all-comprehensive in that they are representative only of the weights of the units tested, the type of pallet used, and the model of the fork-lift truck used.

The largest dynamic g-loads measured were produced during ramming the pallet with the fork-lift truck to simulate the fork engagement with the pallet. As an example, at a ram velocity of 8 ft/sec., a horizontal g-loading of 16.3 g's were experienced by the unit. Perhaps a more realistic maximum ram speed would be 4 ft/sec., which produced g loads ranging from 4 to 10 g's. When dropping the unit on the floor with the pallet on the truck forks, the vertical g load was not greater than 6 g's regardless of the drop height. During a simulated, normal handling operation (supervised), the maximum horizontal dynamic load measured was 1.6 g's. When traversing obstacles 1-3/4 inches high at a normal speed, the vertical g loads measured ranged from 2.8 g to 8.4 g depending on the obstacle width, spacing and weight of the unit.

The test apparatus used during the conduct of the experimental tests was follows:

1. Simulated HTVIS unit mounted on a standard wood pallet filled with wet sand to weigh 300 pounds. Two hundred pounds of metal weights were added for the 500-pound test. See Figures 4 and 5.
2. Acceleration Measurement
 - a. Statham $\pm$ 20g accelerometer, serial No. 10815 mounted on top cross angle of unit (Figure 4).
 - b. Statham $\pm$ 200g accelerometer, serial No. 6150 mounted on the center of wood pallet in front of unit (Figure 4).

Figure 4. Simulated HTVIS Unit for Fork-Lift Truck Tests (300 Pounds)

Figure 5. Simulated HTVIS Unit for Fork-Lift Truck Tests (500 Pounds)

c. Sandborn twin viso model 60-1300, serial No. 891 (recorder). Used to record accelerations.

3. Velocity Measurement

- a. Statham $\pm$ 1 psid pressure transducer.
 - b. 3/8-inch O.D. rubber tubing parallel spaced 10 ft. apart on floor in front of unit.
 - c. Sandborn twin viso model 60-1300 (recorder), automatic second timer. Used to record pressure pulses.
4. Fork-lift truck, Allis-Chalmers, model FPL-40-24-2PS, weight 7190 pounds, capacity 4000 pounds.

All data accumulated during the fork-lift truck tests are presented on Table VII, which was prepared from the original data sheet.

3.3.1 Ram Pallet With Fork-Lift Truck Test

This test was conducted to determine the dynamic g-loadings experienced by the unit and the pallet when engaging the forks of a moving fork-lift truck with the pallet. Acceleration transducers were mounted to monitor the horizontal accelerations for this test. The fork-lift truck ram velocity was measured by recording pressure pulses received by running over two parallel rubber tubes spaced 10 feet apart located in front of the unit. The pressure pulses caused by running over the rubber tubes were recorded on a Twin-Viso recorder which automatically records one second time intervals. The speed of each run was determined by dividing the distance between the rubber tubes (10 feet) by the required time to traverse this distance.

Test runs were conducted using both the 300-pound unit, Run No. 23-32, and the 500-pound unit, Run No. 34-38 (Table VII). The ram velocity ranged from 1.68 ft/sec. to a maximum of 8 ft/sec. After the pallet was rammed the fork-lift truck operator applied the brakes. The distance the pallet moved after ramming is shown on Table VII for each run along with the measured horizontal g-loads and ram velocity. The fork-lift truck ram speed is plotted vs. the horizontal acceleration on Figure 6 for all the runs conducted.

Figure 7 shows the wooden pallet after the ram tests. It can be seen that, with the somewhat high ram speed employed during these tests, the pallet was visibly damaged. In fact, the top end board was lifted from the stringers. The dark areas on the end board are where the lift truck forks contacted the pallet.

3.3.2 Jerk Unit Up From the Floor

This test was conducted to determine the dynamic g-loadings the unit would experience after the fork-lift truck was engaged with the pallet and the operator started to lift the unit from the floor. The accelerometers were mounted to read the acceleration in the vertical direction for this

Allis Chalmers
Model FFL-40-24-2PS
It 7190 4000r CAP

TABLE VII

12-8-66
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FORK LIFT TRUCK DYNAMIC TESTS ON SIMULATED HTVIS MOUNTED ON WOOD PALLET

Run No.	Event	Pallet "g" Dynamic	Unit "g" Dynamic	Distance	Time Sec.	Speed Ft/Sec	
1	Stop on Forks			50" - 40"			
2	"	1.6 (V)	1.58 (V)	49" - 39"			
3	"	1.3	1.35	39" - 12"			
4	"	1.25	1.25	67" - 7"			
5	"	1.5	1.45	57" - 52.5"			
6	"	1.6	1.6	52.5" - 49"			
7	"	1.5	1.5	44.5" - 36"			
8	Drop on Floor	4.5	4.4	1/4"			
9	"	4.4	4.8	1/4"			
10	"	4.3	4.6	1/2"			
11	"	4.6	5.3	1"			
12	"	4.0	4.1	2"			
13	"	4.25	4.5	2"			
14	"	3.3	3.7	4"			
15	"	2.9	3.4	6"			
16	"	2.3	3.8	8"			
17	"	3.4	3.9	12"			
18	"	3.6	3.9	24"			
19	"	3.4	3.7	36"			
20	"	3.4	3.4	48"			
21	Jerk up from Floor	0.4start/1.1stop	0.4start/1.1stop				
22	"	1.0 /1.0	0.8 /1.0				Fast Engine
23	Ram with Truck	6.0 (H)	5.2 (H)	10 Feet	2.52	3.97 ft/sec	
24	"	10.0	10.2	"	2.45	4.08 "	Pallet Move 13"
25	"	7.6	9.2	"	3.6	2.78 "	8-1/8"
26	"	2.1	2.8	"	5.6	1.79 "	4-3/8"
27	"	4.5	5.6	"	3.55	2.82 "	4-7/8"
28	"	6.0	8.3	"	3.17	3.16 "	7-3/4"
29	"	4.0	7.0	"	3.95	2.53 "	9-3/4"
30	"	2.8	3.6	"	5.85	1.71 "	4-1/2"
31	"	7.2	12.5	"	1.57	6.37 "	29-3/8"
32	"	13.9	16.3	"	1.25	8.0 "	37"
33	Normal	1.2 max (H)	1.6 max (H)				

L. R. Niendorf
Linde Division
Tonawanda, New York

SIMULATED HTVIS UNIT MOUNTED WOOD PALLET

[illegible]

Figure 6

Figure 7. Simulated HTVIS Unit After Ram Tests

test. These tests were conducted using both the 300-pound unit, Run No. 21-22, and the 500-pound unit, Run No. 50 (Table VII). In all runs the fork-lift operator opened the fork-lift control valve full open to achieve maximum jerking. Table VII presents the measured vertical accelerations for both the unit and the pallet as determined in these tests. The highest g-loading measured was 1.1 g.

3.3.3 Stop Unit on Forks

This test was conducted to determine the dynamic loads on the unit and the pallet when the system is stopped on the forks of the fork-lift truck after traversing downward a specified distance prior to contacting the floor. The accelerometers were installed to indicate the acceleration in the vertical direction upon stopping for this test.

Run No. 1-7 was conducted on the 300-pound unit and Run No. 51-52 was conducted on the 500-pound unit (Table VII). In all runs the fork-lift truck operator opened the fork control valve full open to start the fall and rapidly closed the valve for the stop to achieve maximum loadings. Figure 8 shows the fork-lift truck with the unit on the forks after one of the stop tests. The six-foot rule shown was used to measure the drop height. The accelerometer recorders are shown on the right side of the figure. The measured dynamic loads upon stopping the forks are presented in Table VII along with the distance traversed prior to stop. In general, the g loads were approximately 1.5 g regardless of the distance traversed because the fork lowering speed is velocity limited.

3.3.4 Drop Unit on Floor (Pallet on Truck Forks)

This test was conducted to determine the dynamic loads that the unit and the pallet would experience when the unit was dropped on the floor from various heights while on the forks of the fork-lift truck. The accelerometers were mounted to indicate the acceleration in the vertical direction when the unit impacted the floor. Run No. 8-20 were conducted with the 300-pound unit and Run No. 39-49 were conducted with the 500-pound unit. The drop height before impact and the corresponding measured vertical g loads are presented in Table VII. During this series of tests the fork lift truck operator maintained the fork control valve in the wide open position to achieve maximum drop velocity and did not close the control valve until after the unit had impacted the floor. There is a great deal of scatter in the data with the g loads ranging from less than 1g to a maximum of 5.3 g's for these drop tests which is most likely due to the unit not contacting the floor flat in all tests. There does not appear to be a correlation between the drop height and the dynamic loads received by the unit as would perhaps be expected as the drop is velocity limited due to the hydraulic system of the fork-lift truck.

3.3.5 Traverse Obstacles on the Floor

This test was conducted to determine the dynamic loads the unit and pallet would experience when the fork-lift truck traversed obstacles placed on the floor as might be expected during unsupervised unit handling. The accelerometers were mounted to indicate the acceleration in the vertical direction during this test.

Figure 8. Simulated HTVIS Unit on Forks of Fork-Lift Truck

As shown on Table VII, Run No. 53-56 were conducted with the 500-pound unit and Run No. 57 and 58 with the 300-pound unit. The obstacles traversed consisted of metal channels 1-3/4 inch high measured from the floor and 5-1/2 inches wide. These channels were rectangular in shape and sufficient length to allow both front wheels of the fork-lift truck to traverse the obstacles at one time. The obstacles were set up in the following patterns:

1. Single channel (Run 53-54)
2. Two channels side by side (11 in. wide) (Run 55)
3. Two channels parallel to each other with a 6-inch gap between channels (Run 56-58)

The traverse tests were conducted at normal speeds ~ 2 fps. The exact speed could not be measured as the accelerometer electrical leads were not long enough to allow the truck to traverse through the speed-measuring instrumentation. The most severe loadings were measured while testing the 300-pound unit and ranged up to 8.4 g's as shown on Table VII (Run 56).

One normal simulated handling run was made (Run No. 33) with the 300-pound unit. Accelerometers were mounted to indicate the dynamic loading in the horizontal direction. During this run the fork-lift truck engaged the pallet, lifted the pallet, stopped the pallet on the forks, moved forward and to the rear, set the pallet on the floor, and disengaged the truck from the pallet. During this simulated supervised handling run, the maximum horizontal g load was 1.6 g's.

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APPENDIX A

PARAMETRIC STUDY FOR VARIOUS SUPPORT SYSTEM CONCEPT

PART I

NECK TUBE SUPPORT SYSTEM

The neck tube wall thicknesses and corresponding heat losses were calculated for the neck tube support system considering dynamic input loadings of greater than 3g. The following example shows the calculation of the 5g case. The consideration that the nuclear shield weight increases and the center of gravity shifts with heat losses greater than 30 watts was factored into the calculation.

Safety factors for the neck tube for buckling in combined bending and internal pressure were calculated for increased neck tube wall thicknesses at 3g, 5g, 8g, and 10g dynamic loading. Graphical presentation of the results is shown on Figure A-1. The method of calculation is presented in Appendix B, page B-6. Safety factors for the neck tube for buckling in combined transverse shear and internal pressure were calculated for increased neck tube wall thicknesses at 3g, 5g, 8g, and 10g dynamic loading. Graphical presentation of the results is shown on Figure A-2. The method of calculation is presented in Appendix B, page B-6. Since the critical buckling strength both in bending and in shear is a function of the geometry and material of the neck tube only, the critical buckling strength is the same for all g-loading. The applied bending stress and shear stress increase linearly with increased g-loading, and therefore the safety factor which is the ratio of critical buckling strength to applied stress is inversely proportional to increased g-loading.

Calculations for 5g dynamic loading

By trial and error solution as presented in Appendix B, page B-20.

Assume 1.56 amplification factor

$$SF = 1.5 \times 1.56 = 2.34$$

From Figures A-1 and A-2

Top neck tube thickness = .030 in.

Bottom neck tube thickness = .027 in.

$$\begin{aligned}
I_{av} &= 1.3391 \text{ in.}^4 \\
A_{av} &= .4410 \text{ in.}^2 \\
l_1 &= 2.5 \text{ in.} \\
l_2 &= 4.52 \text{ in.} \\
l_{in} &= 2.78 \text{ in.} \\
W &= 5(264) = 1320 \text{ lbs.}
\end{aligned}$$

The deflection at the center of gravity,

$$\delta_2 = .00712 \text{ in.}$$

$$\omega_n = 521 \text{ rad/sec}$$

$$\omega/\omega_n = .6026$$

$$y_o/a_o = .5704$$

$$y_o = \frac{5}{3} (.01175) (.5704) = .0117$$

$$AF = y_o/\delta_2 = \frac{.0117}{.00712} = 1.56 \text{ Amplification Factor}$$

The heat leak,

$$q_{min} = k_m A \frac{T_2 - T_1}{L} = 121.8 \frac{.4410}{144} \frac{1325 - 150}{2.5} \frac{1}{3.41} = 51.4 \text{ watts}$$

where,

$$k_m = 121.8 \frac{\text{Btu-in}}{\text{ft}^2\text{-Hr}^\circ\text{F}} \quad (\text{From Figure A-3, and Appendix D})$$

Due to the increased heat loss over 30 watts the fuel loading increases and the shield weight increases. The increased shield weight requires a thicker neck tube, which increases the heat leak loss. By trial and error solution, assume that the increased heat leak is 60 watts.

The fuel load increase is,

$$60 - 21 = 39 \text{ watts}$$

The total required fuel loading is,

$$203 + 39 = 242 \text{ watts}$$

The total fuel load requirement was 203 watts with 30 watts total system heat leak (21 watts through the neck tube and 9 watts through the insulation). From the chart supplied by the 3M Company (letter of November 22, 1966 to Mr. L. R. Niendorf) the shield weight corresponding to 242 watts fuel loading is 266 lbs. The weight at the shield plus the weight of the fuel capsule is,

$$266 + 12 = 280 \text{ lbs.}$$

Since the center of gravity of the shield shifts away from the neck tube .135" per 20 watts fuel loading increase, the change in distance to the center of gravity is,

$$39/20 (.135) = .263 \text{ in.}$$

$$\begin{aligned} \text{Then, } \ell_1 &= 2.5 \text{ in.} \\ \ell_0 &= 4.52 + .263 = 4.78 \text{ in.} \\ \ell_1 + \ell_2 &= 7.28 \text{ in.} \end{aligned}$$

The corrected moment at the cold end of the neck tube is,

$$M_c = M (5g) \frac{7.28}{7} \frac{280}{264} = 1.103 M (5g)$$

The corrected safety factor on original design is,

$$SF_c = \frac{\text{SF at 5g from Figure A-1}}{1.103}$$

By trial and error solution, assume 1.62 amplification factor.

Safety factor required at 5g loading with increased weight and CG shift,

$$SF = 1.5 \times 1.62 \times 1.103 = 2.68$$

From Figure A-1

Top neck tube thickness = .033

The corrected safety factor in shear buckling is,

$$SF_c = \frac{264}{280} SF \text{ (From Figure A-2)}$$

$$SF = 1.5 \times 1.62 \times 1.06 = 2.57$$

From Figure A-2

Bottom neck tube thickness = .030 in.

$$\begin{aligned} t_{av} &= .0315 \text{ in.} & \ell_1 &= 2.5 \text{ in.} \\ I_{av} &= 1.4828 \text{ in}^4 & \ell_{in} &= 2.78 \text{ in.} \\ A_{av} &= .4877 \text{ in}^2 & \ell_2 &= 4.78 \text{ in.} \\ W &= 5 (280) = 1400 \text{ lbs.} \end{aligned}$$

The deflection of the shield at the center of gravity,

$$\delta_2 = .00748 \text{ in.}$$

$$\omega_n = \left[\frac{5 (32.2) (12)}{.00748} \right]^{1/2} = 509 \text{ rad/sec}$$

$$\omega/\omega_n = .6168$$

$$y_0/a_0 = .6143$$

$$C = \frac{5}{3} (.01175) (.6143) = .01203$$

$$AF = .01203/.00748 = 1.60$$

Heat leak through neck tube

$$q_{\min} = 121.8 \frac{.4877}{144} \frac{1325-150}{2.5} \frac{1}{3.41} = 56.8 \text{ watts}$$

Allowing 1.5 mil variation on the outside diameter for manufacturing tolerances

$$t_{av} = .0315 + .00075 = .03225 \text{ in.}$$

$$A_{av} = .4994 \text{ in}^2$$

$$q_{\max} = 58.2 \text{ watts}$$

PART II

NECK TUBE WITH THREE TENSION ROD SUPPORT SYSTEM

Calculations for 5g dynamic loading

The method of calculations for this support system is presented in Appendix B. page B-22.

For .0095 in. thick straight neck tube and 15/64 diameter tension rods the force in the tension rod is calculated with the following data:

l_1	$= 2.5 \text{ in.}$	A	$= .1464 \text{ in}^2$
l_{in}	$= 2.78 \text{ in.}$	E	$= 24.8 \times 10^6 \text{ psi (Hastelloy-X)}$
l_2	$= 4.52 \text{ in.}$	d_R	$= 15/64 \text{ in.}$
l_3	$= 4.16 \text{ in.}$	A_R	$= .04311 \text{ in}^2$
α	$= 30^\circ$	l_R	$= 2.5 \text{ in.}$
$\cos \alpha$	$= .866$	E_r	$= 24.8 \times 10^6 \text{ psi (Hastelloy-X)}$
I	$= .4412 \text{ in}^4$		

The force in the tension rod is $P = 795.7 \text{ lbs.}$

The deflection at the center of gravity is $\delta = .00137 \text{ in.}$

$$\omega_n = 1187 \text{ rad/sec}$$

$$\omega/\omega_n = .2645$$

$$y_0/a_0 = .07523$$

$$v_0 = .00147$$

$$A_1 = 1.01 \text{ (amplification factor due to vibration at 50 cps)}$$

The amplification factor due to shock for the neck tube support system was assumed 1.3 maximum. (See Ref. 5, Figure 5.9, page 117)

The allowable stress in the tension rods for 1.5 safety factor,

$$\sigma_{all} = \frac{38,920}{1.5 \times 1.3} = 19,958 \text{ psi}$$

The stress in the tension rod,

$$\sigma = \frac{795.7}{.04311} = 18,459 \text{ psi}$$

The heat leak through the .0095 in. thick neck tube,

$$q_{\min} = 121.8 \frac{.1464}{144} \frac{1325-150}{2.5} \frac{1}{3.41} = 17.0 \text{ watts}$$

Allowing 1.5 mil variation on the outside diameter for manufacturing tolerances, the maximum heat leak through the neck tube ($t_{av} = .01025"$, $A_{av} = .1580 \text{ in}^2$)

$$q = 121.8 \frac{.1580}{144} \frac{1325-150}{2.5} \frac{1}{3.41} = 18.4 \text{ watts}$$

The heat leak through the three tension rods,

$$q = (3) 121.8 \frac{.04311}{144} \frac{1325-150}{2.5} \frac{1}{3.41} = 15.0 \text{ watts}$$

The heat leak through the neck tube and the three tension rods applying manufacturing tolerances is,

$$\begin{aligned} q_{\min} &= 32.1 \text{ watts} \\ q_{\max} &= 33.4 \text{ watts} \end{aligned}$$

Due to the increased heat loss the fuel load and shield weight increases, by trial and error solution, assume that the increased heat loss will be 34 watts.

The increase in fuel loading is,

$$34 - 21 = 13 \text{ watts}$$

The total required fuel loading is

$$203 + 13 = 216 \text{ watts}$$

The increased shield weight is 257 lbs. The weight of the shield plus fuel capsule is 269 lbs.

The increase in the distance to the center of gravity is,

$$\frac{13}{20} (.135) = .0877 \text{ in.}$$

$$\begin{aligned} l_2 &= 4.52 + .0877 = 4.607 \approx 4.61 \text{ in.} \\ l_3 &= 4.16 + .0877 = 4.247 \approx 4.25 \text{ in.} \end{aligned}$$

The force in the tension rod with the increased distance to the center of gravity is $P = 809.1 \text{ lbs.}$ The stress in the tension rod is,

$$\sigma = \frac{809.1}{.04311} = 18,770 \text{ psi}$$

Since the stress in the tension rod when supporting the increased shield weight is less than the allowable stress 19,958 psi, the diameter of the tension rod will not change.

The .0095" wall thickness of the neck tube is checked for strength in shear buckling using the 1.3 maximum amplification factor due to shock loading, the shear force in the neck tube is,

$$1.3 (W - P) = 1.3 (5 \times 269 - 809.1) = 696.6 \text{ lbs.}$$

The applied shear stress in the neck tube for 1000 lbs. load and different neck tube wall thicknesses have been calculated by the formula given in Appendix B, page B-7, and results were plotted on Figure A-4. The applied shear stress for .0095" wall thickness neck tube with 696.6 lbs. shear load acting on it is obtained from Figure A-4.

$$\tau = 696.6/1000 (13,630) = 9494 \text{ psi}$$

The shear buckling stress (From Appendix B page B-7 is given by,

$$\tau_{SCR} = 1.25 (.00056) E_r = (.7) 10^{-3} E_r$$

At the hot end of the neck tube

$$E = E_r = 21.4 \times 10^{-6} \text{ psi}$$

$$\tau_{SCR} = (.7) \times 10^{-3} 21.4 \times 10^{-6} = 14,980 \text{ psi}$$

The maximum allowable shear stress in the neck tube for 1.5 safety factor (hot end)

$$\tau_{allow} = \frac{\tau_{SCR}}{1.5} = \frac{14,980}{1.5} = 9986 \text{ psi}$$

At the cold end of the neck tube

$$E = E_r = 28.3 \times 10^{-6} \text{ psi}$$

$$\tau_{SCR} = (.7) 10^{-3} 28.3 \times 10^{-6} = 19,810 \text{ psi}$$

The maximum allowable shear stress in the neck tube for 1.5 safety factor (cold end)

$$\tau_{allow} = \frac{\tau_{SCR}}{1.5} = \frac{19,810}{1.5} = 13,206 \text{ psi}$$

Since the applied shear stress 9494 psi is less than the allowable shear stress 9986 psi (hot end) and 13,206 psi (cold end) the .0095" thickness neck tube is safe in shear buckling.

PART III

NECK TUBE WITH THREE COMPRESSION ROD SUPPORT SYSTEM

Calculations for 10g dynamic loading

The method of calculations for this support system is the same as presented for the tension rod support system in Part II. The compression rods will be installed at an angle α to the horizontal in order to support the shield both in the cold and hot conditions. The angle α is calculated as per the method presented in Appendix B, page B-26.

The linear thermal expansion of zirconium dioxide from Reference 11 is 0.325% at a mean rod temperature of 812°F (see Figure A-5 or Appendix D for calculations).

$$\frac{\Delta l}{l} 100 = 0.325$$

$$\Delta l = 0.00325 l = \alpha_m \Delta T l$$

$$a = 1 + \alpha_m \Delta T = 1.0 + 0.00325 = 1.00325$$

The thermal expansion of the rod attachment (Hastelloy-X) in the radial direction

$$C = \alpha R \Delta T \qquad R = 3.125 \text{ (layout)}$$

$$C = (8.76) 10^{-6} (3.125) (1435 - 70) = 0.0373 \text{ in.}$$

Thermal expansion of the shield and neck tube in the axial direction:

Neck tube,

$$\Delta T = 812 \text{ (mean temperature)} - 150 = 662^\circ\text{F}$$

$$\Delta l_N = (8.76) 10^{-6} (3.156) 662 = 0.01830 \text{ in.}$$

Shield,

The linear thermal expansion coefficient of annealed U-8 moly from the chart supplied by the 3M Company, letter to L. R. Niendorf dated November 8, 1966.

$$\alpha = 1.40\% \text{ at } 1360^\circ\text{F} = 1820^\circ\text{R}$$

$$\alpha = \frac{.0140}{(1360-70)} = 10.85 \times 10^{-6} \text{ in/in}^\circ\text{F}$$

Thermal expansion of nuclear shield,

$$\Delta l_s = 10.85 \times 10^{-6} [5.7 + 2.6 (.866)](1360 - 70) = .1112 \text{ in.}$$

$$b = \Delta l_n + \Delta l_s = .0183 + .1112 = .1295 \text{ in.}$$

The average shield temperature is taken as

$$T_{av} = \frac{1435 + 1285}{2} = 1360^\circ\text{F}$$

$$a = 1.00325$$

$$C_1 = 2.16 \text{ (from layout)}$$

$$b = 0.1295$$

$$C_2 = C_1 - C = 2.16 - .0373 = 2.122$$

$$c = 0.0373$$

With the above values of a,b,C₁ and C₂ the distance X is calculated,

$$X = +.652 \text{ or } + 39.13$$

$$\tan \alpha = \frac{.652}{2.16} = .301 \quad \alpha = 16.7^\circ$$

$$= \frac{.652}{\sin 16.7} = 2.27 \text{ in.}$$

For a straight neck tube of .0095 in. wall thickness and 13/32 dia. compression rods, the force in the compression rod is P = 1594 lb.

where,

$$l_1 = 2.5 \text{ in.}$$

$$l_R = 2.27 \text{ in.}$$

$$l_{in} = 2.78 \text{ in.}$$

$$d_R = 13/32 \text{ in.}$$

$$l_2 = 4.52 \text{ in.}$$

$$A_R = .1295 \text{ in}^2$$

$$l_3 = 4.16 \text{ in.}$$

$$E_R = 22.0 \times 10^6 \text{ psi}$$

$$I = .4412 \text{ in}^4$$

$$\alpha = 16.6$$

$$A = .1464 \text{ in}^2$$

$$\tan \alpha = .301$$

$$E_N = 24.8 \times 10^6 \text{ psi}$$

$$\cos \alpha = .956$$

$$c = 3.0 \text{ in.}$$

The stress in the compression rod is

$$\sigma = \frac{1594}{.1295} = 12,308 \text{ psi}$$

The allowable stress in the compression rods with 1.5 safety factor and 1.2 amplification factor due to shock,

$$\sigma_{all} = \frac{30,000}{1.5 \times 1.2} = 16,666 \text{ psi}$$

$$\sigma_{yield} = 30,000 \text{ psi}$$

The wall thickness of the neck tube is checked for strength in shear buckling. The shear force in the neck tube with 1.2 amplification factor (AF) is

$$(AF)(W-P) = 1.2 [10(264) - 1594] = 1255 \text{ lb.}$$

At the cold end of the neck tube,

$$\tau = \frac{1000}{1255} 13,206 = 10,522 \text{ psi}$$

$$\text{where } \tau_{allow} = 13,206 \text{ psi}$$

The required thickness from Figure A-4 is $t = .013$.

At the hot end of the neck tube,

$$\tau = \frac{1000}{1255} 9986 = 7956 \text{ psi}$$

$$\tau_{allow} = 9986 \text{ psi}$$

The required thickness from Figure A-4 is $t = .017$ inch. The average wall thickness is $t_{av} = .015$ inch.

The heat leak through the neck tube,

$$q = 121.8 \frac{.2314}{144} \frac{(1325 - 150)}{2.5} \frac{1}{3.41} = 26.9 \text{ watts}$$

The heat leak through the compression rods,

$$q = 3(1.008) \frac{.1295}{144} \frac{1325 - 150}{2.27} \frac{12}{3.41} = 4.93 \text{ watts}$$

$$k = 1.008 \frac{\text{Btu}}{\text{ft} \cdot \text{hr} \cdot ^\circ\text{F}} \quad (\text{Appendix B, page B-30})$$

The total heat leak is

$$q_{\min} = 26.9 + 4.93 = 31.8 \text{ watts}$$

Allowing 1.5 mil variation on the outside diameter of the neck tube for manufacturing tolerances ($t_{\text{av}} = .01575$, $A_{\text{av}} = .2431 \text{ in}^2$),

$$q_{\max} = 28.3 + 4.9 = 33.2 \text{ watts}$$

The total system heat leak is

$$33.2 + 9 = 42.2 \text{ watts}$$

The fuel load increase is 12.2 watts. The total required fuel loading is $203 + 12 = 215 \text{ watts}$.

The weight of the shield and fuel capsule,

$$256 + 12 = 268 \text{ lb.}$$

The increase in the distance to the center of gravity is

$$\frac{12}{20} (.135) = .081 \text{ in.}$$

$l_2 = 4.52 + .081 = 4.60 \text{ in.}$	$E_N = 24.8 \times 10^6 \text{ psi}$
$l_3 = 4.16 + .081 = 4.24 \text{ in.}$	$e = 3.0 \text{ in.}$
$l_1 = 2.5 \text{ in.}$	$l_R = 2.27 \text{ in.}$
$l_{\text{in}} = 2.78 \text{ in.}$	$d_R = 13/32 \text{ in.}$
$t_{\text{av}} = .015 \text{ in}$	$A_R = .1295 \text{ in}^2$
$I_{\text{av}} = .6990 \text{ in}^4$	$\alpha = 17.7^\circ$
$A_{\text{av}} = .2314 \text{ in}^2$	$\tan \alpha = .301$
$w' = 268 \text{ lb.}$	$\cos \alpha = .956$
	$E_r = 22 \times 10^6 \text{ psi}$

The force in the compression rod with the increased distance to the center of gravity, the increased weight and the increased neck tube wall thickness is $P = 1592 \text{ lb.}$ The stress in the rod is

$$\sigma = \frac{1592}{.1295} = 12,295 \text{ psi}$$

The shear force in the neck tube is

$$1.2 [10(267) - 1592] = 1293 \text{ lb.}$$

At the cold end of the neck tube

$$\tau = \frac{1000}{1293} 13.206 = 10,213 \text{ psi}$$

The required wall thickness from Figure A-4 is $t = .013 \text{ in.}$

At the hot end of the neck tube,

$$= \frac{1000}{1293} 9986 = 7723 \text{ psi}$$

The required wall thickness from Figure A-4 is $t = .017 \text{ in.}$

The neck tube wall thickness and the compression rod diameter are not affected by the increase in fuel load and shield weight.

PART IV

BELLOWS WITH SIX COMPRESSION ROD SUPPORTS

Calculations for 10g dynamic loading

The stresses in the compression rods are calculated from the following equations.

(a) Vertical Position

In the vertical position the shield is supported by three compression rods. The force in each rod will be

$$P = \frac{W}{3 \sin \alpha}$$

The stress in each compression rod is

$$= \frac{P}{A} = \frac{W}{3 \sin \alpha A}$$

(b) Horizontal Position

In the horizontal position the shield is supported by two compression rods. The force in the compression rod is

$$P = \frac{W}{2 \cos \alpha}$$

The stress in each rod is

$$\sigma = \frac{W}{2 \cos \alpha A}$$

In order to determine whether the vertical position or the horizontal position is critical the value of α is calculated.

The angle α is calculated according to the method presented in Appendix B, page B-26.

The linear thermal expansion of zirconium dioxide and the annealed U-8 moly shield have been calculated in Part III of this report. Results were as follows.

$$a = 1.00325 \quad (\text{zirconium dioxide compression rods})$$

$$\alpha = 10.85 \times 10^{-6} \text{ in/in}^\circ\text{F} \quad (\text{annealed U-8 moly shield})$$

For this support system the axial thermal expansion of the compression rod is equal to only half the thermal expansion of the shield and therefore the value of b is recalculated.

$$b = 10.85 \times 10^{-6} \left[1.8 + 2.6 (.866) \right] (1360-70) = .0567 \text{ in}$$

The radial expansion of the shield is

$$c = 10.85 \times 10^{-6} (3.125) (1360-70) = .0437 \text{ in}$$

With the above values of a, b, and c and

$$\begin{aligned} c_1 &= 2.16 \\ c_2 &= c_1 - c = 2.16 - .043 = 2.117 \end{aligned}$$

The distance X is calculated

$$X = + 2.11 \text{ or } + 15.29$$

$$\tan \alpha = \frac{2.11}{2.16} = .976$$

$$\begin{aligned} \alpha &= 44.4^\circ \approx 45^\circ \\ \sin \alpha &= \cos \alpha = .7071 \end{aligned}$$

$$\ell = \frac{2.11}{.7071} = 2.98 \approx 3.0 \text{ in}$$

Since $\alpha = 45^\circ$, the critical loading condition occurs when the unit is in the horizontal position.

The allowable stress in the rod for 1.5 safety factor and 1.2 amplification factor due to shock was calculated in Part III of this report. ($\sigma = 16,666$ psi)

The applied stress in a 13/32 diameter compression rod for 10g loading is,

$$\sigma = \frac{10 W}{2 \cos \alpha A} = 14,415 \text{ psi}$$

where,

$$\begin{aligned} W &= 264 \text{ lbs.} \\ \cos \alpha &= .7071 \\ A &= .1295 \text{ in}^2 \end{aligned}$$

Calculate amplification factor due to vibration at 50 cps.

The strain in the compression rod,

$$\epsilon = \frac{\sigma}{E} = \frac{14,415}{22 \times 10^6} = .000655 \text{ in/in}$$

The total elongation is

$$\delta = L \epsilon = 3 (.000655) = .001965 \text{ in}$$

The natural frequency

$$\omega_n = \left(\frac{k}{m} \right)^{1/2} = \left(\frac{10 (32.2)(12)}{.001965} \right)^{1/2} = 1405 \text{ rad/sec}$$

$$\omega/\omega_n = .2234$$

$$y_0/a_0 = .0525$$

$$y_0 = 10/3 (.01175) (.05256) = .00203$$

$$AF = .00203/.001965 = 1.03$$

Check buckling strength in compression as per the equation presented in Appendix B, page B-31.

$$P^c = 8010 \text{ lbs.}$$

where,

$$\begin{aligned} E &= 22.0 (10^6) \\ I &= .00133 \text{ in}^4 \\ L &= 3.0 \text{ in} \end{aligned}$$

The critical buckling stress is

$$\sigma_c = \frac{P'}{A} = 61,853 \text{ psi}$$

Since the applied stress was 14,415 psi, the design is safe in buckling.

Calculate stresses in the bellows due to 2 atm. outside pressure. The .008" wall thickness of the bellows was selected based on minimum manufacturing capabilities. The dimensions of the proposed bellows is shown on the sketch below:

$$\begin{aligned} t &= .008 \\ a &= 2.749 \\ b &= .219 \\ N &= 11 \text{ (number of semi-circular corrugations)} \end{aligned}$$

The maximum meridional bending stress, (Ref. 6, page 274, case 23)

$$s_1 = \frac{.955p}{(1 - \nu^2)^{1/3}} \left(\frac{ab}{t^2}\right)^{2/3} = 518 \text{ psi}$$

$$p = 2 \times 14.7 = 29.4 \text{ psi}$$

$$\nu = .32 \text{ (Poisson's ratio for Hastelloy-X)}$$

The maximum circumferential stress is

$$\Delta_2 = .955 p (1 - \nu^2)^{1/6} \left(\frac{ab}{t}\right)^{2/3} = 491 \text{ psi}$$

The allowable stress is

$$\sigma_{\text{allow}} = \frac{\sigma_{\text{yield}}}{\text{SF}} = \frac{38,920}{1.5} = 25,946 \text{ psi}$$

$$\sigma_{\text{yield}} = 38,920 \text{ psi for Hastelloy-X at } 1325^\circ\text{F}$$

Heat leak through the bellows and the six compression rods

$$q \text{ (bellows)} = k A \frac{T_2 - T_1}{L_o} = 6.94 \text{ watts}$$

where,

$$k = 121.8 \text{ Btu-in/ft}^2\text{Hr}^\circ\text{F} \text{ (From Figure A-3)}$$

$$L = N (\text{OD} - \text{ID}) = 11 (5.937 - 4.900) = 11.4 \text{ in} \\ \text{(approximate bellows length)}$$

$$A = \pi D_m t = \pi (5.937 + 4.9)/2 (.008) = .1358 \text{ in}^2 \\ \text{where } D_m = \text{mean bellows diameter}$$

$$T_2 - T_1 = 1325 - 150 = 1175^\circ\text{F}$$

$$L_c = 1/2 L \text{ (The radiation heat transfer here is approximately} \\ \text{calculated by using only half of the bellows length for} \\ \text{heat conduction)}$$

The heat leak through the six compression rods,

$$l = 6 k A \frac{\Delta T}{L} = 25.52 \frac{\text{Btu}}{\text{Hr}} = 7.48 \text{ watts}$$

where,

$$k = 1.008 \text{ Btu/Ft.Hr}^\circ\text{F}$$

$$A = .1295/144 = .00089 \text{ Ft}^2$$

$$T = 1175^\circ\text{F}$$

$$l = 3/12 \text{ in} = .25 \text{ ft.}$$

The total heat leak through the bellows and the six compression rods is
14.4 watts

PART V

BELLOWS WITH SIX TENSION ROD SUPPORTS

The method of calculation for this system is the same as for the bellows with six compression rod support system.

The angle α is calculated according to the method presented in Appendix B, page B-26.

The thermal expansion coefficient for the tension rods (Hastelloy-X) is $\alpha_m = 8.76 \cdot 10^{-6}$ in/in°F (Reference 1).

$$a = 1 + \alpha_m \Delta T = 1 + (8.76) \cdot 10^{-6} \cdot 812^\circ = 1.00711 \text{ in.}$$

The linear thermal expansion of annealed U-8 moly shield is $\alpha = 10.85 \cdot 10^{-6}$ in/in°F as calculated in Part III of this report.

The axial expansion of the shield is

$$b = 10.85 \cdot 10^{-6} [1.8 + 2.6(.866)(1360 - 70)] = .0567 \text{ in.}$$

The radial expansion of the shield is

$$c = 10.85 \cdot 10^{-6} (3.125)(1360 - 70) = .0437 \text{ in.}$$

With the above values of a, b and c and $C_1 = 2.16$, $C_2 = C_1 - C = 2.117$, the distance X is calculated.

$$\begin{aligned} X &= \frac{2(-.0567) \pm [4(.0567)^2 - 4(1-1.00711^2)(.0567^2 + 2.117^2 - (1.00711^2)(2.16^2))]^{\frac{1}{2}}}{2(1 - 1.00711^2)} \\ &= \frac{-.113 \pm (-.00127)^{\frac{1}{2}}}{-.0285} \end{aligned}$$

The minus sign under the square root indicates that no real solution exists. Physically this means that the expansion of the tension rods is large compared to the expansion of the shield and therefore it is not possible to solve for an angle between the tension rods and the shield and the tension rods will not be able to support the shield both in the hot and cold conditions.

If the tension rods were installed at 45° to the axial centerline, the stress in the rods due to thermal expansion is calculated as follows: The thermal expansion of the Hastelloy-X rod is

$$\Delta l_1 = \alpha_m \Delta T = 8.76 \cdot 10^{-6} \cdot 812(3) = .0213 \text{ in.}$$

The thermal expansion of the zirconium compression rod installed at 45° is

$$\Delta l_2 = \alpha_m \Delta T = .00325(3) = .0097 \text{ in.}$$

It was shown for the bellows with six compression rod support system that if the compression rods were installed at 45° the rods are not stressed due to thermal expansion. It follows that if the tension rods are installed at 45° they will be compressed by the difference of the thermal expansion between the tension and compression rods.

$$\epsilon = \Delta l_1 - \Delta l_2 = .0213 - .0097 = .011 \text{ in.}$$

The compressive stress in the rod due to the .011 in. compression

$$\tau = E\epsilon = E \frac{\sigma}{L} = 24.8 \cdot 10^6 \frac{.011}{3} = 90,768 \text{ psi}$$

Since the yield strength of Hastelloy-X is 38,920 psi at 1325°F, designing the rod for 1.5 safety factor based on the yield strength is not possible. As a result of the above findings, this support system is not suitable for this application.

PART VI

BELLOWS WITH COMPRESSED MULTI-LAYER INSULATION SUPPORT

The static loading on the compressed insulation is calculated at 3g and 20g loading. In the vertical position assuming that compressed insulation is used under the entire shield, the compressive loading is,

$$P = \frac{N_g W}{A} = \begin{array}{l} 13.6 \text{ psi at 3g loading} \\ 90.9 \text{ psi at 20g loading} \end{array}$$

$$A = \frac{8.6^2}{4} = 58.05 \text{ in}^2$$

$$W = 264 \text{ lbs.}$$

$$N_g = \text{number of g loading}$$

In the horizontal position assuming that compressed insulation is used under the full cylindrical surface of the shield the compressive loading is,

$$P = \frac{N_g W}{A} = \begin{array}{l} 11.0 \text{ psi at 3g loading} \\ 73.9 \text{ psi at 20g loading} \end{array}$$

$$A = DL = 8.6 (8.3) = 71.38 \text{ in}^2$$

In the vertical upside-down position assuming that the compressed insulation is used around the bellows under the entire shield,

$$P = \frac{N_g W}{A} = \begin{array}{l} 26 \text{ psi at 3g loading} \\ 173 \text{ psi at 20g loading} \end{array}$$

$$A = \frac{8.6^2 - 5.937^2}{4} = 30.38 \text{ in}^2$$

$$W = 264 \text{ lbs.}$$

$$N_g = \text{number of g loading}$$

Under the static loading conditions, the maximum allowable compressive load is about 150 psi. The highest compressive load, 173 psi, was calculated in the vertical upside-down position. To limit the compressive load to 150 psi, the maximum allowable g-loading under static loading conditions could not exceed 17g. During the 45-minute vibration tests under the high compressive loading, it is not known how well the insulation will remain intact as no dynamic experimental data are available. For this reason this support system is not recommended for this application.

PART VII

ALLOWABLE "G" LOADING ON HTVIS SYSTEM WHEN SHIELD IS RIGIDLY SUPPORTED DURING SHIPPING FOR VARIOUS SUPPORT SYSTEMS

A. Neck Tube Support System

If the shield is supported through a rod inside the neck tube during shipping, the neck tube will only have to support the enclosure instead of the shield as shown on the sketch below.

$$W = \text{weight of enclosure} = 41.4 \text{ lb.}$$

The method of calculation for this support system is the same as presented in Part I of this report. In the equations for designing the neck tube, the weight of the shield is replaced by the weight of the enclosure. The center of gravity of the enclosure was calculated to be 7.22 inches away from the upper head. The center of gravity of the shield was 7.29 inches away from the upper head (see calculation of center of gravity, Appendix C, page C-5). Since the center of gravity of the enclosure is at essentially the same location as the center of gravity at the shield, the distances l_1 and l_2 are not changed. The safety factors for the neck tube for buckling in combined bending and internal pressure and in combined transverse shear and internal pressure were calculated for different neck tube wall thicknesses and increased dynamic loadings. Graphical presentation of the results are shown on Figures A-6 and A-7.

Calculate allowable g-loading during shipping for the 5g neck tube design.

The deflection of the enclosure at the center of gravity is

$$\sigma_2 = \frac{W(\text{enclosure})}{W(\text{shield})} \sigma_2(\text{shield}) = \frac{41.4}{280} (.71) \times 10^{-3}$$

$$= .1049 \times 10^{-3} \text{ inches}$$

$$i_n = \left[\frac{5 \times 32.2 \times 12}{001049} \right]^{1/2} = 1362$$

$$i_n / a_n = .2305$$

$$y_o / a_o = .05612$$

$$y_o = 5/2 (.01175)(.05612) = .00109$$

$$AF = .00109 / .001049 = 1.048 \sim 1.05$$

The required safety factor is

$$SF = 1.05 \times 1.5 = 1.57$$

The neck tube wall thicknesses for the 5g neck tube design from Table IV, text,

$$\text{Top} = .033 \text{ in.}$$

$$\text{Bottom} = .030 \text{ in.}$$

To determine how many g-loadings the 5g neck tube design will carry during shipping, enter with .033 wall thickness into Figure A-6 (the cold end is designed for buckling in bending and internal pressure) and at 1.57 safety factor read off 50g allowable g-loading. To determine the allowable g-loading at the hot end of the neck tube in transverse shear, enter with .030 wall thickness into Figure A-7 and at 1.57 safety factor read off 50g allowable g-loading.

B. Neck Tube With Three Tension Rod Supports

The method of calculation for this system support was presented in Appendix B, page B-22. A schematic diagram of the support system when the neck tube and three tension rods are supporting the enclosure is shown on the sketch below.

$$\begin{aligned}
 W &= 41.4 \text{ lb.} \\
 l_1 &= 2.5 \\
 l_{in} &= 2.78 \\
 l_3 &= 4.16 - 1.6 = 2.56
 \end{aligned}$$

Rod supporting shield
during shipping

Determine the allowable g-loading during shipping for the 5g design case. From Table III

Top neck tube thickness = .0095 in.

Bottom " " " = .0095 in.

The force in the tension rod was calculated to be $P = 890.3 \text{ lb.}$

The allowable g-loading during shipping

$$N_g = (5g) \frac{269*}{41.4} \frac{890.3}{809.1**} = 35g$$

C. Neck Tube With Three Compression Rod Supports

The procedure to calculate the allowable g-loading for this support system during shipping is the same as presented in Part VIIb of this report.

For the 10g design case the force in the compression rod is 1865.9 lb. The allowable g-loading during shipping is

$$N_g = (10g) \frac{268}{41.4} \frac{1599}{1865.9} = 55.4 \text{ g}$$

D. Neck Tube With Six Compression Rod Supports

The allowable g-loading during shipping for this support system was obtained by multiplying the design g-loading by the ratio of shield weight to enclosure weight.

* Weight of shield and fuel capsule for the 5g design case.

**Force in the tensile rods calculated for the 5g design case.

FIGURE A-2

THE EFFECT OF DIMENSIONLESS LENGTH
ON THERMAL CONDUCTIVITY IN A ROD (ASTELLOY X)
WITH ONE DIMENSIONAL HEAT CONDUCTION

FIGURE A-3

MAXIMUM SHEAR STRESS IN THE NECK TUBE
FOR 1000 LBS. SHEAR LOAD
FOR DIFFERENT NECK TUBE WALL THICKNESSES

SAFETY FACTORS FOR TAPERED NECK TUBES
FOR BUCKLING IN COMBINED BENDING
AND INTERNAL PRESSURE WHEN SHIELD
IS RIGIDLY SUPPORTED DURING SHIPPING
(WEIGHT OF ENCLOSURE 41.5 LBS.)

Not End

A-29

SAFETY FACTOR

20

15

10

5

.010 .015 .020 .025 .030 .035 .040 .045 .050 .060

NECK TUBE WALL, THICKNESS, INCHES

30

50

100

150

200

300

400

500

600

700

700 A-

SAFETY FACTORS FOR TAPERED NECK TUBES FOR BUCKLING
IN COMBINED TRANSVERSE SHEAR AND INTERNAL PRESSURE
WHEN SHIELD IS RIGIDLY SUPPORTED DURING SHIPPING
(WEIGHT OF ENCLOSURE 41.5 LBS.)

APPENDIX B

This appendix originally appeared as a portion of Appendix 1 of Report No. 3. It is included in this topical report to provide reference for Appendix A.

NECK TUBE STRESS ANALYSIS

1. Vertical position (neck tube open end up)

A.) Combined tension and internal pressure

The weight on the neck tube acting downward under 3 g accelerations

$$W = 3 \times 264 = 792 \text{ lbs.}$$

The stresses at the hot end of the tube are as follows:

The tensile stress due to W

$$\sigma_t = \frac{W}{A} = \frac{4W}{\pi(D_o^2 - D_i^2)} = \frac{4 \times 792}{\pi(4.922^2 - 4.900^2)} = 4669 \text{ psi}$$

The circumferential or the hoop stress due to 2 atmospheres inside pressure

$$\sigma_1 = \frac{PR}{t} = \frac{2 \times 14.7 \times 2.461}{.011} = 6577 \text{ psi}$$

The longitudinal stress

$$\sigma_2 = \frac{PR}{2t} = 3288 \text{ psi}$$

The maximum stress in the longitudinal direction

$$\sigma_{\max} = \sigma_t + \sigma_2 = 4669 + 3288 = 7957 \text{ psi}$$

The safety factor based on the maximum stress theory

$$SF = \frac{\sigma_{\text{yield}}}{\sigma_{\max}} = \frac{38,920}{7957} = 4.8$$

$$\sigma_{\text{yield}} = 38,920 \text{ psi for Hastelloy X at } 1325^\circ\text{F. Ref. (1)}$$

At the cold end of the tube the safety factor will be larger because both the cross sectional area and the yield strength is increased. For 1 atmosphere inside pressure the maximum tensile strength will be decreased which will also result in a higher safety factor.

2. Up-side down vertical position

A.) Combined compression and internal pressure

At the hot end of the neck tube -

The compressive stress due to W

$$\sigma_c = \frac{W}{A} = \frac{792}{.1696} = 4669 \text{ psi}$$

The longitudinal stress due to 1 atmosphere internal pressure

$$\sigma_2 = \frac{P R}{2 t} = \frac{14.7 \times 2.461}{2 \times .011} = 1644 \text{ psi}$$

The net compressive stress in the longitudinal direction

$$\sigma_{\max} = \sigma_c - \sigma_2 = 4669 - 1644 = 3025 \text{ psi}$$

The safety factor in compression based on the yield strength

$$SF = \frac{\sigma_{\text{yield}}}{\sigma_{\text{MAX}}} = \frac{38,920}{3025} = 12.8$$

B.) Buckling in combined compression and internal pressure.

At the hot end of the neck tube -

Using Figure 4a in Reference (2)

For 1 atmosphere pressure

$$\frac{P R}{E t} = \frac{14.7 \times 2.461}{21.2 \times 10^6 \times .011} = .0221$$

$$E = 21.2 \times 10^6 \text{ psi at } 1325^\circ\text{F Ref. (1)}$$

$$\frac{L}{R} = \frac{2.497}{2.461} = 1.01$$

$$\frac{R}{t} = \frac{2.461}{.011} = 237.2$$

The straight line for $R/t = 237$ shown on Fig. 4a was determined by extrapolating $(\sigma_{ccR}/E_r) (R/t)$ for values of R/t less than 500.

Knowing $\frac{P R}{E t} = .007$ and having the design curve for $R/t = 237$, the value of $(\sigma_{CCR}/E_r) (R/t)$ is now obtained from Fig. 4a.

$$\frac{\sigma_{CCR}}{E_r} \frac{R}{t} = .27$$

$$\sigma_{CCR} = (.27) \frac{t}{R} E_r = .00113 E_r$$

To calculate σ_{CCR} we need to know the reduced modulus of elasticity corresponding to the maximum compressive stress. Using a trial-and-error procedure, first assume a modulus of elasticity and calculate a maximum compressive stress. If the modulus of elasticity at the calculated maximum compressive stress is different than the assumed value, the calculation is repeated with a corrected modulus of elasticity.

Using the stress-strain curve for Hastelloy X given in Ref. (1), stress - versus modulus of elasticity was plotted and is shown on Fig. B-1. Results of the trial-and-error procedure gives

$E \times 10^6$	$\sigma_{CCR} - \sigma_2$	$E_r \times 10^6$ (from Fig. 1)
23.5	$26.555 - 1644 = 24.910$	24.7
23.6	$26.668 - 1644 = 25.020$	24.2
23.7	$26.781 - 1644 = 25.137$	23.8

$$\sigma_{CCR} = (.27) \frac{.011}{2.461} 23.7 \times 10^6 = 26.781 \text{ psi}$$

$$\text{Safety factor} = \frac{26.781}{3025} = 8.8$$

3. Horizontal Position

$$3W = 3 \times 264 = 792 \text{ lbs.}$$

$$L = 7''$$

(+) tension

(-) compression

$$t = .011$$

A.) Combined bending and internal pressure

The combined stress

$$\sigma_c = \pm \frac{M_c}{I} + \frac{P R}{2t}$$

The moment at the cold end of the neck tube

$$M = 3 W L = 792 \times 7 = 5544 \text{ in. lbs.}$$

The section modulus

$$Z = \frac{I}{c} = \pi R^2 t = \pi (2.455)^2 (.011) = .2082 \text{ in.}^3$$

The bending stress

$$\sigma_B = \pm \frac{M_c}{I} = \frac{M}{Z} = \frac{5544}{.2082} = 26,621 \text{ psi}$$

The maximum tensile stress for 2 atmospheres internal pressure.

$$\begin{aligned}\sigma_t &= + \frac{M_c}{I} + \frac{P R}{2t} = + 26,621 + \frac{2 \times 14.7 \times 2.461}{2 (.011)} = \\ &= 26,621 + 3238 = 29,909 \text{ psi}\end{aligned}$$

The safety factor based on the maximum stress theory.

$$SF = \frac{\text{Yield Stress}}{\text{Maximum Applied Stress}} = \frac{50,500}{29,909} = 1.67$$

$$\sigma_{\text{yield}} = 50,500 \text{ psi for Hastelloy X at } 150^\circ\text{F Ref. (1)}$$

B.) The shear stresses

$$\tau = \frac{3 W \sin \alpha}{\pi R t} = \frac{792}{\pi 2.455 (.011)} = 9317 \text{ psi (shear)}$$

Taking the yield strength of Hastelloy X in shear as

$$\tau_{\text{yield}} = 1/2 \sigma_{\text{yield}} = 1/2 50,500 = 25,250 \text{ psi}$$

$$SF = \frac{25,250}{9317} = 2.7$$

C.) Buckling in combined bending and internal pressure.

Determine critical bending stress using Figure 8 Ref. (3)

P = 1 atmosphere internal pressure

t = .011"

E = 28.3×10^6 psi cold end of neck tube

E_r (see Figure 1)

$$\frac{P}{E} \left(\frac{r}{t} \right)^2 = \frac{14.7}{28.3 \times 10^6} \left(\frac{2.461}{.011} \right)^2 = .0259$$

$$\frac{\sigma_{cR}}{E_r} \frac{r}{t} = .34$$

$$\sigma_{cR} = (.34) \frac{t}{r} E_r = (.34) \frac{.011}{2.461} E_r = .001519 E_r$$

$$E \times 10^6 \quad \sigma_{cR} = \frac{P R}{2 t} E_r$$

$$19.0 \quad 28,870 - 1654 = 27,216 \quad 19.1$$

$$SF = \frac{\sigma_{BcR}}{M/Z} = \frac{28,870}{26,621} = 1.08$$

Critical buckling stresses and safety factors were calculated for various wall thicknesses ranging from .011 to .018". Results are shown on Figure B-2.

Determine critical bending stress using Figure 5. Ref. (2)

P = 1 atmosphere internal pressure

t = .011"

E = 28.3×10^6 psi (cold end of neck tube)

E_r see Figure 1

The straight line for $R/t = 237$ shown on Figure 5 Ref. (2) was obtained by extrapolation.

$$\frac{P R}{E t} = \frac{14.7 \times 2.461}{28.3 \times .011} = .0166$$

From Figure 5 Ref. (2)

$$\frac{\sigma_{BcR}}{E_r} \frac{R}{t} = .26$$

$$\sigma_{BcR} = (.26) \frac{t}{R} E_r = .26 \frac{.011}{2.461} E_r = .001117 E_r$$

By trial-and-error procedure,

$E \times 10^6$		$\sigma_{cR} = \frac{P R}{2 t}$		$E_r \times 10^6$
24.1		$26,926 - 1644 = 25,282$		24
		$\sigma_{BcR} = 26,926$		

$$SF = \frac{\sigma_{BcR} + \frac{P R}{2 t}}{M/Z} = \frac{26,926 + 1644}{26,621} = 1.07$$

Critical bending stresses and safety factors were calculated for various wall thicknesses ranging from .011 to .018". Results are shown on Figure B-2.

D.) Buckling in combined transverse shear and internal pressure.

Determine the critical shear stress by using Figure 6a Ref. (2)

$$\frac{PR}{Et} = \frac{14.7 \cdot 2.461}{21.4 \times 10^6 \cdot .011} = 153 \cdot 10^{-6} = .153 \times 10^{-3}$$

$$E = 21.4 \times 10^6 \text{ psi for Hastelloy X at } 1325^\circ\text{F. See Figure B-10.}$$

Since $PR/Et < 10^{-3}$, the internal pressure does not affect the torsion buckling stress. From Figure 6a (Ref. 2).

$$\frac{\tau_{cR}}{E_r} \times 10^3 = .56$$

The same result can be obtained from Figure 2 Ref. (2) for the buckling stress in pure torsion.

$$\tau_{cR} = (.56) 10^{-3} E_r = .00056 \cdot 21.4 \times 10^6 = 11,934 \text{ psi}$$

The shear buckling stress

$$\tau_{ScR} = K \tau_{cR} (\text{torsion}) = 1.25 (11,934) = 14,980 \text{ psi}$$

The applied shear stress,

$$\tau_{\text{applied}} = \frac{3 W \sin \alpha}{\pi R t} = \frac{792 (1)}{\pi 2.455 (.011)} = 9317 \text{ psi}$$

Safety factor in combined shear and internal pressure,

$$SF = \frac{14,980}{9317} = 1.6$$

Safety factors were calculated for various wall thicknesses ranging from .011 to .016. Results are shown on Figure B-3.

VIBRATION AND SHOCK ANALYSIS

A.) Vibration Analysis

The vibration model for the cantilever beam

The equation for resonance for the case of forced vibration without damping from Ref. (4) equation 2.26 page 45

$$\frac{y_o}{a_o} = \frac{\left(\frac{\omega}{\omega_n}\right)^2}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

y_o is the relative motion between the mass and the fixed end of the spring.

a_o is the amplitude of motion at the fixed end of the spring.

frequency of motion at fixed end.

ω_n natural frequency of spring mass system.

The maximum vibration frequency is 50 cps. at 3 g peak acceleration as specified in the dynamic vibration test.

The amplitude of motion at 50 cps and 3 g peak acceleration is calculated from the relationship.

$$X = a_o \sin \omega t$$

$$\dot{X} = a_o \omega \cos \omega t$$

$$\ddot{X} = -a_o \omega^2 \sin \omega t$$

$$3 g = a_o \omega^2$$

$$a_o = \frac{3g}{\omega^2} = \frac{3 g}{(2 \pi f)^2} = \frac{3 \times 32.2 \times 12}{4 \pi^2 50^2} = .01175 \text{ in.}$$

The natural frequency of the system

$$\omega_n = \sqrt{\frac{k}{M}}$$

The spring constant for the system is taken as the force acting at the free end of the cantilever divided by the total deflection at the end of the cantilever. The total deflection is due to the force and moment acting on the beam plus the rotation of the cantilever (neck tube) due to the deflection at the flange.

Deflection due to the load 3 W

$$\delta_1 = \frac{1}{3} \frac{P l^3}{E_{Av} I_{av}} = \frac{792 (2.497)^3}{3 (24.8) 10^6 (.628)} = .000263 \text{ in.}$$

$$I_{av} = \frac{I_{\text{cold end}} + I_{\text{hot end}}}{2}$$

For .0115" wall thickness at the hot end

$$I_{\text{hot end}} = \frac{\pi}{64} (D^4 - d^4) = \frac{\pi}{64} (4.923^4 - 4.900^4) = .5347 \text{ in.}^4$$

For .0155 wall thickness at the cold end

$$I_{\text{cold end}} = \frac{\pi}{64} (D^4 - d^4) = \frac{\pi}{64} (4.931^4 - 4.900^4) = .7226 \text{ in.}^4$$

$$I_{\text{av}} = \frac{.5347 + .7226}{2} = .6286 \text{ in.}^4$$

$$E_{\text{av}} = \frac{E_{\text{cold end}} + E_{\text{hot end}}}{2} = \frac{28.3 + 21.4}{2} \times 10^6 = 24.8 \times 10^6$$

The deflection due to the moment

$$\delta_2 = \frac{1}{2} \frac{M \ell^2}{E_{\text{av}} I_{\text{av}}} = \frac{1}{2} \frac{3572 (2.497)^2}{24.8 \times 10^6 (.6286)} = .000714 \text{ in.}$$

$$M = 3 W \ell = 792 (4.511) = 3572 \text{ in. lbs.}$$

The rotation of the neck tube due to the deflection of the flange is $\delta_3 = .00008 \text{ in.}$ using .375" thick flange.

The total deflection of the beam is

$$\delta_{\text{TOT}} = \delta_1 + \delta_2 + \delta_3 = .000263 + .000714 + .00008 = .001057 \text{ in.}$$

The natural frequency of the system

$$\omega_n = \left[\frac{3 W / \delta_{\text{TOT}}}{M} \right]^{1/2} = \left[\frac{792 \times 32.2 \times 12}{(.001057)(264)} \right]^{1/2} = 1047 \text{ rad./sec.}$$

$$\omega = 2 \pi f = 2 \pi 50 = 314 \text{ rad./sec.}$$

$$\frac{\omega}{\omega_n} = \frac{314}{1047} = .2999$$

$$\frac{y_o}{a_o} = \frac{\left(\frac{\omega}{\omega_n}\right)^2}{1 - \left(\frac{\omega}{\omega_n}\right)^2} = \frac{(.2999)^2}{1 - (.2999)^2} = .09882$$

$$y_o = a_o \left(\frac{y_o}{a_o}\right) = (.01175)(.09882) = .00116 \text{ in.}$$

The force acting on the end of the neck tube due to the vibration.

$$P = k y_o = \frac{792}{.001057} (.00116) = 869.1 \text{ lbs.}$$

The equivalent acceleration at the end of the neck tube is

$$a = \frac{P}{W} = \frac{869.1}{264} = 3.292 \text{ g} \quad \sim 3.3 \text{ g}$$

$$\text{Amplification factor} = \frac{869.1}{792} = 1.097 \quad \sim 1.1 \text{ g}$$

The Model A (heavy wall)

$$\delta_1 = \frac{P \ell^3}{3 E_{av} I_{av}} = \frac{792 (2.497)^3}{3 \times 24.8 \times 10^6 (.7226)} = .000229 \text{ in.}$$

For .0135 wall thickness at the hot end

$$I_{\text{hot end}} = \frac{\pi}{64} (D^4 - d^4) = \frac{\pi}{64} (4.927^4 - 4.900^4) = .6285 \text{ in.}^4$$

For .0175 wall thickness at the cold end.

$$I_{\text{cold end}} = \frac{\pi}{64} (D^4 - d^4) = \frac{\pi}{64} (4.935^4 - 4.900^4) = .8167 \text{ in.}^4$$

$$I_{av} = \frac{.6285 + .8167}{2} = .7226$$

$$\delta_2 = \frac{M \ell^2}{2 E_{av} I_{av}} = \frac{3572 (2.497)^2}{2(24.8) 10^6 (.7226)} = .000621 \text{ in.}$$

$$\delta_{TOT} = .0000229 + .000621 + .00008 = .00093 \text{ in.}$$

The natural frequency of the system

$$\omega_n = \sqrt{\frac{3W/\delta_{TOT}}{M}} = \left[\frac{792 \times 32.2 \times 12}{.00093 \times 264} \right]^{1/2} = 1116 \text{ rad./sec.}$$

$$\frac{\omega}{\omega_n} = \frac{314}{1116} = .281$$

$$\frac{y_o}{a_o} = \frac{\left(\frac{\omega}{\omega_n}\right)^2}{1 - \left(\frac{\omega}{\omega_n}\right)^2} = \frac{(.281)^2}{1 - (.281)^2} = .0859$$

$$y_o = a_o \left(\frac{y_o}{a_o}\right) = (.01175)(.0859) = .00101 \text{ in.}$$

$$P = k y_o = \frac{792}{.00093} (.00101) = 860.1 \text{ lbs.}$$

$$a = \frac{P}{W} = \frac{860.1}{264} = 3.258 \sim 3.26 \text{ g}$$

$$\text{Amplification factor} = \frac{860.1}{792} = 1.086 \sim 1.09$$

B.) Shock Analysis

The maximum shock level which the unit has to withstand is the following:
(See dynamic shock test specifications.)

t_d = nominal duration of
shock = 6 ms

Determine shock response by Figure 5.9 page 117 Ref. (5)

Model A Neck Tube

$$f_n = \frac{\omega_n}{2\pi} = \frac{1047}{2\pi} = 166.72 \text{ cps}$$

$$2 \times f_n \times t_d = 2 (166.72) 6 \times 10^{-3} = 2.00 \text{ cycles}$$

The acceleration from Fig. 5.9, Ref. (5).

$$a = 1.05 \times 3 g = 3.15 g$$

Model A (heavy wall)

$$f_n = \frac{\omega_n}{2\pi} = \frac{1116}{2\pi} = 177.70 \text{ cps}$$

$$2 \times f_n \times t_d = 2 \times 177.70 \times 6 \times 10^{-3} = 2.13 \text{ cycles}$$

From Figure 5.9 Ref. (5)

$$a = (.95) 3 g = 2.85 g$$

In order to reduce the amplification factor to nearly 1.0, the required time duration is:

$$t_d = \frac{5}{2 \times 166.72 \times 10^{-3}} = 14.9 \sim 15 \text{ ms}$$

Model A heavy wall

$$t_d = \frac{5}{2 \times 177.70 \times 10^{-3}} = 14.0 \text{ ms}$$

HEAT LEAK CALCULATIONS THROUGH STRAIGHT AND TAPERED NECK TUBES

1.) Straight Neck Tube

$$\begin{aligned} L &= 2.497 \text{ in} \\ d_i &= 4.900 \text{ in.} \\ d_o &= 4.922 \text{ in.} \\ t &= .011 \\ A &= .1696 \text{ in.}^2 = \frac{.1696}{144} \text{ Ft.}^2 \end{aligned}$$

$$q = -k(T) A \frac{dT}{dX}$$

k versus temperature plot is shown on Figure 4-B for Hastelloy X,

The equation of the straight line on Figure 4-B.

$$y = \frac{y_2 - y_1}{x_2 - x_1} (x - x_1) + y_1$$

$$\left. \begin{aligned} x_1 &= 150^\circ\text{F} \\ y_1 &= k_1 = 71 \text{ Btu-in./ft.}^2\text{-hr.-}^\circ\text{F} \\ x_2 &= 1325^\circ\text{F} \\ y_2 &= k_2 = 161 \text{ Btu-in./ft.}^2\text{-hr.-}^\circ\text{F} \end{aligned} \right\} \text{ cold end}$$

$$y = k = \frac{161-71}{1325-150} (T - 150) + 71 = .07659 T + 59.511$$

$$k = a T + b \quad \text{where:} \quad \begin{aligned} a &= .07659 \\ b &= 59.511 \\ A &= \frac{\pi}{4} \end{aligned}$$

$$q = -(a T + b) A \frac{dT}{dX}$$

$$\int_0^L q \, dX = \int_{150}^{1325} -(a T + b) A \, dT$$

$$q = - \frac{\left[\frac{a T^2}{2} + b T \right]_{150}^{1325} A}{L}$$

$$\begin{aligned} q &= - \left[\frac{(.076595) 1325^2}{2} + (59.511)(1325) - \frac{(.076595) 150^2}{2} - (59.511)(150) \right] \frac{.1696}{2.497 \times 144} = \\ &= 64.28 \text{ Btu/hr.} \end{aligned}$$

$$q = 64.28 \text{ [Btu/hr.]} \times \frac{1}{3.41} \text{ [watts hr./Btu.]} = 18.83 \text{ watts}$$

2.) Tapered Neck Tube

$$r_i = 2.450$$

$$r_o = 2.461$$

$$r_L = 2.462$$

$$t_{\text{bottom}} = .011$$

$$t_{\text{top}} = .012$$

$$r = aX + b$$

The boundary conditions

$$X = 0 \quad r = r_o = 2.461 = b$$

$$X = L \quad r = r_L = 2.462 = aL + 2.461$$

$$a = \frac{2.462 - 2.461}{2.497} = \frac{.001}{2.497} = .0004004857$$

$$A(X) = \left[(aX + b)^2 - r_i^2 \right] \frac{\pi}{144} = \frac{\pi}{144} \left[(aX + 2.461)^2 - 2.450^2 \right]$$

$$g = -k(T) A(X) \frac{dT}{dX}$$

$$\int \frac{d(X)}{A(X)} = \frac{144}{\pi} \int \frac{dX}{(aX + 2.461 + 2.450)(aX + 2.461 - 2.450)} \times$$

From Table of Integrals

$$\int \frac{dX}{(aX + b)(cX + d)} = \frac{1}{b, c - ad} \ln \frac{cX + d}{aX + b}$$

$$a = c = .0004004857$$

$$b, = 2.461 + 2.450 = 4.911$$

$$d = 2.461 - 2.450 = .011$$

$$\int \frac{dX}{A(X)} = \frac{144}{\pi} \frac{1}{4.911 (.0004004857 - .0004004857 (.011))} \times$$

$$\times \left[\ln \frac{.001 + .011}{.001 + 4.911} - \ln \frac{.011}{4.911} \right] = 2026.13$$

$$q = - \int \frac{k(T) dT}{\frac{dX}{AX}}$$

Using the numerical value of $\int k(T) dT$ from the calculations for a straight neck tube.

$$q = \frac{136295.433}{2026.1371 \times 3.413} = 19.7 \text{ watts}$$

Heat leaks through straight and tapered neck tubes were calculated for various top and bottom wall thicknesses and results are shown on Figure B-5, Appendix two.

Heat leak for Model A neck tube

Bottom thickness .0115"
Top thickness .0155"

$$q = 23.4 \text{ watts}$$

Heat leak for Model A (heavy wall) neck tube

Bottom thickness .0135"
Top thickness .0175"

$$q = 27.4 \text{ watts}$$

RUPTURE LIFE OF NECK TUBE

Stress rupture life for Hastelloy X is shown on Figure B-6.

The bending moment at the hot end of the neck tube under 1 g acceleration is

$$M = W \cdot L = 264 \times 4.511 = 1190 \text{ in.-lbs.}$$

From Figures B-7 and B-8 it was estimated that the total oxidation of the Hastelloy X neck tube is .001" over the five year life. The average stress in the neck tube during the five year life was calculated for a reduced wall thickness due to oxidation. The average wall thickness used to determine the stress in the neck tube was .0005" less than the original wall thickness.

The average thickness of the Model A (light wall) neck tube at the hot end allowing .0005" for oxidation.

$$t = .0115" - .0005" = .011"$$

The maximum tensile stress in the neck tube when the unit is in the horizontal position is the sum of the tensile stresses due to bending and internal pressure.

$$\sigma = \frac{M}{Z} + \frac{P R}{2 t} = \frac{1190}{.2082} + \frac{2 \times 14.7 \times 2.461}{2 \times .011} = 9004 \text{ psi}$$

$$Z = \pi R_{av}^2 t = \pi \left(\frac{2.461 + 2.450}{2} \right)^2 (.011) = .2082 \text{ in.}^3$$

From Figure B-5, the extrapolated rupture stress of Hastelloy X for 5 year life is 9300 psi. Since the rupture stress is larger than the applied stress the design is safe for the required 5 year life.

OPTIMIZATION OF NECK TUBE LENGTH

Top and bottom wall thicknesses for tapered neck tubes were calculated for 1.5 and 1.75 safety factors based on strength. The top wall thickness of the neck tube (cold end) was determined based on buckling in combined bending and internal pressure, and the bottom wall thickness (hot end) was determined based on buckling in transverse shear. A trial and error procedure was used to calculate the wall thicknesses. First, an amplification factor was assumed and the top and bottom wall thicknesses were calculated using the buckling criteria. Knowing the wall thicknesses, the actual amplification factor was calculated. If the actual amplification factor was different than originally assumed, the calculation was repeated with a new amplification factor. Wall thicknesses and corresponding heat leaks were calculated for 1", 2", 3", and 4" added neck tube lengths. Results are summarized in Table I, and Figure B-11 in the Appendix. The optimum length was found at 3" added neck tube length. For this case the calculations were as follows:

Total length of neck tube

$$L = 2.497 + 3 = 5.497$$

The critical buckling strength in combined bending and internal pressure and in shear buckling were calculated for various wall thicknesses using Figures 5, and 6a in Ref. 2. Results are shown on Figures B-2 and B-3. By trial and error solution; assume an amplification factor of 1.65.

For 1.5 safety factor and 1.65 amplification factor, the nominal safety factor is:

$$SF = 1.5 \times 1.65 = 2.47$$

From Figures B-2 and B-3.

Top thickness .0275"

Bottom thickness .017"

The average moment of inertia,

$$I_{av} = 1.042 \text{ in}^4$$

The total deflection is

$$\delta_{tot} = .00445 \text{ in}$$

The natural frequency

$$f_n = 510 \text{ rad/sec}$$

Amplification factor

$$AF = 1.61$$

Allowing 1.5 mil variation on the diameter for manufacturing tolerances

$$\text{Top thickness} = .0275 + .00075" = .02825"$$

$$\text{Bottom thickness} = .017 + .00075" = .01775"$$

The heat leak,

$$q = q^* \frac{L(\text{original})}{L(\text{new})} \frac{A(\text{new})}{A(\text{original})} =$$
$$= 18.83 \frac{2.497}{5.497} \frac{.3555}{.1696} = 17.92 \text{ watts}$$

$$q^* = \text{heat leak for straight neck tube, .011 thick, 2.497" long}$$

The second neck tube (back-up heavy wall design)

Assume 1.55 amplification factor

The nominal safety factor,

$$SF = 1.75 \times 1.55 = 2.71$$

From Figures B-2 and B-3,

$$\text{Top thickness} = .0295$$

$$\text{Bottom thickness} = .0185$$

The average moment of inertia,

$$I_{av} = 1.1254 \text{ in}^4$$

The total deflection

$$\delta_{tot} = .00418$$

$$\omega_n = 526 \text{ rad/sec}$$

$$\frac{y_o}{a_o} = .5535 \quad y_o = .0065 \text{ in}$$

$$AF = \frac{y_o}{\delta_{tot}} = \frac{.0065}{.00418} = 1.55$$

Allowing 1.5 mil on the diameter for manufacturing tolerances

$$\text{Top thickness} = .0295 + .00075 = .03025$$

$$\text{Bottom thickness} = .0185 + .00075 = .01925$$

$$q = 18.83 \frac{2.497}{5.497} \frac{.3827}{.1696} = 19.3 \text{ watts}$$

OPTIMIZATION OF NECK TUBE LENGTH INCLUDING THE EFFECT OF SHEAR DEFLECTION

In the previous analysis shear deflections have been neglected. The fact that shear deflection is not negligible for a short beam like the neck tube will be illustrated here by evaluating the ratio of shear deflection to bending deflection for a cantilever beam.

The shear deflection for a cantilever beam with end load P (Ref. 6, page 120).

$$y_s = F \left(\frac{P \ell}{A G} \right)$$

where $F = 2$ for thin-walled hollow circular section

G modulus of rigidity

A cross sectional area of beam

ℓ length of cantilever beam

The ratio of shear deflection to bending deflection

$$y_B = \frac{P \ell^3}{3EI}$$

$$G = \frac{E}{2(1+\mu)} = \frac{E}{2(1+.32)} = \frac{E}{2(1.32)} = \frac{E}{2.64}$$

$\mu = .32$ for Hastelloy X Ref. (1)

μ = Poisson's Ratio

$$y_s = 2 \frac{P \ell}{AE} \frac{2.64}{1} = 5.28 \frac{P \ell}{2 r \pi t E}$$

$$A = 2 r \pi t$$

$$y_B = \frac{P \ell^3}{3 E \pi r^3 t}$$

$$\frac{y_s}{y_B} = \frac{\frac{5.28 P \ell}{2 \pi r t E}}{\frac{P \ell^3}{3 E \pi r^3 t}} = 7.92 \left(\frac{r}{\ell}\right)^2$$

For the original length neck tube, $\frac{r}{\ell} \approx 1$, the shear deflection is 7.9 times longer than the bending deflection.

By trial and error solution as before, assume 1.85 amplification factor,

$$SF = 1.5 \times 1.85 = 2.77$$

For the original length neck tube ($L = 2.497''$)

From Figures B-2 and B-3,

$$\text{Top thickness} = .0225''$$

$$\text{Bottom thickness} = .019''$$

The average moment of inertia,

$$I_{av} = .9704 \text{ in}^4$$

The average cross sectional area,

$$A_{av} = .3206 \text{ in}^2$$

The total deflection at the center of gravity including shear deflection is given by the formula derived in the Appendix.

$$\delta_2 = W \left[\frac{\ell_1^2 (2 \ell_1 + 3 \ell_2) + 3 \ell_1 \ell_2 (\ell_1 + 2 \ell_2)}{6 E I} + \frac{2 \ell_1 (2.64)}{A E} + \frac{2 (\ell_{1n} + \ell_2)^2}{64.9 \times 10^6} \right]$$

$$\ell_1 = 2.5 \quad \ell_2 = 4.52$$

$$\ell_{1n} = 2.78$$

$$= 792 \left[\frac{2.5^2 [2 (2.5) + 3 (4.52)]}{6 (24.8) (.9704) 10^6} + \frac{3 (2.5) (4.52) (2.5 + 2 (4.52))}{6 (24.8) (.9704) 10^6} + \frac{2 (2.5) (2.64)}{(.3206) (24.8) 10^6} + \frac{2 (2.78 + 4.52)^2}{(64.9) 10^6} \right] =$$

$$= 792 \left[3.51 \times 10^{-6} + 1.66 \times 10^{-6} + 1.64 \times 10^{-6} \right] = .00539 \text{ in}$$

$$u_n = 464 \text{ in/sec}$$

$$u/u_n = .6767$$

$$y_o/a_o = \frac{.4579}{.5420} = .8447$$

$$y_o = .0099$$

$$AF = 1.84$$

Allowing 1.5 mil on the diameter for manufacturing tolerances

$$\text{Top thickness} = .0225 + .00075 = .02325$$

$$\text{Bottom thickness} = .019 + .00075 = .01975$$

$$\text{Average cross sectional area } A_{av} = .3322 \text{ in}^2$$

$$q = q^* \frac{A_{new}}{A_{old}} = 18.83 \frac{.3322}{.1696} = 36.8 \text{ watts (max.)}$$

$$q = q^* \frac{A_{new}}{A_{old}} = 18.83 \frac{.3206}{.1696} = 35.6 \text{ watts (min.)}$$

Neck tube wall thickness and corresponding heat leak calculations were repeated for 1" and 1.5" added neck tube lengths. Graphical presentation of the results are shown on Figure B-12.

Tension Rod Support Calculations

The diameter of the tension rod support was determined by using a trial-and-error procedure. First the rod diameter was assumed and the force in the rod was calculated. Then the amplification factor and the stresses in the tension rod and in the necktube were calculated. If the stress in the tension rod was higher than the yield strength of the material divided by 1.5 safety factor, the calculations were repeated with a larger rod diameter. The stresses in the necktube were checked for buckling in combined bending and internal pressure and in transverse shear.

The equation for calculating the force in the tension support is derived in Appendix C.

The force in the tension support,

$$P = \frac{W}{E_r A_r} \frac{\left[\frac{l_1^2(2l_1 + 3l_2 + 3l_3) + 3l_1 l_2(l_1 + 2l_2 + 2l_3)}{EI} \right] + \frac{2l_1(2.64)}{AE} + \frac{2(l_{in} + l_2)(l_{in} + l_2 + l_3)}{64.9 \times 10^6}}{\frac{l_r}{E_r A_r} + \frac{l_1^2(2l_1 + 3l_2 + 3l_3) + 3l_1(l_2 + l_3)(l_1 + 2l_2 + 2l_3)}{EI} + \frac{2l_1(2.64)}{AE} + \frac{2(l_{in} + l_2 + l_3)^2}{64.9 \times 10^6}}$$

For a straight necktube of .0095" wall thickness and the original length $l_1 = 2.5$ in

$$\begin{array}{ll} l_1 = 2.5 \text{ in} & A = .1464 \text{ in}^3 \\ l_{in} = 2.78 \text{ in} & l_r = 3.5 \text{ in} \\ l_2 = 4.52 \text{ in} & E_r = E = 24.8 \times 10^6 \text{ psi} \\ l_3 = 4.16 \text{ in} & d_r = 11/64 \text{ in} \\ I = .4412 \text{ in}^4 & A_r = .02318 \text{ in}^2 \end{array}$$

$$P = \frac{792(13.21 + 3.63 + 2.57)10^{-6}}{(6.08 + 22.64 + 3.63 + 4.04)10^{-6}} = 422.44 \text{ lbs.}$$

The stress in the wire,

$$\sigma_r = \frac{P}{A} = \frac{422.44}{.02318} = 18,224 \text{ psi}$$

The force and the stress in the rod were calculated for different rod diameters and results were plotted on Figure B-13.

The deflection at the center of gravity by the formula derived in the Appendix C,

$$\delta_{D_2} = W \left[\frac{l_1^2(2l_1 + 3l_2) + 3l_1l_2(l_1 + 2l_2)}{6EI} + \frac{2P(2.64) + 2(l_{in} + l_2)^2}{AE} \right] \\ - P \left[\frac{l_1^2(2l_1 + 3l_2) + 3l_1(l_2 + l_3)(l_1 + 2l_2)}{6EI} + \frac{2P(2.64) + 2(l_{in} + l_2)(l_{in} + l_2 + l_3)}{64.9 \times 10^6} \right]$$

$$\delta_{D_2} = \left[792 \quad 7.72 + 3.63 + 1.64 \right] 10^{-6} - 422.44 (13.21 + 3.63 + 2.57) 10^{-6} =$$

$$.010288 - .008199 = .002088 \text{ in}$$

The natural frequency,

$$\omega_n = 746 \text{ rad/sec}$$

$$\omega/\omega_n = .4209$$

$$y_0/a_0 = \frac{.1771}{.8228} = .2152$$

$$y_0 = .00252$$

$$Af = \frac{.00252}{.002088} = 1.21$$

For 1.5 safety factor and 1.21 amplification factor the normal safety factor is,

$$SF = 1.5 \times 1.21 = 1.81$$

the allowable stress in the tension rod,

$$\sigma_{\text{allowable}} = \frac{\sigma_{\text{Yield}}}{SF} = \frac{38,920}{1.81} = 21,500 \text{ psi}$$

$$\sigma_{\text{Yield}} = 38,920 \text{ psi for Hastelloy X at } 1325^\circ\text{F, Ref. 1}$$

Since the stress in the rod ($\sigma_r = 18,224 \text{ psi}$) is less than the allowable stress ($\sigma_{\text{all}} = 21,500 \text{ psi}$) the design of the rod is safe.

The applied stress in the rod under 1-g loading condition

$$\sigma_r = \frac{18,224}{3} = 6074 \text{ psi}$$

The applied stress is less than the extrapolated rupture stress of Hastelloy X at 1350°F for 5 years (rupture stress = 9300 psi from Figure B-6).

The ratio of the moment in the necktube without tension support to the moment in the necktube with tension support is

$$f_b = \frac{M}{M_t} = \frac{W(\ell_1 + \ell_2)}{W(\ell_1 + \ell_2) - P(\ell_1 + \ell_2 + \ell_3)} = \frac{792(2.5 + 4.52)}{792(2.5 + 4.52) - 422.4(2.5 + 4.52 + 4.16)} = 6.64$$

The safety factor for buckling in combined bending and internal pressure versus different necktube wall thicknesses are shown on Figure B-14. It can be seen from Figure B-14 that with tension rod support the .0095" straight necktube is very safe for buckling in combined bending and internal pressure.

The ratio of the shear force in the necktube without tension support to the shear force with the tension support is

$$f_s = \frac{W}{W-P} = \frac{792}{792-422.4} = 2.14$$

The safety factor for buckling in transverse shear versus different necktube wall thicknesses for a .0095" thick straight necktube and different rod diameter tension supports are shown on Figure B-15. It can be seen from Figure B-15 that the .0095" thick straight necktube is safe in shear buckling.

The elongation of the tension rod, ($d_r = 11/64$ in)

$$S_r = \frac{P_r \ell_r}{A_r E_r} = \frac{422.44(3.5)}{(.02318)(24.8) 10^6} = .002571 \text{ in.}$$

The deflection of the cantilever beam at the tension support

$$\delta_{b_3} = 792(13.21 + 3.63 + 2.57) 10^{-6} - 422.44(22.64 + 3.63 + 4.04) = .002567 \text{ in}$$

The elongation in the rod, of course, is the same as the deflection of the beam at the tension support.

The total heat leak through the necktube and the 3 tension supports.

a) .0095" wall thickness necktube

$$q = q^* \frac{A}{A^*} = 18.83 \frac{.1464}{.1696} = 16.25 \text{ watts}$$

q^* = heat leak for straight necktube .011 thick, 2.497" long

A^* = cross-sectional area for straight necktube .011 thick

b) Tension support

$$q = k A \frac{T_2 - T_1}{L} = (.8055)(.02318) \frac{1325 - 150}{3.5} \frac{1}{3.41} = 1.83 \text{ watts}$$

$$k_{av} = 116 \left[\frac{\text{Btu} \cdot \text{in}}{\text{Ft}^2 \cdot \text{HR} \cdot ^\circ\text{F}} \right] = .8055 \left[\frac{\text{Btu} \cdot \text{in}}{\text{in}^2 \cdot \text{HR} \cdot ^\circ\text{F}} \right] \text{ from Figure 4}$$

$$q \text{ (3 rods)} = 3 \times 1.83 = 5.49 \text{ watts}$$

$$q_{\text{total}} = 16.25 + 5.49 = 21.7 \text{ watts}$$

Allowing 1.5 mil variation on the diameter for manufacturing tolerances.

Max thickness of necktube,

$$t_m = .0095 + .00075" = .01025"$$

$$A_{\text{max}} = .1580 \text{ in}^2$$

$$q_{\text{max}} = q^* \frac{A}{A^*} = 18.83 \frac{.1580}{.1696} = 17.54 \text{ watts}$$

$$q_{\text{tot}} = 17.54 + 5.49 = 23.03 \text{ watts}$$

The heat leak for different tension rod diameters are shown in Figure B-16.

The above calculations were repeated for 3/4" and 1-1/2" added necktube lengths. Required tension rod diameters and total heat leaks are plotted on Figure B-17.

Initial layouts of the tension rod support in the HTSI System indicate that the length of the tension rod supports will be approximately 2.5 inches. Preliminary calculations of the thermal expansion of the tension supports and the shield and

necktube indicate that the tension supports have to be installed at approximately 30° angle to the horizontal in order to support the shield both in a cold and hot condition without prestressing the rod from thermal expansion. The tension rod and the shield in the cold and hot conditions are shown in the diagram below

From the triangles shown on the diagram

$$X^2 + C_1^2 = l^2$$

$$(X + b)^2 + C_2^2 = a^2 l^2$$

$$X^2 + C_1^2 = l^2$$

$$X^2 + 2bx + b^2 + C_2^2 = a^2 l^2 = a^2(X^2 + C_1^2)$$

$$X^2(1-a^2) + 2bx + b^2 + C_2^2 - a^2 C_1^2 = 0$$

$$X = \frac{-2b \pm \sqrt{4b^2 - 4(1-a^2)(b^2 + C_2^2 - a^2 C_1^2)}}{2(1-a^2)} = \begin{matrix} + 20.98 \text{ in} \\ + .949 \text{ in} \end{matrix}$$

where,

$$a = 1.00452$$

$$b = .09937$$

$$c = .03435$$

$$C_1 = 2.16 \text{ (from layout)}$$

$$C_2 = C_1 - C = 2.16 - .03435 = 2.1256$$

$$\tan \theta = \frac{X}{C_1} = \frac{.949}{2.16} = .4393 \quad \alpha = 23.4^\circ$$

$$l = \frac{X}{\sin \theta} = \frac{.949}{.401} = 2.36 \text{ in}$$

The thermal expansion of the tension rod,

$$\Delta l = \alpha_m l \Delta T$$

$$(l + \Delta l) = a l = l + \alpha_m l \Delta T = l (1 + \alpha_m \Delta T)$$

$$\alpha_m = 8.76 \text{ micro inches/in deg } ^\circ\text{F (Ref 1)}$$

(Mean coefficient of thermal expansion between $79^\circ - 1350^\circ\text{F}$)

$$\Delta T = 666 - 150 = 516^\circ\text{F (See Figure B-19)}$$

$$a = 1 + \alpha_m \Delta T = 1 + (8.76) 10^{-6} 516 = 1.00452$$

Thermal expansion of the rod attachment in the radial direction, (Hastelloy X),

$$c = \alpha R \Delta T$$

$$R = 3.125 \text{ (from layout)}$$

$$C = (8.76) 10^{-6} (3.125) (1325 - 70) = .03435 \text{ in}$$

Thermal expansion of the shield and necktube in the axial direction,

Necktube,

$$\Delta l_n = (8.76) (10^{-6}) (3.156) 516 = .01426 \text{ in}$$

Shield,

$$\alpha = 13-15 \times 10^{-6} \text{ in/in } ^\circ\text{C} = 7.2 - 8.34 \times 10^{-6} \text{ in/in } ^\circ\text{F}$$

$$\Delta l_s = (8.34) 10^{-6} \left[5.7 + 2.625 (.866) \right] (1350 - 70) = .08511 \text{ in}$$

$$b = \Delta l_n + \Delta l_s = .01426 + .08511 = .09937 \text{ in}$$

For the case when $\alpha = 30^\circ$ and the length of the tension rod is 2.5 inches, the heat leak was calculated approximately from

$$q_{\text{rods}} = 3q_1^* \frac{l^*}{l} \frac{1}{\cos \alpha} = 3(1.83) \frac{3.5}{2.5} \frac{1}{.866} = 8.87 \text{ watts}$$

$q_1^* =$ heat leak through .171 dia. rod 3.5" long $\alpha = 90$ degrees

$l^* = 3.5$ in

d rod = .171"

$l = 2.5$ in

Heat leak for the 3 rod tension support versus added necktube lengths shown on Figure B-18.

COMPRESSION ROD SUPPORT CALCULATIONS

The equation to calculate the force in the compression support is the same as was used for the tension support.

For a straight neck tube of .0095" wall thickness and the original length $l_1 = 2.5$ in.

$$\begin{array}{lll}
 l_1 = 2.5 & E_b = 24.8 \cdot 10^6 & d_r = 1/4 \text{ (assume)} \\
 l_n = 2.78 & I_b = .4412 \text{ in}^4 & l_r = 1.5 \text{ in} \\
 l_2 = 4.52 & A_b = .1464 \text{ in}^2 & E_r = 22 \cdot 10^6 \text{ zirconium} \\
 l_3 = 1.5 & & \text{(Ref. 11)}
 \end{array}$$

Solving for the force in the rod,

$$P = \frac{792 \left[9.7 + 3.63 + 1.98 \right] 10^{-6}}{(1.39 + 12.19 + 3.63 + 2.38) 10^{-6}} = 618.9 \text{ lbs.}$$

The deflection at the support,

$$\delta_J = \frac{P l_r}{E A} = \frac{618.9 (1.5)}{22 \times 10^6 (.04906)} = .00860$$

$$\delta_{b3} = 792 (15.31) - 618.9 (18.2) = .00861 \text{ in}$$

The deflection at the center of gravity,

$$\delta_{b2} = 792 \left[7.72 + 3.63 + 1.64 \right] - 618.9 \left[9.7 + 3.63 + 1.97 \right] = .000818 \text{ in}$$

$$\omega_n = 1196 \text{ rad/sec}$$

$$\omega/\omega_n = .2625$$

$$y_o/a_o = \frac{.06892}{.9310} = .07402$$

$$y_o = .00086$$

$$AF = 1.06$$

$$SF = 1.5 \times 1.06 = 1.59$$

$$\sigma_{\max} = \frac{30,000}{1.59} = 18,867 \text{ psi}$$

$$\sigma_{\text{yield}} = 30,000 \text{ psi for zirconium}$$

The stress in the rod,

$$\sigma_{\text{rod}} = \frac{618.9}{.04906} = 12,615 \text{ psi}$$

Heat leak through 3 zirconium rods,

$$q = \frac{A}{L} \int_{T_1}^{T_2} k \, dT = \frac{A}{L} \int_{T_1}^{T_2} (a + bT + CT^2) \, dT = \frac{A}{L} 1185.44$$

$$k = a + bT + CT^2 = .9906 - .0001396T + 18.4 \cdot 10^{-8} T^2 \left[\frac{\text{Btu}}{\text{hr ft } ^\circ\text{F}} \right]$$

from Ref. (12)

$$q = 1185.44 \frac{A}{L} = \frac{1185.44 (.04906)}{(12) (1.5) (3.413)} = .944 \text{ watts}$$

$$q_{\text{tot}} = 3 \times .944 = 2.83 \text{ watts}$$

Heat leak through 3 zirconium rods are shown on Figure B-18 for different added neck tube lengths.

AXIAL BUCKLING STRENGTH

$$P' = \frac{\pi^2 EI}{4l^2}$$

(Ref. 2, Table XV, page 304)

$$\begin{aligned} E &= 22 \times 10^6 \text{ psi} \\ d &= .250 \text{ in} \\ I &= .000191 \text{ in}^4 \\ l &= 1.5 \text{ in} \end{aligned}$$

$$P' = \frac{3.14^2 (22) 10^6 (.000191)}{4 (1.5)^2} = 4617 \text{ lbs.}$$

The critical buckling stress,

$$\sigma = \frac{4617}{.04906} = 94,109 \text{ psi}$$

The critical buckling stress is higher than the compressive yield stress and therefore the rod is safe in buckling.

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SAFETY FACTOR FOR NECK TUBE IN BULKING - COMBINED BENDING AND INTERNAL
PRESSURE VS. NECK TUBE WALL THICKNESS (COLD END OF NECK TUBE)

FIGURE B-2

FIGURE 3-3

FIGURE B-4

FIGURE B-5

FIGURE 9-6

FIGURE B-7

FIGURE B-8

FIGURE B-9

FIGURE B-10

FIGURE B-11

FIGURE 5-12

FIGURE B-13

SAFETY FACTOR FOR NECK TUBE IN BUCKLING IN
COMBINED BENDING AND INTERNAL PRESSURE WITH
THREE TENSION ROD SUPPORTS VS. WALL THICKNESS
(COLD END OF THE NECK TUBE)

13-45

FIGURE B-14

B-45

FIGURE B-15

FIGURE B-16

B-48

FIGURE B-17

FIGURE B-18

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APPENDIX C

PART 1

Derivation of equations for tension and compression rod support systems.

a.) Deflection due to end load (W-P)

$$\delta_1 = \frac{(W-P) l_1^3}{3 E I}$$

b.) Deflection due to end moment

$$\delta_2 = \frac{[W l_2 - P(l_2 + l_3)] l_1^2}{2 E I}$$

c.) Shear deflection of the neck tube as a short beam.

$$\delta_s = \frac{2 (W - P) l_1}{A G}$$

Total deflection at point 1.

$$\delta_{b1} = \delta_1 + \delta_2 + \delta_3 = \frac{2 (W - P) l_1^3}{6 E I} + \frac{3(W l_2 - P l_2 - P l_3) l_1^2}{6 E I} + \frac{2 (W - P) l_1}{A G}$$

2. Rotation of beam at point 1.

a.) Rotation due to end load.

$$\Theta_1 = \frac{(W - P) l_1^2}{2 E I}$$

b.) Rotation due to end moment

$$\Theta_2 = \frac{[W l_2 - P(l_2 + l_3)] l_1}{E I}$$

3. Deflection of beam at point 3 due to rotation at point 1.

$$\delta_{3\Theta} = (\Theta_1 + \Theta_2)(l_2 + l_3) = \left[\frac{(W-P)l_1^2}{2EI} + \frac{[Wl_2 - P(l_2 + l_3)]l_1}{EI} \right] (l_2 + l_3)$$

4. Total deflection at point 3.

$$\begin{aligned} \delta_3 &= \delta_{bl} + \delta_{3\Theta} = \\ &= W \left[\frac{l_1^2 (2l_1 + 3l_2 + 3l_3) + 3l_1 l_2 (l_1 + 2l_2 + 2l_3)}{6EI} + \frac{2l_1}{AG} \right] \\ &- P \left[\frac{l_1^2 (2l_1 + 3l_2 + 3l_3) + 3l_1 (l_1 + 2l_2 + 2l_3)(l_2 + l_3)}{6EI} + \frac{2l_1}{AG} \right] \end{aligned}$$

5. Rotation of neck tube at point 3 due to deflection of flange and top head.

a.) Deflection of flange,

$$\Theta_{3F} = \frac{M}{\alpha E t^3} = \frac{W(l_{1N} + l_2) - P(l_{1N} + l_2 + l_3)}{\alpha E t^3}$$

$$\alpha = 44 \text{ (Ref. 2)} \quad E = 28 \times 10^6 \text{ psi} \quad t = .375"$$

$$\delta_{3F} = \frac{W(l_{1N} + l_2) - P(l_{1N} + l_2 + l_3)}{64.9 \times 10^6} (l_{1N} + l_2 + l_3)$$

b.) Deflection of top head,

$$\Theta_{3T} = \frac{Ma}{ET^3} = \frac{W(\ell_{1N} + \ell_2) - P(\ell_{1N} + \ell_2 + \ell_3)}{ET^3} \quad \text{where } \ell = 124.6 \quad (\text{Ref. 2})$$

$$\delta_{3T} = \frac{W(\ell_1 + \ell_2) - P(\ell_{1N} + \ell_2 + \ell_3)}{184 \times 10^6} (\ell_{1N} + \ell_2 + \ell_3)$$

$$\delta_{3T} \approx \delta_{3F}$$

$$\delta_{3F} + \delta_{3T} = 2 \delta_{3F} = 2 \frac{W(\ell_{1N} + \ell_2) - P(\ell_{1N} + \ell_2 + \ell_3)}{64.9 (10^6)} (\ell_{1N} + \ell_2 + \ell_3)$$

6. Total deflection at point 3 including rotation of neck tube due to flange and top head deflections.

$$\delta_{b3} = \delta_3 + 2 \delta_{3F} =$$

$$= W \left[\frac{\ell_1^2 (2\ell_1 + 3\ell_2 + 3\ell_3) + 3\ell_1 (\ell_1 + 2\ell_2 + 2\ell_3)}{6 E I} + \frac{2\ell_1}{A G} + \right.$$

$$\left. + \frac{2(\ell_{1N} + \ell_2)}{64.9(10^6)} (\ell_{1N} + \ell_2 + \ell_3) \right] -$$

$$- P \left[\frac{\ell_1^2 (2\ell_1 + 3\ell_2 + 3\ell_3) + 3\ell_1 (\ell_1 + 2\ell_2 + 2\ell_3) (\ell_2 + \ell_3)}{6 E I} + \frac{2\ell_1}{A G} + \right.$$

$$\left. + \frac{2(\ell_{1N} + \ell_2 + \ell_3)^2}{64.9 \cdot 10^6} \right]$$

7. Deflection in the tension or compression support.

$$\delta_r = \sum_r E_r \quad \Sigma_r = \frac{\delta_r}{E_r}$$

$$\delta_r = \ell_r \Sigma_r = \ell_r \frac{\delta_r}{E_r} = \frac{P \ell_r}{E_r A_r}$$

8. The load in the tension or compression rod -

The load in the rod is obtained by equating the deflection in the rod to the deflection of the beam

$$\delta_r = \delta_{b3}$$

$$P = \frac{W \left[\frac{l_1^2 (2l_1 + 3l_2 + 3l_3) + 3l_1 l_2 (l_1 + 2l_2 + 2l_3)}{6EI} + \frac{2l_1}{AG} + \frac{2(l_{1N} + l_2)(l_{1N} + l_2 + l_3)}{64.9 \times 10^6} \right]}{\frac{l_r}{AE_r} + \frac{l_1^2 (2l_1 + 3l_2 + 3l_3) + 3l_1 (l_1 + 2l_2 + 2l_3)(l_2 + l_3)}{6EI} + \frac{2l_1}{AG} + \frac{2(l_{1N} + l_2 + l_3)^2}{64.9 \times 10^6}}$$

If the tension or compression rod is installed at an angle 90° to the axial centerline, the load in the rod is obtained by dividing the right hand side of the equation by $\cos^2 \alpha$

9. The deflection at point 2 (center of gravity)

$$\delta_{b2} = W \left[\frac{l_1^2 (2l_1 + 3l_2) + 3l_1 l_2 (l_1 + 2l_2)}{6EI} + \frac{2l_1}{AG} + \frac{2(l_{1N} + l_2)^2}{64.9 (10^6)} \right] -$$

$$- P \left[\frac{l_1^2 (2l_1 + 3l_2) + 3l_1 (l_2 + l_3)(l_1 + 2l_2)}{6EI} + \frac{2l_1}{AG} + \right.$$

$$\left. + \frac{2(l_{1N} + l_2)(l_{1N} + l_2 + l_3)}{64.9 (10^6)} \right]$$

PART 2

CALCULATION OF CENTER OF GRAVITY

$$V_1 = \frac{7.100^2 - 5.060^2}{4} \pi (.645) = 12.560 \text{ in.}^3$$

$$V_2 = 7.900^2/4 \pi (.400) = 19.596 \text{ in.}^3$$

$$V_3 = 8.480^2/4 \pi 1.205 = 68.006 \text{ in.}^3$$

$$V_4 = 8.596^2/4 \pi 3.430 = 198.943 \text{ in.}^3$$

$$V_5 = \text{Area of quarter circle} \times \text{path of the center of gravity} =$$

$$= 2.618^2 \pi/4 \times 5.582 \pi = 94.277 \text{ in.}^3$$

$$V_6 = 3.360^2/4 \pi 2.618 = 23.198 \text{ in.}^3$$

$$V_{\text{TOT}} = 416.580 \text{ in.}^3$$

$$\bar{X} = \frac{V_1 (.322) + V_2 (.845) + V_3 (1.647) + V_4 (3.965) + V_5 (6.791) + V_6 (6.989)}{V_{\text{TOT}}} =$$

$$= \underline{4.137''}$$

Location of C. G. from cold end of neck tube = $3.156 + 4.137 = 7.293 \sim \underline{7.3}$

To beginning of .011" section $7.293 - .285 = \underline{7.008''}$

NOTE:

Center of a quarter circle

$$\bar{y} = \frac{4r}{3\pi} = \frac{4 \times 2.618}{3\pi} = \underline{1.111''}$$

NOTE: This Appendix originally appeared as Appendix 2 of Report No. 3.

APPENDIX D

DERIVATION OF EQUATION FOR TEMPERATURE DISTRIBUTION IN A ROD

WITH ONE DIMENSIONAL HEAT CONDUCTION AND VARIABLE

THERMAL CONDUCTIVITY

If the thermal conductivity of the material varies significantly in the conduction region the Laplace equation is not valid in such a case. The proper differential equation is determined from the general conduction equation. The equation and the boundary conditions are (from Reference 18):

$$\frac{d}{dX} \left(k \frac{dt}{dX} \right) = 0 \quad \text{at} \quad \begin{array}{ll} X = 0 & t = t_1 \\ X = L & t = t_2 \end{array}$$

where k may be a function of X and t . The solution for this equation is presented below for the case when the thermal conductivity is a function of temperature, that is $k = f(t)$. Integrating and solving for the constants,

$$k \frac{dt}{dX} = C_1$$

$$\int_{t_1}^t k dt = C_1 \int_0^X dX = C_1 X$$

from $t = t_2$ at $X = L$

$$\int_{t_1}^{t_2} k dt = C_1 L$$

$$C_1 = \frac{1}{L} \int_{t_1}^{t_2} k dt = -k_m \frac{t_1 - t_2}{L}$$

$$k_m = \frac{1}{t_1 - t_2} \int_{t_1}^{t_2} k dt$$

where k_m is defined as the integrated value of k from t_1 to t_2 . For a linear variation of k with temperature this is equal to the average of k_1 and k_2 . The temperature distribution is given by

$$\int_{t_1}^t k dt = - k_m \frac{t_1 - t_2}{L} X$$

$$k = f(t) = a t + b$$

For Hastelloy-X material and $t_1 = 1325^\circ\text{F}$ and $t_2 = 150^\circ\text{F}$ it was shown in Appendix B, page B-13 that,

$$k = .07659 t + 59.511 \quad \text{Btu-in/Ft}^2\text{Hr}^\circ\text{F}$$

then,

$$k_1 = .07659 (1325) + 59.511 = 161 \quad \text{Btu-in/Ft}^2\text{Hr}^\circ\text{F}$$

$$k_2 = .07659 (150) + 59.511 = 71 \quad \text{Btu-in/Ft}^2\text{Hr}^\circ\text{F}$$

$$k_m = k_1 + k_2 / 2 = 116 \quad \text{Btu-in/Ft}^2\text{Hr}^\circ\text{F}$$

$$\int_{t_1}^t k dt = \int_{t_1}^t (a t + b) dt = \left[\frac{a t^2}{2} + b t \right]_{t_1}^t =$$

$$\frac{a t^2}{2} + b t - \frac{a t_1^2}{2} - b t_1 = - k_m (t_1 - t_2) \frac{X}{L}$$

$$\frac{a t^2}{2} + b t - \left[\frac{a t_1^2}{2} + b t_1 - k_m (t_1 - t_2) \frac{X}{L} \right] = 0$$

Solving the quadratic equation for t

$$t = \frac{-b \pm (b^2 + a^2 t_1^2 + 2 a b t_1 - 2 a k_m (t_1 - t_2) \frac{X}{L})^{1/2}}{a}$$

where, $a = .07659 \frac{\text{Btu-in}}{\text{Hr Ft}^2}$

$$b = 59.511 \frac{\text{Btu-in}}{\text{Hr Ft}^2\text{F}}$$

$$\begin{aligned}
t_1 &= 1325^\circ\text{F} \\
t_2 &= 150^\circ\text{F} \\
k_m &= 116 \text{ Btu-in/Ft}^2\text{Hr}^\circ\text{F} \\
\frac{X}{L} &\text{ dimensionless length ratio}
\end{aligned}$$

using the above equation the temperature distribution in a Hastelloy-X rod, or neck tube was calculated. Graphical presentation of the results is shown in Appendix A, page A-28, Figure A-5.

The integrated mean temperature in the rod is defined as

$$t_m = \frac{1}{y_2 - y_1} \int_{y_1}^{y_2} t \, dy$$

where y was substituted for $\frac{X}{L}$

$$t = -\frac{b}{a} \pm \frac{1}{a} (\alpha - \beta y)^{1/2}$$

$$\begin{aligned}
\text{where } \alpha &= b^2 + a^2 t_1^2 + 2 a b t_1 = 25,908.06 \\
\beta &= 2 a k_m (t_1 - t_2) = 20,878.48
\end{aligned}$$

$$t_m = \frac{1}{1 - 0} \int_0^1 \left(-\frac{b}{a} \pm \frac{1}{a} (\alpha - \beta y)^{1/2} \right) dy =$$

$$= \left[-\frac{by}{a} \pm \frac{1}{a} \left(-y \right)^{3/2} / (1/2 + 1)\beta \right]_0^1$$

$$t_m = -\frac{b}{a} \pm \frac{(\alpha - \beta)^{3/2}}{1.5 a \beta} \pm \frac{\alpha^{3/2}}{1.5 a \beta}$$

$$t_m = -\frac{b}{a} - \frac{\alpha - \beta^{3/2}}{1.5 a \beta} + \frac{\alpha^{3/2}}{1.5 a \beta} = 812^\circ\text{F}$$

The same answer was also obtained by numerical integration.

Knowing the temperature distribution in the rod, the thermal conductivity is calculated for different X/L ratios.

$$k = .07659 t + 59.511$$

$\frac{X}{L}$	t (From Figure A-5)	k = f (X/L, t) Btu-in/Ft ² Hr °F
0	1325	161.0
.2	1147	147.3
.4	953	132.5
.6	732	115.5
.8	476	95.9
1.0	150	71.0

The thermal conductivity versus X/L plot is given in Appendix A, page A-26, Figure A-3.

A mean thermal conductivity of 121.8 Btu-in/Ft²Hr°F was obtained by numerical integration of the curve shown on Figure A-3.

APPENDIX III

AGENCIES AND ORGANIZATIONS CONTACTED IN SUPPORT OF THIS STUDY

APPENDIX III

AGENCIES AND ORGANIZATIONS CONTACTED IN SUPPORT OF THIS STUDY

Packaging Research, Division of Wood Engineering
U. S. Forest Service, Madison, Wisconsin

Environmental Operations Analysis Group
Sandia Corporation, Albuquerque, New Mexico

General American Research Division
General American Transportation Corporation
Niles, Ohio

U. S. Navy, Bureau of Ships
Washington, D. C.
 Auxiliary Vessels Division
 Material and Weapons Handling Equipment Division
 Dock Handling Division
 Submarine Division

David Taylor Model Basin
Washington, D. C.

Naval Civil Engineering Laboratory
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Scripps Institute of Oceanography
San Diego, California

U. S. Navy Electronics Laboratory
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Woods Hole Oceanographic Institute
Woods Hole, Massachusetts

Naval Ordnance Test Station
China Lake, California

Naval Underwater Ordnance Station
Newport, Rhode Island

Naval Underwater Sound Station
Newport, Rhode Island

General Electric Company
Defense Electronics Division
Syracuse, New York

APPENDIX IV

SUMMARY REPORT ON PRELIMINARY STUDY OF THE SHOCK AND VIBRATION ENVIRONMENT FOR TRANSPORTATION AND OCEAN IMPLANTMENT OF A SNAP-21 GENERATOR

SUMMARY REPORT

on

PRELIMINARY STUDY OF THE SHOCK AND VIBRATION
ENVIRONMENT FOR TRANSPORTATION AND OCEAN
IMPLANTMENT OF A SNAP-21 GENERATOR

to

MINNESOTA MINING AND MANUFACTURING COMPANY

January 18, 1967

by

R. H. Prause, J. J. Ketchman, R. E. Mesloh
and A. J. Coyle

BATTELLE MEMORIAL INSTITUTE
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505 King Avenue
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January 18, 1967

Mr. Ralph Wickenberg
Minnesota Mining and Manufacturing Company
2501 Hudson Road
St. Paul, Minnesota

Dear Mr. Wickenberg:

Enclosed is the final report for the "Preliminary Study of the Shock and Vibration Environment for Transportation and Ocean Implantment of a SNAP-21 Generator". Because of the limited time available for this study (approximately two weeks), the results are based on several assumptions designed to give conservative estimates of the dynamic loads. If these loads appear too severe to be used as a design criteria, it is recommended that a more refined analysis be considered to obtain more realistic results. In particular, the effect of the generator foundation structure has been neglected in the water impact analysis and for the implantment dynamics. This structure should reduce the water impact accelerations and its structural stiffness characteristics are necessary in order to calculate bottom impact loads.

If there are any questions regarding this report please contact me or Art Coyle at your convenience. We have enjoyed the opportunity to participate in this study and will be glad to discuss possibilities for future projects with you.

Sincerely yours,

Robert H. Prause
Research Engineer
Machine Dynamics Division

RHP:js
In sextuplicate

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PRELIMINARY STUDY OF THE SHOCK AND VIBRATION
ENVIRONMENT FOR TRANSPORTATION AND OCEAN
IMPLANTMENT OF A SNAP-21 GENERATOR

by

R. H. Prause, J. J. Ketchman, R. E. Mesloh
and A. J. Coyle

INTRODUCTION

Minnesota Mining and Manufacturing Company (3M) is designing an isotopic power generator designated SNAP-21. This generator will be implanted on the ocean bottom in water depths which could vary from only a few feet to 23,000 feet. Battelle has been requested to provide a preliminary estimate of the dynamic loads which the generator could be expected to encounter from the time it leaves the manufacturing plant until it is safely implanted on the ocean bottom. This includes handling and shipping loads from common carrier transportation by air, rail or road from the plant to a ship loading site, loads due to ship transport to the implantment location as well as the loads which may result from the actual implantment procedure.

Data available from the literature for common carrier transportation and handling are presented and discussed along with the pertinent vibration and shock test specifications. Similar data for the vibration environment for ship transportation are also discussed in addition to calculations of ship motions in response to a wide range of sea states. These ship motions were used to calculate the generator response during implantment in order to estimate bottom impact velocities. Also considered were water impact shock loads which would result from dropping the generator from the ship or from a large wave striking the generator as it was suspended over the ship's side.

The very limited time available for this project (approximately two weeks) necessitated several approximations as to the type of ship that would be used and the effect that an implantment foundation for the generator would have on the generator dynamics during implantment. In every case, the assumptions have been made to produce a conservative estimate of the dynamic loads, i.e., more severe than would actually be required. For this reason, the predicted loads are to be considered preliminary.

SUMMARY AND CONCLUSIONS

The Atomic Energy Commission regulations specify drop tests for packaged radioactive materials to ensure that the packages will not become dangerous as a result of transportation handling. These specifications are too severe for design loads after which the generator is expected to operate. Less severe drop tests given in MIL-STD-810A are more suitable for determining design loads representing common carrier transportation handling.

Data from the literature indicate that the vibration and shock tests given in MIL-STD-810A are also adequate for road, rail and air common carrier transportation. If these tests appear too severe, the possibility of special handling and transportation for the SNAP-21 system should be considered. The primary vibrations encountered during transport by ship to the implantment location are the vibrations from the ship's propellers and other rotating equipment. Specifications for these vibrations are given in MIL-STD-167.

The vibration due to ship motions appears negligible even for severe storms because the frequency, and hence the accelerations are low. The ship motion is important because it represents the input excitation for the generator

and cable as the generator approaches the ocean bottom. The ship motions analysis uses regular, sinusoidal waves and does not include accelerations from ship slamming. This would be an additional shock acceleration but is not expected to be as high as the specifications for handling and common carrier transportation.

Braided polypropylene rope appears quite suitable for use in implantment and is recommended rather than wire rope because the polypropylene floats and, therefore, eliminates the sheathing or buoys required to support the static weight of wire. The most dangerous depths for implanting the generator are those where the frequency of the ship's motion coincides with the resonant frequency of the generator and cable. With polypropylene rope, these depths are from 120 to 400 feet. If steel cable is used, the most dangerous depths would be from 2900 to 7000 feet. This represents a wider range of "critical" depths and a range of depths more likely to be used for implantment than the critical range for the polypropylene rope. Maximum impact velocities of 16 feet per second due to generator oscillation, in addition to the normal descent rate, can be expected when implanting with either wire or polypropylene cable and a sea state with 5-foot waves. This impact velocity can only be converted to a shock load by considering the deformation of the foundation at impact with bedrock.

Two conditions of water impact represent loads which may be caused by dropping the generator into the water or by the impact of a wave on the generator while it is suspended from the ship prior to release. Accelerations up to 73 g's are estimated for a vertical drop of 10 feet for either the 10 or 20 watt generators. Lateral accelerations corresponding to impact from a 5-foot wave (maximum sea-state considered feasible for implantment) are 141 and 101 g's for the 10 and 20 watt generators, respectively. However,

these estimates are based on the generator impacting a flat water surface at the characteristic wave speed and do not consider the effects of a foundation to "break" the impact or the more realistic case where the wave will strike the generator at some angle other than broadside.

Refined calculations should include the effect of the generator's foundation on the water impact and its contribution to the damping of generator oscillations during implantment. The structural stiffness characteristics of the foundation will also be important in determining the bottom impact shock loads. It may be necessary to limit the bottom impact loads to safe values by designing the foundation with sufficient flexibility and energy absorbing capability, based on these predicted impact velocities. There are also several possibilities for increasing the damping of the generator and thereby reducing the bottom impact velocity significantly in the "critical" depth range.

TECHNICAL WORK

Shock and Vibration Environment in Ships

During transport of the generator from land to its implantment location, the generator and its shipping coffin will be subjected to and must survive the ship's vibration environment. Principal causes of steady-state vibration are propeller blade excitation and unbalance from the propeller and its shafting. In most vessels the vibration frequencies encountered vary from zero to approximately 25 cps. Typical vibration data for various ships can be found in Reference (1). These or similar data have been used to formulate the vibration test specification in MIL-STD-167, Reference (2), for shipboard equipment, and, therefore, are applicable as the input excitation for the

generator shipping container during transport.

Table A-1 lists these vibration test specifications. The test procedure in MIL-STD-167 requires first an exploratory test to determine resonances. This consists of varying the frequency from 5 to 33 cps with a constant displacement amplitude of 0.010 ± 0.002 inch. Frequency increments are 1 cps with 15 sec vibration at each frequency to note resonances. The second test phase requires a repeat of the above frequency range using the amplitudes listed in Table A-1 and dwelling at each integral frequency for 5 minutes. The final and most severe test is to dwell at each of the resonances, or if no resonances were observed, the test is to be performed at 33 cps for a minimum of 2 hours total vibration time. The exception to this is that if the equipment is to be placed on a particular class ship with propeller excitation below 33 cps, the equipment does not have to be tested at frequencies above the propeller excitation frequency.

In addition to the above vibration environment, the ship's motion should also be considered. However, the frequencies are so low it is doubtful if these vibrations will be damaging, but the ship motions must be known in order to estimate the generator motion during implantment. The ship motions, consisting of pitching* and heaving amplitudes, circular frequency and phase angles, were determined over a range of sea states by using a ship motions computer program. Briefly, this program calculates the amplitude and phase angle for the heaving and pitching motions and dynamic components of the longitudinal bending moment and shear force at any cross section for a ship subjected to a regular or sinusoidal wave. By the dynamic components of the bending moment (or shear force) is meant those caused by the hydrodynamic forces acting on the hull and the acceleration forces (caused by heaving and

* Pitching was selected because this motion presents the most severe dynamic condition.

pitching motion) which act on the hull and cargo. Following are the basic assumptions underlying the computer calculations:

- (1) The problem is only considered in two dimensions, the wave is two-dimensional and the only degrees of freedom are heave and pitch.
- (2) Only the steady-state solution of the governing differential equations of motion is considered.
- (3) The calculations are based on the ship motion theory of Korvin-Kroukovsky, Reference (3), which assumes the motions to be in the linear range, and uses strip theory.
- (4) The ship is assumed to be a completely rigid body.

The input to the program is the characteristics of the ship consisting of the ship's speed (if any), the length and the mass or weight distribution, beam, draft, and sectional area coefficient, all as functions of length, and the characteristics of the wave, namely its length, λ , and amplitude, a , or one-half the peak-to-peak height, h . The program output is that given above where the phase angles are lagging when referenced to the wave node occurring at the longitudinal center of gravity of the ship. The pitch angle, θ_p , is the angle of rotation taken around this center and the heave, H , is the vertical motion of this point.

As the specific geometry of the implanting ship was unknown, a worst case geometry was assumed by using a flat bottom barge with a geometry of

Length	= 100 feet
Beam	= 30 feet
Draft	= 5 feet
Sectional Coefficient	= 1.

The displacement and mass distribution for the ship was assumed as being directly proportional to the draft. This assumption will affect the response of the ship if the true weight distribution, actually the rotary inertia, is greatly different than the assumed one. However, the treatment of this case does give representative indications of the expected ship motions.

In Figure B-1 are presented the pitch, heave and circular frequency, ω , of the motions at the stern or bow of the vessel as a function of the ratio of wave length to ship length. These normalized pitch and heave amplitudes must be multiplied by the given wave amplitudes of interest, which are one-half the peak-to-trough height of the waves, to determine the actual amplitudes for a given sea state. The phase angle between the pitching and heaving motions is approximately 90 degrees for all but the shortest wave lengths.

As observed from Table A-2 and Figure B-2 there are preferred wave heights (twice wave amplitude) for a given wave length and visa versa. (One would not expect a 10-foot wave to have an 80-foot length or a 5-foot wave to have a 200-foot length). Consequently in computing the actual pitching or heaving motions from Figure B-1 the average wave height for a given wave length from Figure B-2 should be used. Figure B-3 shows the resulting amplitude at the stern of the barge, assuming the pitch and heave are in phase for a conservative case. Table A-3 summarizes data from Figure B-3 for the ship's response in sea conditions up to 20-foot waves, to include the possibility of a storm during transportation.

The shock environment normally considered for ships (naval in particular) is that caused by underwater explosions from near misses and air blast from the ship's own armament. These have not been considered applicable for the generator, so the only shocks will be those for normal transportation handling, which will be discussed in a later section.

Underwater Implantment

Two occurrences resulting in high accelerations have been considered possible while the generator is suspended over the side of the ship but prior to its entering the water. The first possibility is the case where the generator is inadvertently dropped so that it impacts the water in free-fall. The second possibility is that a wave may strike the generator on its side, producing high lateral accelerations. Both these conditions are similar in that they involve an impact between the generator and water. In the first case it has been assumed that the generator enters the water with its axis perpendicular to the water surface and at an impact velocity, V , related to the drop height, h , by

$$V = \sqrt{2gh} \quad (1)$$

where

g = gravitational acceleration.

In the second case it has been assumed that the generator strikes the wave with its axis parallel to the wave's surface and the impact velocity corresponds to the wave velocity for a particular sea state. A sea state with 5-foot waves is considered the maximum condition under which implantment would be attempted. Assumed weights of 1000 pounds for the 10 watt generator and 1200 pounds for the 20 watt generator include the foundation; however, the effect of the foundation on the impact was neglected entirely to give a conservative prediction of maximum accelerations. The foundation would be expected to spread the impact over a longer time period, thereby producing lower accelerations than those calculated for the blunt generator body alone.

The general water impact problem is quite involved, however, the upper limit of the impact force, $F(t)$, and consequently the deceleration can be determined from the equation given by Skalak and Feit in Reference (4).

$$F(t) = \rho c \bar{V}_o A_o \quad (2)$$

where

ρ = density

c = acoustic velocity

$\bar{V}_o$ = mean velocity of impact on area A_o

A_o = body area.

The general assumptions which accompany the use of this equation are presented in Reference (4). The deceleration of an impacting body can be determined by digital computer means from this equation and the impulse-momentum equation.

$$F(t)\Delta t = m\Delta V \quad (3)$$

where

Δt = time step

m = mass of the body

ΔV = change in velocity.

Equation (3) can be solved for the change in velocity

$$\Delta V = F(t)\Delta t/m \quad (4)$$

which substituted in Equation (2) produces

$$\Delta V = \rho c V A \Delta t/m \quad (5)$$

where

$$V = V(t) = V(t-\Delta t) - \Delta V(t-\Delta t)$$

$$A = \text{area (function of penetration).}$$

By proper substitution of the capsule's mass and projected area, A, as a function of penetration, Equation (5) was progressively solved by use of a digital computer program. In Figures B-4 through B-7 are presented the acceleration curves for both vertical and lateral wave impact as a function time. The pertinent shock parameters for the two capsules are tabulated in Tables A-4 and A-5. The symbols in these tables are defined as follows:

$$V_0 = \text{initial impact velocity}$$

$$a = \text{peak acceleration}$$

$$V = \text{velocity at peak acceleration}$$

$$t_1 = \text{time of peak acceleration}$$

$$X_1 = \text{penetration at peak acceleration.}$$

Cable Characteristics

In order to analyze the dynamics of the generator during underwater implantment, it was first necessary to consider the type of cable that would be suitable for this task. A fundamental requirement for this cable is that it be a nonrotating design. Most of the standard wire and fabric rope designs will rotate when a single, long length of rope is used in tension. However, nonrotating designs of both polypropylene rope and wire rope are available and are suitable to use for ocean implantment.

A considerable amount of very useful information about the techniques and equipment for the implantment and retrieval of submersible test units is presented in Reference (6). These test units ranged in weight from 5200 to 8750 pounds and 1-inch diameter polypropylene braided rope with a breaking strength

of 21,000 pounds was used extensively. An important advantage of this material over wire rope is that the specific gravity of the polypropylene is about 0.9, so the rope is buoyant. Wire rope would require a thick sheathing of a light material or a series of buoys to support the cable weight at great depths. Without this, the cable would have to be sized to support its own weight at 23,000 feet in addition to that of the generator. This would require a considerable increase in the size of the shipboard implantment and retrieval equipment, a seemingly unnecessary penalty.

Test results reported in Reference (6) for the polypropylene rope indicate that exposure to sea water does not effect its strength or buoyancy. The material has been found to be inert to those chemicals normally found in sea water and does not rot. On the other hand, steel cable has been known to corrode under continual exposure because of the inability of oxygen to penetrate the crevices between cable strands. The only apparent advantage of steel cable is that it is somewhat more resistant to abrasion, however, the experience with the polypropylene indicates that it is entirely satisfactory.

For analysis purposes, 5/8-inch diameter braided polypropylene rope with a breaking strength of 8000 pounds and a weight of 0.1 pound per foot was selected. This rope has an effective stiffness of 22,850 lb/ft for a one-foot length, and a maximum elongation of 35%. For comparison, an 18 x 7 nonrotating design wire rope with a breaking strength of 7800 pounds, weight of 0.169 pounds per foot, and an effective stiffness of 548,000 lb/ft for a one-foot length has been used.

Dynamic Analysis

During the process of lowering the generator to the ocean bottom, the generator will oscillate in a vertical direction in response to the ship's motion. This presents some possible difficulty when the generator approaches the bottom because the oscillation velocity superimposed on the normal descent rate may result in an excessive landing shock. In order to calculate the oscillation velocity for implantment at different depths, it is necessary to consider the dynamics of the generator and supporting cable.

The generator will be supported by a single cable, which will be attached to the ship. For this analysis it was assumed that the generator would be lowered from the ship's stern so that the ship's motion due to pitch and heave (most severe motion) reported in a previous section can be considered as the excitation amplitude of the cable end. It will also be assumed that the cable remains straight in a vertical plane so that the dynamics of the generator and cable are adequately represented by the solution for an end weight attached to a uniform bar which is excited in the longitudinal direction. The exact solution for this problem is given in Reference (7) for all the possible cases resulting from the variation of the relative weights of the end mass and bar, or cable in this case. If the mass of the cable is negligible compared to the end mass, or generator, then it can be neglected and the problem is reduced to that of a single-degree-of-freedom system with the spring rate determined by the static stiffness of the cable. If the generator mass is negligible relative to the total cable mass, the solution becomes the same as that for a "fixed-free" bar. As the generator is lowered over the side, the cable mass becomes increasingly more important until, at 23,000 feet, the cable mass is 2300 pounds, which is more than twice the effective mass of either the 10 watt or 20 watt generators. For this range of relative sizes, the lowest natural frequency of the generator and cable for

vertical oscillations is given in Reference (7) as

$$f_n = \frac{1}{2\pi} \sqrt{\frac{AE}{\ell} \left(\frac{1}{M + m\ell/3} \right)} \quad (6)$$

where

- f_n = cyclic frequency, cps
- AE = stiffness per unit length
- M = effective generator mass
- m = mass per unit length of the cable
- ℓ = cable length.

The generator effective mass, M , is the sum of the actual generator mass plus an additional hydrodynamic mass to account for the behavior in water. Considering only the generator body in the absence of details of the foundation structure, the hydrodynamic mass, calculated according to Reference (8), is 50 pounds for the 10 watt generator and 65 pounds for the 20 watt generator. Therefore, the assumed effective mass used in the dynamic analysis was 1050 pounds for the 10 watt generator and 1265 pounds for the 20 watt generator.

Figure B-8 shows the resonant frequencies from Equation (6) for the two generators as a function of line length. Superimposed on this figure is the band of frequencies that will be excited by the ship's motion, see Figure B-3. For example, 1.3-foot waves occur at a frequency of 0.36 cps while larger waves occur most often at lower frequencies. The predominant frequency for a sea state of 5-foot waves is 0.21 cps. This is the most severe sea state considered practical for generator implantment or retrieval. Also shown on this figure are the resonant frequencies using the 5/16-inch diameter wire rope with the 10 watt generator.

The maximum generator vibration will occur when the resonant frequency coincides with the predominant frequency of the ship's motion. With the polypropylene cable, this occurs at depths of approximately 120 to 400 feet. If the steel cable were used, maximum vibration would be excited at 2900 feet to 7000 feet, a much greater range of depths and probably much closer to the expected implantment depths. These regions of resonance represent the most dangerous depths for implanting the generators. This is verified by Figure B-9, which shows the maximum generator impact velocity as a function of line length, or bottom depth, for the maximum wave height of 5 feet and an intermediate sea state of 3.5-foot waves. The resonant response is evident at the depths where the ship's frequency and generator frequency coincide. At greater depths the generator becomes progressively less sensitive to ship's motion and the oscillation velocity becomes negligible relative to the normal descent rate. This observation is verified in Reference (6). If the wire cable is considered, resonance will occur at those depths indicated previously, and the maximum oscillatory velocities will be the same as those given for the polypropylene cable.

The curves shown in Figure B-9 are based on the response equation for a damped single-degree-of-freedom system.

$$V_m = \frac{2\pi f_n Y_o}{\left[\left(1 - \frac{f^2}{f_n^2}\right)^2 + \left(2\beta \frac{f}{f_n}\right)^2 \right]^{1/2}} \quad (7)$$

where

- V_m = maximum impact velocity, ft/sec
- Y_o = displacement amplitude at ship's stern, feet (Figure B-3)
- f = frequency of ship's motion, cps (Figure B-3)
- f_n = generator natural frequency, cps (Figure B-8)

and β is the damping ratio given by

$$\beta = \frac{C}{4\pi f_n (M + m\ell/3)} \quad (8)$$

for a viscous damping coefficient, C , based on the fluid drag on the generator and cable.

The damping of the generator is very important at resonance, $f = f_n$, so it was necessary to consider this in detail in order to calculate the resonant response from Equations (7) and (8). The line damping, C_ℓ , due to viscous shear from the water can be obtained from an exact solution given in Reference (9).

$$C_\ell = \mu \pi d \left(\frac{\ell}{3}\right) (\pi f / \nu)^{1/2} \quad (9)$$

where

μ = fluid viscosity, 3.8×10^{-5} lb-sec/ft² for water

d = line diameter, feet

ℓ = line length, feet

ν = fluid kinematic viscosity, 1.82×10^{-5} ft²/sec for water.

Substituting into Equation (9) for the polypropylene cable gives

$$C_\ell = 3.8 \times 10^{-4} \ell \quad (\text{lb-sec/ft}) \quad (10)$$

There is no exact solution for the damping of an oscillating body similar to the generator configuration, so the results for the steady-state drag of a sphere, Reference (10), have been used together with Equation (9) applied to the cylindrical body. The drag force for the spherical domes is

$$D = C_d \frac{\rho V^2}{2} A \quad (11)$$

where

D = drag force, pounds

C_d = drag coefficient, 0.47 for high Reynolds number

ρ = fluid density, $2.08 \text{ lb-sec}^2/\text{ft}^4$ for water at 10,000 feet

A = generator cross-sectional area, 1.39 ft^2 - 10 watt generator
 1.76 ft^2 - 20 watt generator

V = generator velocity, ft/sec.

Equation (11) is nonlinear and not directly applicable for obtaining a linear viscous coefficient. However, an equivalent viscous coefficient can be calculated that gives the same total energy dissipation over a complete cycle. This equivalent damping coefficient for the generator dome, C_{gd} , is

$$C_{gd} = \frac{C_d \rho A V_m}{2} \quad (12)$$

where V_m is the maximum oscillatory velocity defined by Equation (7). The body drag $C_{gb} = 0.044 \text{ lb-sec/ft}$ for the 10 watt generator and $C_{gb} = 0.049 \text{ lb-sec/ft}$ for the 20 watt generator.

The final equation for the total damping coefficient is obtained by combining the above results for the 10 watt generator to give

$$C = 3.8 \times 10^{-4} \ell + 0.044 + 0.685 V_m \quad (13)$$

Because the damping coefficient C was still dependent on the oscillation amplitude, it was necessary to iterate the solutions from Equations (7), (8), and (13) to calculate the curves given in Figure B-9.

Terminal Velocity

There are several possible failures that could possibly happen during implantment or retrieval that would result in the generator sinking freely and impacting on the ocean bottom. Examples of these are cable failures, structural failure of the lowering mechanism, or inadvertent release of the cable winch brakes. Provided there is sufficient depth, the generator and foundation will accelerate until the drag forces become great enough to equal the net generator weight in water. A buoyant force of 290 pounds was used for the 10 watt generator and foundation and 350 pounds was used for the 20 watt generator and foundation. The terminal velocity was calculated using Equation (12) and a drag coefficient of $C_d = 0.25$, for Reynolds numbers greater than those resulting from the oscillation, Reference (10). The terminal velocity for the 10 watt generator is 44 ft/sec and the velocity for the 20 watt generator is 43 ft/sec. The increased drag due to the larger diameter of the 20 watt generator more than compensates for its increased net weight, resulting in a slightly lower terminal velocity.

The terminal velocity is nearly the same at all depths unless it is assumed that all of the line is attached. The additional drag and buoyant force from the polypropylene line reduces the terminal velocity of the 10 watt generator to 35 ft/sec at 23,000 feet, and the corresponding value for the 20 watt generator is 36 ft/sec.

Impact Shocks

The conversion of ocean bottom impact velocity to a corresponding shock pulse requires detailed information about the characteristics of the generator and foundation structure. The conservative assumption is that the bottom will be bedrock, so that the structure represents the predominant flexible element in which to dissipate the impact energy. The shock loads predicted from

the impact of a rigid generator on water demonstrate the high loads which can be obtained during impacts. It may be necessary to design the generator foundation to limit the impact shock loads to a safe value, based on the predicted impact bottom velocities.

Vibration and Shock Environment For Road, Rail, and Air Transport

Two significant environmental conditions that affect the design of transportable devices and packages are the vibration environment and the shock environment. Vibration and shock are encountered in all phases of package transport. The maximum allowable acceleration a body experiences as a result of the environment is determined by the damping characteristics and natural frequencies of the components involved. In the case of a protected package, the natural frequencies of important components are generally far greater than that of the package cushioning. Under these conditions the damage tends to be directly proportional to the maximum acceleration. The ensuing discussion covers the prime sources of vibration and shock disturbances that represent the excitation applied to the shipping container of the SNAP-21 generator in road, rail, and air transport and pertinent regulations and specifications. A general case of drop impact is also described. It has been assumed that the total shipping weight of the 10 watt generator is 1000 pounds and the 20 watt generator is 1200 pounds, which includes an estimated weight of 400 pounds for the shipping container.

Shock Isolation and Handling Loads

In many cases the shock received during handling is greater than that experienced in normal vehicular transport. A free drop of four feet onto an unyielding surface is considered in normal conditions of transport for packages weighing less than 10,000 pounds according to the Atomic Energy Commission's Rules and Regulations for the Packing of Radioactive Material for Transport, Reference (11). Such an impact, though probably not causing fuel release or prohibitively high radiation levels, may prove to be destructive to a SNAP device from the standpoint of internal structural integrity and power producing ability. However, packages weighing more than 1000 pounds will normally be handled with heavy equipment and skilled operators, which reduces the likelihood of severe drops. If the package is protected by a shock isolation medium, the maximum acceleration experienced by the package is a function of the package weight and the stiffness of the isolating medium. The general treatment of an idealized system is briefly described.

The standard shock test for packages is a free fall drop in which the package strikes a rigid surface. This is treated by Franklin and Hatae, Reference (12), by considering the model shown in Figure B-10. The maximum acceleration experienced by the package, $\ddot{x}$, is given by the following expression.

$$\ddot{x} = \frac{2\pi f_n \sqrt{2gh}}{g} + 1 \quad (14)$$

where

$\ddot{x}$ = dimensionless maximum acceleration, g's

g = gravitational acceleration

h = drop height, inches

f_n = natural frequency of package, cps.

The natural frequency of the package may be determined from Equation (15).

$$f_n = \frac{1}{2\pi} \left(\frac{kg}{W} \right)^{1/2} \quad (15)$$

where

k = stiffness of isolation medium, lb/in.

W = total package weight, lb.

Figure B-11 illustrates the dependence of 'x' on the drop height and natural frequency of the package as given by Equation (14). The above analysis was greatly simplified by considering linear and independent spring constants and neglecting damping; however, Figure B-11 may be used for estimation purposes if the design of a shock isolation envelope is being considered.

AEC rules and regulations for packaging radioactive material for transport apply to the SNAP-21 package as a whole. Areas specifically applicable to the handling or shock environment are the following:

- (a) A package weighing less than 10,000 pounds must be designed to withstand a 4-foot free fall drop onto a flat unyielding horizontal surface, striking in a position for which maximum damage is expected, without permitting escape of fuel or dangerously high radiation levels to occur.
- (b) In shipment tie-down devices must be capable of withstanding a static force, applied to the center-of-gravity of the package with the following components:

Longitudinal	- 10 W
Vertical	- 2 W
Lateral	- 5 W

where W represents the package weight. The above requirements are design requirements for safety and may be too severe for design loads after which the generator is expected to operate.

Uniform methods for environmental testing of equipment to withstand natural and induced environments peculiar to military operations are set forth in MIL-STD-810A (USAF), Reference (13). The standards include shock tests (Method 516.1). For basic design the shock pulse shown in Figure B-12 should be duplicated. Drop tests may also be employed. No specification for flat drops for 1000-pound packages is stated in MIL-STD-810A; however, edge and corner drops from a height of two to three feet are recommended. Acceleration levels for such shocks may be estimated using Figure B-11 if the characteristics of the package are known. Recommended shock and vibration test procedures for radioactive packages for transport are stated by the United Kingdom Atomic Energy Authority, Reference (14). Table A-6 gives the suggested impact test for packages weighing over 1000 pounds.

Shock and Vibration in Road Vehicles

In road vehicles the vibration environment varies and is determined by the specific vehicle, its loading, its speed and the surface on which it travels. The sources responsible for the origination of shock and vibration can be both external and internal.

Road and ground surface irregularities are the main external source of disturbance. It can be shown, Reference (15), that the acceleration of the tire surface is proportional to the square of the speed of the vehicle; therefore, impact loads due to road irregularity increase rapidly with speed. Wind pulsation, braking, and ramp backing are other external sources of vibration and shock.

The internal sources of vibration in a vehicle are the engine, the drive mechanism, and the tire-wheel assembly. In internal-combustion engines imbalance gives rise to a forcing frequency proportional to multiples of the engine speed. Whipping of the drive shaft may also be caused by imbalance causing excitation with a frequency proportional to the engine speed multiplied by the transmission ratio. The analysis of vibration resulting from the tire-wheel assembly is very complex because the road surface induces a combination of rotational and translational oscillation of the axles under normal conditions, and wheel "shimmy" under abnormal conditions. Therefore, the effects of both internal and external sources on the cargo are best evaluated experimentally.

A device weighing more than 1000 pounds will normally be transported on a large truck or tractor-trailer combination. Tables A-7 and A-8 and Figure B-13 present data for Military Transport Vehicles. The data for Table A-8 were obtained for operation of the vehicles over rough roads at speeds above which damage to the vehicle and endangerment to the driver were probable. The same is true for Figures B-14 and B-15 which present vibration conditions for both military and civilian vehicles. Average vertical accelerations for normal operating conditions are presented in Figure B-16. Presumably lateral accelerations are of lesser magnitude. Little data are available regarding random deceleration of vehicles; however, a value of 0.6 g is used in trailer design, Reference (16), for the longitudinal acceleration due to braking.

Adams and McKay, Reference (17), report tests in which a container was exposed to various road conditions in a 2-1/2-ton 6 x 6 military truck at speeds from one to 25 mph. The container was not secured to the truck bed but was tied to the sides of the truck to prevent excessive sliding. Figure B-17 presents data for the acceleration and shocks received at a point on the inside

bottom of the container while traveling over a rough road at 20 to 25 mph. These data were included to indicate the nature of the shock spectrum and pulse shape. The peak acceleration was 2.3 g's and the severest shock recorded was a 9.2 g pulse with a rise time of 12 msec and a duration of 20 msec. U.S. Military test specifications and recommended British test plans for packages transported by road vehicles are presented in the following paragraph.

Figures B-18 and B-19 outline vibration tests for equipment transported in ground vehicles and by common carrier. Examination of the road vehicle environmental data reveals that vibratory accelerations are less than 4 g's in amplitude with a frequency range from 1 to 300 cps. Shock peaks to 9 g's are possible but infrequent. Considering the above factors, designing to meet the common carrier test specifications appears most reasonable, since 5 g acceleration levels will rarely be exceeded in road transport.

Shock and Vibration in Rail Vehicles

The origin of shock and vibration in rail vehicles is primarily from external sources. The principal source of vibration is the impact of the steel wheels on the rails. This induced vibration is in both the vertical and lateral direction. The former is caused by discontinuities in the track and the latter by the taper of the "wheel tread" which causes hunting of the wheel on the rail. Figure B-20 presents data regarding the fundamental forcing frequencies encountered in rail transport, Reference (18).

The major sources of shock in railroad cars are the impacts due to switching. Take-up of slack in the train upon starting, "pull-outs" at tops of grades, and "run-ins" at bottoms of grades are other sources of shock loading. Severe impacts generally occur infrequently during the period of shipment and seldom exceed an acceleration of 1 g. A comparison of impact speeds in yard and road service is given in Figure B-21.

The environmental data for rail car vibration and shock categorizes continuous road service and switching operation conditions. The intensity of vertical accelerations for heavy cars is approximately half that of longitudinal accelerations. Additionally, lateral loads are generally less than vertical loads. The vibration environment for continuous road service is presented in Figures B-22 and B-23. The envelope in Figure B-23 represents values of acceleration which are rarely exceeded while the zones of Figure B-22, Reference (19), depict typical conditions.

The average coupling speed for rail car impacts is about 7.5 mph, as illustrated by Figure B-24. Experimental data for associated longitudinal and vertical accelerations are presented in Figures B-25 and B-26. Several points lie outside the envelope curve of Figure B-25. The occurrence of such points is rare and often represents a single peak of a shock transient as determined by peak measuring accelerometers. This is illustrated by the data cited immediately below.

Tests reported by Cross and McWhirter, Reference (20), report shock spectrums for various cargo positions under conditions of rail car impact varying from 3.4 to 10 mph. Figure B-27 shows a plan view of a Class A, ATSF automobile car without shock absorbers, and with a spring-type draft gear. The data for positions A and B are presented in Figures B-28 and B-29 for a 10 mph impact. These data indicate not only the magnitude of typical shock peaks, but also the shape and duration of the pulses.

Switching shocks provide the most severe acceleration levels for rail cargo. For 7.5 mph impacts Figures B-25 and B-26 show that associated shock peaks up to 20 g's are possible for cargo spaces. One common method for reducing the shock transmitted to rail cargo is to employ a cushioning device or floating apparatus.

Such floating decks are quite effective as illustrated by the data shown in Figure B-30. Maximum vibratory and shock accelerations in road service rarely exceed 3 g's. The predominant frequency range is 2 to 10 cps. If package cushioning is employed, (refer to Figure B-30) the common carrier test specification, Figure B-19, again appears adequate for rail transported cargo.

Shock and Vibration of Aircraft

Aircraft vibration is produced by a complex interaction of localized and distributed forces interacting with the vehicle structure. The main localized forcing functions are ground reactions due to taxi, take-off and landing, and reactions on the structure at the mounting point of the propulsion system. Distributed forces over the surface of flight vehicles are caused by fluctuating air pressures. These take the form of acoustic waves and aerodynamic forces in the form of gusts.

Atmospheric turbulence in the form of gusts changes the vehicle's apparent angle of attack with respect to the air resulting in pressure changes normal to the vehicle's surface. Goland and Luke, Reference (21), present calculations of acceleration time histories of vibrations for a typical fighter aircraft encountering a sharp gust at high speed. The results reveal a peak acceleration of the plane's center of gravity of less than 3.5 g's. It can be expected that a cargo transport craft would experience a significantly less severe reaction under the same dynamic loading conditions.

Vibratory acceleration at the airplane center of gravity normally does not exceed 0.2 g's during taxi over smooth surfaces and 0.5 g's during taxi over rough surfaces, take-off or hard-braking, Reference (22). Characteristic frequencies are 1 to 2.5 cps. The landing operation includes vehicle descent and ground touchdown. A maximum value of descent speed for cargo airplanes of about 7 ft/sec, compared to a mean of 1.4 ft/sec observed for

187 landings by Saunders and Piper, Reference (23). Associated with these descent rates were peak acceleration values at the center-of-gravity of 0.6 g's and 0.1 g's, respectively. Landing impacts normally generate the highest acceleration levels in aircraft. Vertical acceleration measurements during landing at various positions along the fuselage of several large aircraft are shown in Figure B-31. The fuselage station is the distance from the main landing gear. The frequency range for these vibrations is 2.5 to 7 cps.

In-flight vibrations encountered in jet-engine and reciprocating-engine aircraft are shown in Figures B-32 and B-33, Reference 24. Displacements in the vertical, lateral, and longitudinal directions are combined. The acceleration at any point enclosed by the envelope may be calculated by the following equation if a harmonic vibration is assumed.

$$g = X_{pp} f^2 (0.052) \quad (16)$$

where

- g = acceleration, g
- X_{pp} = peak-to-peak displacement, inch
- f = vibration frequency, cps.

It is seen from the figures that, with the exception of the empennage and propeller plane regions the maximum vibratory accelerations do not exceed 3.5 g's for most aircraft. Military vibration and constant acceleration tests, Reference (13), for airborne equipment are presented in Figure B-35 and Table A-9. A 5 g vibratory acceleration level for testing is applicable to packages which are not located in the tail or engine regions.

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TABLE A-1. VIBRATION AMPLITUDES - MIL-STD-167
(Reference 2)

Frequency Range, cps	Table Amplitude, 0-peak inches	Acceleration, 0-peak g's
5-15	0.030 ± 0.006	0.077 - 0.69
16-25	0.020 ± 0.004	0.52 - 1.27
26-33	0.010 ± 0.002	0.69 - 1.12

TABLE A-2. WIND AND SEA SCALE FOR FULLY ARISEN SEA (Reference 5)

WIND AND SEA SCALE FOR FULLY ARISEN SEA													
WIND							SEA						
DESCRIPTION	BEAUFORT	RANGE (KNOTS)	WIND VELOCITY (KNOTS)	AVERAGE	SIGNIFICANT	AVERAGE/HIGHEST	WAVE HEIGHT FEET	SIGNIFICANT PERIODS (SECONDS)	T _{max} (PERIOD OF MAXIMUM ENERGY OF SPECTRUM)	T̄ (AVERAGE PERIOD*)	L̄ (AVERAGE WAVE LENGTH)	MINIMUM FETCH (NAUTICAL MILES)	MINIMUM DURATION (HOURS)
							up to 12 sec						
CALM	0	0-1	0	0	0	0	—	—	—	—	—	—	—
LIGHT AIR	1	1-3	2	0.05	0.08	0.10	up to 12 sec	0.7	0.5	10 in	5	18 min	
LIGHT BREEZE	2	4-6	5	0.18	0.29	0.37	0.4-2.8	2.0	1.4	6.7 ft	8	39 min	
GENTLE BREEZE	3	7-10	8.5	0.6	1.0	1.2	0.8-5.0	3.4	2.4	20	9.8	1.7 hrs	
			10	0.88	1.4	1.8	1.0-6.0	4	2.9	27	10	2.4	
MODERATE BREEZE	4	11-16	12	1.4	2.2	2.8	1.0-7.0	4.8	3.4	40	18	3.8	
			13.5	1.8	2.9	3.7	1.4-7.6	5.4	3.9	52	24	4.8	
			14	2.0	3.3	4.2	1.5-7.8	5.6	4.0	59	28	5.2	
			16	2.9	4.6	5.8	2.0-8.8	6.5	4.6	71	40	6.6	
FRESH BREEZE	5	17-21	18	3.8	6.1	7.8	2.5-10.0	7.2	5.1	90	55	8.3	
			19	4.3	6.9	8.7	2.8-10.6	7.7	5.4	99	65	9.2	
			20	5.0	8.0	10	3.0-11.1	8.1	5.7	111	75	10	
STRONG BREEZE	6	22-27	22	6.4	10	13	3.4-12.2	8.9	6.3	134	100	12	
			24	7.9	12	16	3.7-13.5	9.7	6.8	160	130	14	
			24.5	8.2	13	17	3.8-13.6	9.9	7.0	164	140	15	
			26	9.6	15	20	4.0-14.5	10.5	7.4	188	180	17	
MODERATE GALE	7	28-33	28	11	18	23	4.5-15.5	11.3	7.9	212	230	20	
			30	14	22	28	4.7-16.7	12.1	8.6	250	280	23	
			30.5	14	23	29	4.8-17.0	12.4	8.7	258	290	24	
			32	16	26	33	5.0-17.5	12.9	9.1	285	340	27	
FRESH GALE	8	34-40	34	19	30	38	5.5-18.5	13.6	9.7	322	420	30	
			36	21	35	44	5.8-19.7	14.5	10.3	363	500	34	
			37	23	37	46.7	6-20.5	14.9	10.5	376	530	37	
			38	25	40	50	6.2-20.8	15.4	10.7	392	600	38	
			40	28	45	58	6.5-21.7	16.1	11.4	444	710	42	
STRONG GALE	9	41-47	42	31	50	64	7-23	17.0	12.0	492	830	47	
			44	36	58	73	7-24.2	17.7	12.5	534	960	52	
			46	40	64	81	7-25	18.6	13.1	590	1110	57	
WHOLE GALE *	10	48-55	48	44	71	90	7.5-26	19.4	13.8	650	1250	63	
			50	49	78	99	7.5-27	20.2	14.3	700	1420	69	
			51.5	52	83	106	8-28.2	20.8	14.7	736	1560	73	
			52	54	87	110	8-28.5	21.0	14.8	750	1610	75	
			54	59	95	121	8-29.5	21.8	15.4	810	1800	81	
STORM *	11	56-63	56	64	103	130	8.5-31	22.6	16.3	910	2100	88	
			59.5	73	116	148	10-32	24	17.0	985	2500	101	
HURRICANE *	12	>64	>64	>80 ^b	>128 ^b	>164 ^b	10-(35)	(26)	(18)	~	~	~	

* For hurricane winds (and often whole gale and storm winds) required durations and fetches are rarely attained. Seas are therefore not fully arisen.

a) A heavy box around this value means that the values tabulated are at the center of the Beaufort range.

b) For such high winds, the seas are confused. The wave crests blow off, and the water and the air mix.

TABLE A-3. SHIP MOTION AT STERN

Wave Height, feet (Crest to Trough)	Frequency, cps	Displacement, feet	Acceleration, g's
1.6	0.33	$\pm$ 0.092	$\pm$ 0.01
3.3	0.25	$\pm$ 0.95	$\pm$ 0.056
5.0 (1)	0.215	$\pm$ 3.0	$\pm$ 0.17
7.3	0.18	$\pm$ 6.5	$\pm$ 0.28
10.2	0.165	$\pm$ 9.8	$\pm$ 0.33
14.0	0.13	$\pm$ 11.9	$\pm$ 0.24
20.0 (High Sea)	0.11	$\pm$ 16.6	$\pm$ 0.25

(1) Maximum sea considered for generator implantment.

TABLE A-4. IMPACT PARAMETERS FOR VERTICAL DROP

	20 Watt Generator				10 Watt Generator			
	20	15	10	5	20	15	10	5
Drop Height, feet	20	15	10	5	20	15	10	5
V_o , in./sec	430	373	304	215	430	373	304	215
A, g's	120	96	72	43	123	99	73	44
V, in./sec	291	253	206	145	292	254	206	145
t_1 , msec	4.4	4.7	5.2	6.2	4.3	4.6	5.1	6.0
X_1 , inch	1.68	1.56	1.40	1.18	1.63	1.59	1.36	1.15

TABLE A-5. IMPACT PARAMETERS FOR LATERAL WAVE IMPACT

	20 Watt Generator				10 Watt Generator			
Wave Height, feet	11	8	5	2	11	8	5	2
V_o , in./sec	430	373	304	215	430	373	304	215
A, g's	162	133	101	64	224	185	141	90
V, in./sec	324	281	227	162	321	280	226	160
t_1 , msec	2.1	2.2	2.4	2.6	1.5	1.6	1.7	1.9
X_1 , inch	0.81	0.73	0.65	0.51	0.59	0.53	0.47	0.37

TABLE A-6. IMPACT TESTS
(Reference 14)

(One Sample)

Test	Package Shape	
	Cube	Cylinder
5 ft/sec Impact	A	C
1-ft Sloping Drop	B	B
Rolling	-	D

- Notes: A) Impact - The package to stand on its base and strike with each end in turn against a 12-in.-wide steel beam having a maximum deflection at the center of 0.01 in. under a 50-ton static load.
- B) Sloping Drop - The package to stand on its base and each end in turn to be raised and released.
- C) Impact - The package to lie on the cylindrical surface and strike with each plane face in turn against a beam as in (A) above.
- D) Rolling - For a distance of 30 feet in both directions.

TABLE A-7. ACCELERATION AT CARGO IN MOTOR TRANSPORT VEHICLES
(2-1/4-TON TRUCK, M-104 TRAILER) IN BEACHING AND
ROAD TEST (Reference 25)

Type of Action	Maximum Acceleration, g			Vector Total	Frequency, cps
	Longitudinal	Lateral	Vertical		
Beaching	2.5	1.0	4.5	5.3	3.3
Ungraded Road (30 mph)	0.5	1.0	1.5	1.9	-
Graded Road (30 mph)	1.0	0.25	1.0	1.4	-

TABLE A-8. PREDOMINANT FREQUENCIES MEASURED IN THE CARGO SPACE OF
VARIOUS MILITARY TRANSPORT VEHICLES (Reference 26)

Type of Vehicle	Direction of Acceleration	Predominant Frequencies, cps		
		Body (Sprung Mass)	Vibration on Tires	Resonant Vibration of Structural Part of the Body
Truck (2-1/2 T)	Vertical	2-4	8-13	70-180
	Longitudinal	--	10-20	70-100
	Lateral	2	10-20	100-200
Truck (3/4 T)	Vertical	2-3	5-10	60-110
	Longitudinal	--	--	70-100
	Lateral	--	--	60- 70
Trailer (1 Ton)	Vertical	3-5	8-10	50-100
	Longitudinal	--	--	50-100
	Lateral	2	--	50-120
M-14 Trailer	Vertical	1-4	7-10	50- 70
	Longitudinal	3-4	8-10	200
	Lateral	2-4	--	--
M-1 Trailer (2 Ton)	Vertical	2.5-5	7.75-10.5	100-150

TABLE A-9. ACCELERATION TEST LEVELS FOR AIRCRAFT
(Reference 13)

	Fore	Aft	Up	Down	Lateral
Structural Test	9.0	3.0	4.5	13.5	6.0
Operational Test	6	2.0	3.0	9.0	4.0

FIGURE B-2. OCEAN WAVE HEIGHT AND AVERAGE LENGTH

FIGURE B-3. BARGE RESPONSE VERSUS SEA STATE

FIGURE B-4. SHOCK PULSE FOR VERTICAL DROP OF 20 WATT GENERATOR

B-5

FIGURE B-5. SHOCK PULSE FOR LATERAL WAVE IMPACT ON 20 WATT GENERATOR

B-6

FIGURE B-6. SHOCK PULSE FOR VERTICAL DROP OF 10 WATT GENERATOR

B-7

FIGURE B-7. SHOCK PULSE FOR LAIFRAI WAVE IMPACT OF 10 WATT GENERATOR

FIGURE B-8. RESONANT FREQUENCIES OF GENERATOR AND LINE

FIGURE B-9. MAXIMUM DYNAMIC IMPACT VELOCITY FOR GENERAL REPLACEMENT USING 5/8-INCH POLYPROPYLENE ROPE

FIGURE B-10. SHOCK ISOLATION MODEL

FIGURE B-11. MAXIMUM ACCELERATION OF DROPPED PACKAGES⁽¹²⁾

D = NOMINAL DURATION IN MILLISECONDS

A = SPECIFIED PEAK ACCELERATION IN GRAVITIES (g 's)

g = ACCELERATION OF GRAVITY ON EARTH 386 IN./SEC./SEC.

PROCEDURE	TEST	PEAK VALUE	NOMINAL DURATION
		<u>A</u>	<u>D</u>
I	BASIC DESIGN	20 g	10 ms
IV	CRASH SAFETY	40 g	10 ms
V	HIGH INTENSITY	100 g	6 ms

FIGURE B-12. SAWTOOTH SHOCK PULSE CONFIGURATION⁽¹³⁾

FIGURE B-13. VIBRATION AND SHOCK IN MILITARY ROAD VEHICLES

B-14

FIGURE B-14. VIBRATION CONDITIONS IN THE CARGO SPACES OF TRUCKS AND TRAILERS DURING ROUGH HANDLING⁽²⁶⁾

B-15

FIGURE B-15. VERTICAL ACCELERATION OF CARGO SPACE OF TRUCKS IN HIGHWAY TRAVEL UNDER ROUGH DRIVING CONDITIONS⁽¹⁶⁾

FIGURE B-16. AVERAGE ACCELERATION MEASURED UNDER NORMAL OPERATIONAL CONDITIONS FOR COMMERCIAL AND MILITARY VEHICLES⁽²⁶⁾

FIGURE B-17. SHOCK SPECTRUM AND ACCELERATION-TIME CURVE FOR SHIPPING CRATE ON 2-1/2 TON 6 x 6 MILITARY TRUCK, ROUGH UNGRADED ROAD, 20 TO 25 MPH(17)

FIGURE B-18. VIBRATION TEST CURVES, EQUIPMENT INSTALLED IN GROUND VEHICLES⁽¹³⁾

FIGURE B-19. VIBRATION TEST CURVES, EQUIPMENT TRANSPORTED BY COMMON CARRIER⁽¹³⁾

FIGURE B-20. FORCING FREQUENCIES OF RAIL JOINTS AND WHEEL UNBALANCE AS A FUNCTION OF SPEED OF CAR⁽¹⁸⁾

FIGURE B-21. DISTRIBUTION OF IMPACT SPEED OF 1,568 CARS IN YARD AND ROAD SERVICE DURING A PERIOD OF 3 MONTHS⁽²⁹⁾

FIGURE B-22. RANGES OF PREDOMINANT FREQUENCIES AND CORRESPONDING MAXIMUM AMPLITUDES IN RAILROAD FREIGHT CARS⁽¹⁹⁾

FIGURE B-23. VERTICAL ACCELERATION OF RAIL CARS OR LADING IN RAIL SERVICE (16)

FIGURE B-24. SPEED DISTRIBUTION OF RAIL CAR IMPACTS
IN YARD SWITCHING⁽¹⁶⁾

FIGURE B-25. LONGITUDINAL ACCELERATION OF RAIL CARS IN YARD SWITCHING⁽¹⁶⁾

FIGURE B-26. VERTICAL ACCELERATION OF RAIL CARS
IN YARD SWITCHING⁽¹⁶⁾

FIGURE B-27. PLAN VIEW OF INSTRUMENTED CAR⁽²⁰⁾

Longitudinal Direction

Vertical Direction

FIGURE B-28. ACCELERATION SPECTRUM, (20) 10 MPH IMPACT - POSITION A

Longitudinal Direction

Vertical Direction

B-29

FIGURE B-29. ACCELERATION SPECTRUM, ⁽²¹⁾ 10 MPH IMPACT - POSITION B

FIGURE B-30. MAXIMUM VERTICAL ACCELERATION EXPERIENCED BY SOLID BLOCKED AND CONTROLLED FLOATING LADING AS A FUNCTION OF CAR SPEED AT IMPACT (30)

FIGURE B-31. MAXIMUM ACCELERATIONS AT FUSELAGE STATIONS FOR SEVERAL LARGE AIRCRAFT⁽³¹⁾

Vibration envelopes for fuselages of jet-engine-powered flight vehicles, based on measurements in six aircraft. Solid envelopes apply to locations aft of jet engines; dotted envelopes apply to locations forward of jet engines.

FIGURE B-32. VIBRATION ENVELOPES FOR FUSELAGES OF JET-ENGINE-POWERED FLIGHT VEHICLES(24)

FIGURE B-33. VIBRATION ENVELOPES FOR STEADY-STATE VIBRATION OF FUSELAGES AND EMPENNAGES OF RECIPROCATING-ENGINE AIRCRAFT, BASED ON MEASUREMENTS ON 12 AIRCRAFT⁽²⁴⁾

Curve Aircraft Equipment Location

- A Equipment installed on vibration isolated panels and racks when the panel or rack is not available for test and when specified in table 514-1.
- B Equipment in forward half of fuselage or equipment in wing areas of aircraft with engines at rear of fuselage.
- C Equipment in rear half of fuselage or equipment in wing areas of aircraft with wing mounted engines.
- D Equipment located in the engine compartment or pylon.
- E Equipment mounted directly on aircraft engine.

Air Launched Missile (captive phase)

- B Equipment in missile attached to wing of aircraft with engine in rear of fuselage.
- C Equipment in missile carried in aircraft fuselage or attached to wing in aircraft with wing mounted engines.

FIGURE B-34. VIBRATION TEST CURVES, AIRCRAFT EQUIPMENT⁽¹³⁾

APPENDIX V
ANALYSIS OF WATER AND SHIP IMPACT VELOCITIES DUE TO SHIP ROLLING

APPENDIX V

ANALYSIS OF WATER AND SHIP IMPACT VELOCITIES DUE TO SHIP ROLLING

OBJECTIVES - Determine the effects of ship rolling on implantment of the SNAP-21 system while suspended from a boom. Specifically, this requires the following:

- (1) The clearance between the suspended weight and the side of the ship, or the conditions required to maintain a clearance.
- (2) The maximum impact velocity of the system with the water due to ship rolling.

ASSUMPTIONS - Roll motion of the ship is assumed to be harmonic and of a specified amplitude and frequency. This rolling will result in horizontal and vertical components of velocity for a specific point on a boom extended over the side of the ship. The amplitude of roll considered is between 0 and 25°, and frequency, or period of roll, is between 5 and 20 seconds.

ANALYSIS - If roll motion is harmonic, boom movement can be described by the following

$$\theta = \theta_o \sin wt$$

where θ_o is amplitude of roll.

The velocity of a point "A" on the boom is then given as follows:

$$V_T = R\dot{\theta} = R\omega\theta_o \cos wt$$

where R is distance from roll axis to the end of the boom

w is the roll frequency $\left(\frac{2\pi}{t}\right)$ in radians per second

θ_o is the roll amplitude

t is the roll period in seconds.

The maximum velocity V_T is $Rw \theta_o$. Figure V-1 plots the relationship between these variables.

The velocity of point "A" can be broken into vertical and horizontal components and if the hanging weight follows the boom motion, it has the same horizontal and vertical velocities. Since the component velocities depend upon the angle ϕ , the maximum component velocities are plotted in Figures V-2 and V-3 as functions of R , θ_o , ϕ and w .

The assumption that the hanging weight follows the horizontal motion of the boom is based on the following:

- (a) Natural period of the weight, when treated as a pendulum, is very near the roll frequencies of the ship, and
- (b) It is assumed that the outward swing of the weight can be easily restricted by a line connecting the weight to the ship. This will prevent any amplification of the oscillation as the ship rolls.

Figure V-1. Maximum Boom Velocity

Figure V-2. Maximum Vertical Boom Velocity

Figure V-3. Maximum Horizontal Boom Velocity

The component velocities are the maximum impact velocities that can occur for a given configuration. These velocities are also the impact velocities for the system with the ship and with the water under particular circumstances.

In the vertical case, the boom must be passing downward through the point "A" at the same time the system enters the water - any other position of the boom results in a lower impact velocity.

Similarly, for maximum horizontal impact, the boom must be passing through "A" on the upswing and hit the ship's side at the same time.

Impact with the side can be avoided by extending the boom far enough over the side. The condition for no side impact is that the value of $R \cos (\phi + \theta)$ must be larger than $1/2$ the beam of the ship (Figure V-4).

TOTAL VERTICAL IMPACT VELOCITIES

In the analysis of roll velocities, two parameters, R and ϕ , were used. If the ship size and boom dimensions are known, values for R and ϕ can be found. Since these dimensions are variable, the best that can be done is to bound the parameters. The following size ranges are assumed for the ship and boom combination.

Figure V-4. Horizontal Distance from Roll Axis to Suspended Weight

$$R_{\max} = (x_{\max}^2 + y_{\max}^2)^{1/2} = (45^2 + 30^2)^{1/2} = 55'$$

$$R_{\min} = (x_{\min}^2 + y_{\min}^2)^{1/2} = (15^2 + 5^2)^{1/2} = 16'$$

Corresponding angles are:

$$\phi = \tan^{-1} \left(\frac{30}{45} \right) = \tan^{-1} (0.667) = 34^\circ$$

$$\phi = \tan^{-1} \left(\frac{5}{15} \right) = \tan^{-1} (0.333) = 18^\circ$$

The limiting values for vertical velocity are found from Figure V-2. Using a sea state condition defined by $\theta/T = 2$. The limits are as follows:

For $R = 55'$ and $\phi = 34^\circ$

$$\frac{V_y}{R} = 0.180$$

$$V_y = 0.18(55) = 9.9'/\text{sec}$$

For $R = 16'$ and $\phi = 18^\circ$

$$\frac{V_y}{R} = 0.207$$

$$V_y = 16(0.207) = 3.3'/\text{sec}$$

Similarly for the horizontal velocity (Figure V-3)

For $R = 55'$ and $\phi = 34^\circ$

$$\frac{V_x}{R} = 0.123 \qquad V_x = 0.123(55) = 6.75'/\text{sec}$$

For $R = 16'$ and $\phi = 18^\circ$

$$\frac{V_x}{R} = 0.675 \qquad V_x = 0.675(16) = 10.8'/\text{sec}$$

Consequently, for $\theta/T = 2$, the assumed dimensions result in a maximum vertical velocity between 3.3 and 9.9'/sec and a maximum horizontal velocity between 6.75 and 10.8'/sec.

The vertical wave velocity is found from data on wave height and frequency.

$$y = y_o \sin wt$$

$$V = Y_o w \cos wt \qquad w = 2\pi F$$

$$V_{\max} = Y_o w$$

<u>Wave Height</u>		<u>Frequency</u>		<u>Velocity</u>
Ft.	cps		Radian/sec	Ft/sec
2	0.310		1.95	3.90
3	0.265		1.67	5.01
4	0.235		1.48	5.92
5	0.215		1.35	6.75

Cable speed will vary between 2 and 10'/sec (assumed).

The resultant impact velocity is the sum of the roll velocity, the wave velocity and the cable speed.

Roll	3.3 to 9.9	(worst condition)
Wave	4.0 to 7.0	(worst condition)
Cable	2.0 to 10.0	

Total vertical velocity range = 9.3 to 26.9'/sec for a sea state of

$$\theta/T = 2$$

For a given water impact velocity, the corresponding payload acceleration in G's can be determined from Figure V-5.

Figure V-5. Maximum Payload Acceleration Due to Water Impact

APPENDIX VI

DYNAMIC ANALYSIS OF SNAP-21 10-WATT GENERATOR BY 3M COMPANY

- INITIAL STRESS CONDITIONS
- GENERATOR SHOCK ANALYSIS
- GENERATOR VIBRATION ANALYSIS
- EFFECT OF DYNAMIC LOAD ON GENERATOR
- APPENDIX
- REFERENCES

APPENDIX VI

DYNAMIC ANALYSIS OF SNAP-2, 10-WATT GENERATOR BY 3M COMPANY

Initial Stress Conditions in the Generator Case and Hot Frame

Noting that the spring force per leg is about 4 pounds, the total load P_o on the hot frame is about 400 pounds. The generator will also be subjected to 10 psig internal pressure.

Stresses in the hot frame are found from theory on circular plates.¹ Two sets of results will be obtained depending upon the type of edge constraint applied to the plate.

(1) Clamped edge

$$w = \frac{pR^4}{64D} \left(1 - \frac{r^2}{R^2} \right)^2$$

$$M = \frac{pR^2}{16} \left(- (1 + u) + (1 + 3u) \frac{r^2}{R^2} \right)$$

(2) Simply supported edge

$$w = \frac{pR^4}{64D} \left(\frac{5 + u}{1 + u} - \frac{6 + 2u}{1 + u} \frac{r^2}{R^2} + \frac{r^4}{R^4} \right)$$

$$M = \frac{3 + u}{16} p R^2 \left(1 - \frac{r^2}{R^2} \right)$$

The load p on the plate is derived from the pressure and the spring load.

$$p = 10 + \frac{400}{A} = 10 + \frac{400}{(4.310^2 - 4.300^2) \pi/4} = 38 \text{ psi}$$

The quantity D is the plate stiffness and is calculated as:

$$D = Et^3/12 (1 - u^2) = 28 \times 10^6 (0.25)^3/12 (1 - 0.09) = 39500 \text{ in-lb}$$

The maximum stress and deflection occur at the center of the plate.

(1) Clamped edge

$$w = 38 (2.15)^4/64 (39500) = 0.00032 \text{ in.}$$

$$M = 38 (2.15)^2 (0.70)/16 = 7.7 \text{ in-lb}$$

$$S = 6M/t^2 = 6 (7.7)/0.0625 = 740 \text{ psi}$$

(2) Simply supported edge

$$w = 38 (2.15)^4 (5.3)/64 (39500) (1.3) = 0.0013 \text{ in.}$$

$$M = 38 (2.15)^2 (3.3)/16 = 36.4 \text{ in-lb}$$

$$S = 6M/t^2 = 6 (36.4)/0.0625 = 3500 \text{ psi}$$

The actual constraint at the edge of the plate will be between the two cases. Therefore, maximum bending stress and deflection at the center of the plate will be between the respective values of 740 and 3500 psi and 0.00032 and 0.0013 inch. The bending moment at the edge of the plate can also be determined from the above relationships and will have a value between 0 and 7.7 in/lbs.

The membrane stresses in the case as a result of the internal pressure and the spring load are found by the following.¹

In the direction of the cylinder axis;

$$S_1 = F/A = 400/((4.310^2 - 4.300^2)\pi/4) = 400/0.0675 = 5940 \text{ psi}$$

$$S_2 = pr/2t = 10 (2.15)/2(0.005) = 2150 \text{ psi}$$

In the circumferential direction

$$S_3 = pr/t = 10 (2.15)/0.005 = 4300 \text{ psi}$$

In addition to the membrane stress, there are bending or secondary stresses set up in the cylindrical case. These are caused by bending of the hot frame and by the difference in expansion between the thick and thin portions of the case when subjected to internal pressure. This bending stress is found by considering part of the cylinder as a beam; the remainder being an elastic foundation. Theory for semi-infinite beams on elastic foundations is then applied.¹

Deflection is given by

$$y = \frac{2Bp}{k} F_4 (Bx)$$

where:

$$B = \sqrt{\frac{4}{r^2 t^2} \frac{3(1 - \nu^2)}{E}} = \frac{1.28}{\sqrt{rt}}$$

$$k = Et/r^2$$

$$F_4 = (Bx) = e^{-Bx} \cos (Bx)$$

The bending moment by

$$M = (P/B) F_2 (Bx)$$

where:

$$F_2(Bx) = e^{-Bx} \sin(Bx)$$

Bending stress is determined from the bending moment

$$S = 6M/t^2$$

The approximation of a semi-infinite beam to a finite beam is good if (Bx) is larger than 3. For values less than one, the beam can be considered as a rigid body. Examining two sections of the case and calculating values for B , the thicker portion of the case can be considered as rigid since Bx is about 1. The thin section should correspond to theory very well since Bx is about 25.

For the thick section:

$$B = 1.28 / \sqrt{(2.15)(0.03)} = 5.05 \quad Bx = 1.23$$

For the thin section:

$$B = 1.28 / \sqrt{(2.15)(0.005)} = 12.3 \quad Bx = 26.6$$

Since the lower part of the case can be assumed rigid, bending of the hot frame transmits a force to the thin part of the case. The moment is replaced by a force P' such that $P'x'$ is equal to the moment. The moment resulting from bending of the hot frame was previously calculated as a value between 0 and 7.7 in-lbs. The distance x' is equal to $2(0.186) + 0.25$ or 0.622 in. The force P' then varies between 0 and 12.4 lbs. The force P is then determined from the ratio of distances from the neutral plane of the hot frame.

The maximum deflection occurs at x equal to zero. At x_0 the function F_4 is equal to 1.0.

$$k = Et/r^2 = 28 \times 10^6 (0.005)/(2.15)^2 = 30200$$

$$y = 2BP F_4 (Bx)/k = 2 (12.3) (10.1) (1.0)/30200 = 0.0062 \text{ in.}$$

The maximum bending moment occurs at a value of Bx equal to 0.80 or x equal to 0.80/12.3 or 0.065 in. $F_2 (0.80)$ equals 0.322.

$$M = P F_2 (Bx)/B = 10.1 (0.322)/12.3 = 0.198 \text{ in-lbs.}$$

The corresponding stress:

$$S = 6M/t^2 = 6 (0.198)/(0.005)^2 = 47,600 \text{ psi}$$

It should be noted that this is the maximum bending stress that can occur. Since the bending moment is a value between 0 and 7.7, the bending stress it generates will be between 0 and 47,600 psi. The fillet at the point of thickness change will lower this value as will deflection of any part of the case which was assumed rigid. Since the nature of the bending is indeterminate, the only conclusion to be drawn is that bending of the hot frame can result in high stress concentrations. Since these are local in character, localized yielding can also take place which will change the stress distribution.

When internal pressure is applied to the generator, the thicker part will deform less than the thin part. A discontinuity results which again generates a bending stress that can be calculated as the previous case.

$$\Delta r_1 = S_t/Et = pr^2/Et = 10 (2.14)^2/0.03 (28 \times 10^6)$$

$$\Delta r_1 = 54 \times 10^{-6} \text{ in.}$$

$$\Delta r_2 = S_t/Et = 3900 (2.15)/28 \times 10^6 = 0.000298 \text{ in.}$$

$$\Delta r_2 = \Delta r_1 = 0.000298 - 0.000054 = 0.000244 \text{ in.}$$

To account for the discontinuity, a ring load P is applied to the case and is determined from the deflection equation.

$$y = 2BP F_4(Bx)/k = 2BP/k \Big|_{x=0}$$

$$P = yk/2B = 2.44 \times 10^{-4} \times 3.02 \times 10^4/2 (12.3) = 0.299 \text{ lbs.}$$

The moment is then determined;

$$M = P F_2(Bx)/B = (P/B) (0.322) \Big|_{\substack{Bx = 0.80 \\ x = 0.065 \text{ in.}}}$$

$$M = 0.299/12.3 = 0.0243 \text{ in/lbs.}$$

The corresponding bending stress is:

$$S = 6M/t^2 = 0.1458/24 \times 10^{-6} = 5840 \text{ psi}$$

Again it will be noted that the fillet and any deflection of the rigid part of the case will lower this value.

Summarizing the initial stresses present in the generator case and hot frame:

The hot frame is expected to experience a bending stress of 3500 psi at most. Since this is fairly low in comparison to stresses in the outer case, it is not considered as a critical area.

The generator case will be subjected to membrane stresses which will vary over the length of the case due to the variation in thickness. These will be fairly constant and distributed over the entire case. In addition to the membrane stresses, there will be bending stresses present of an unknown magnitude and of a localized nature. The following figure and table summarize these stresses.

MEMBRANE STRESSES IN THIN SECTION		
DIRECTION	FORCE	STRESS
AXIAL	SPRING	5940 psi
AXIAL	PRESSURE	2150 psi
CIRCUMFERENTIAL	PRESSURE	4300 psi
BENDING STRESS AT POINT A		
AXIAL	SPRING	0 to 23,800 psi
AXIAL	PRESSURE	5480 psi

Generator Shock Analysis

Consideration is first given to a shock applied vertically to the generator as shown in the accompanying figure. It is assumed that the cold frame, hot frame and case are rigid. There are two effects the shock can generate: 1) relative displacement between the parts can occur and 2) the hot frame can experience dynamic loading which either increases or decreases the already existing load.

Any displacement that occurs between the rigid foundation and the internal parts depends upon the magnitude of the shock, the spring constant, the mass of the part involved and any damping effects of the insulation material. There are several modes of displacement possible:

- (1) If the packing material has no damping effect, motion occurs at point A.
- (2) If the packing has an infinite damping effect, motion results at B.
- (3) If the material is of an intermediate magnitude with respect to damping, motion could result at points C, D and E in addition to that at A and B.

Four of these points are analyzed. The transient response of a spring mass system with zero damping subjected to a sawtooth shock pulse is derived in the appendix as follows.

$$x_{tr} = \frac{\ddot{u}(t)}{k} \left(\frac{t}{T} - \frac{\sin(w_n t)}{w_n T} \right) \text{ For } 0 < t < T$$

$$x_{tr} = \frac{\ddot{u}(t)}{k} \left(-\frac{\sin(w_n t)}{w_n T} + \frac{\sin(w_n(t-T))}{w_n T} + \cos(w_n(t-T)) \right) \text{ for } t > T$$

where:

$\ddot{u}(t)$ is the acceleration of the foundation

T is the pulse time

w_n is the natural frequency of the system

Case 1 The packing material has no effect on the parts; corresponds to displacement at A.

$$k_{eq} = 96 \times 7.2 = 690 \text{ lbs/in.}$$

$$W = 48 (0.0092 + 0.0129 + 0.0006 + 0.00187) + 0.120 = 1.30 \text{ lbs.}$$

$$w_n = \sqrt{k/m} = \sqrt{kg/W} = 453 \text{ r/s} = 72 \text{ cps}$$

$$\ddot{u}(t)/k = 3W/k = 0.00565 \text{ in.}$$

Case 2 Packing material is assumed to hold the insulator in place. This results in a change of the mass and corresponds to displacement at C.

$$k_{eq} = 96 \times 7.2 = 690 \text{ lbs/in.}$$

$$W = 48 (0.0092 + 0.0129 + 0.0006 + 0.00187) = 1.18 \text{ lbs.}$$

$$w_n = \sqrt{k/m} = \sqrt{kg/W} = 476 \text{ r/s} = 76 \text{ cps}$$

$$\ddot{u}(t)/k = 3W/k = 0.00513 \text{ in.}$$

Case 3 Packing material is assumed to hold the buttons and electrodes as well as the insulator in place, corresponds to motion at point D.

$$k_{eq} = 2 \times 7.2 = 14.4 \text{ lbs/in.}$$

$$W = 0.0092 + 0.0129 = 0.0221 \text{ lbs.}$$

$$w_n = \sqrt{k/m} = \sqrt{kg/W} = 502 \text{ r/s} = 80 \text{ cps}$$

$$\ddot{m}(t)/k = 3W/k = 0.0046 \text{ in.}$$

Case 4 The only part assumed able to move is the follower; corresponds to motion at point B.

$$k_{eq} = 7.2 \text{ lbs/in.}$$

$$W = 0.0046 \text{ lb.}$$

$$w_n = \sqrt{kg/W} = 778 \text{ r/s} = 124 \text{ cps}$$

$$\ddot{m}(t)/k = 3W/k = 0.00192 \text{ in.}$$

The response for all four cases is plotted in the following figure.

Since the transient response is independent of initial conditions², this is the relative displacement that occurs between the parts. The effect of displacement is twofold. 1) packing material could sift between the parts, and 2) unseating of the parts could result. The following table shows the parts under examination, the depth of seating, the maximum amplitude for a 3 g shock and the acceleration required to give an amplitude equal to the minimum seating depth. The comment added here is that no consideration of damping has been made. With damping in the system, the amplitude is attenuated. It is also noted here that displacement can only occur if the shock is applied downward. If applied upward, displacement cannot occur because of the rigid hot frame.

PART	SEATING	MAXIMUM AMPLITUDE-3 g	LIMITING ACCELERATION
Insulator Hot Frame	$\frac{0.026}{0.032}$	0.00565 in.	14 g
Electrode Insulator	$\frac{0.020}{0.023}$	0.00510 in.	11.7 g
Leg Button Follower	$\frac{0.011}{0.019}$	0.0046 in.	7.2 g
Leg	$\frac{0.060}{0.063}$	0.00192 in.	94.8 g

Since the case is thin in cross section, the assumption of a rigid case may not apply. Defining a spring constant by Hooke's law, the case will consist of a series of springs.

$$F/A = Ee/L \quad \text{or} \quad F/e = AE/L = k$$

$$k_1 = (\pi/4) (4.42^2 - 4.30^2) 28 \times 10^6 / 0.35 = 72 \times 10^6$$

$$k_2 = (\pi/4) (4.31^2 - 4.30^2) 28 \times 10^6 / 1.81 = 1.05 \times 10^6$$

$$k_3 = (\pi/4) (4.35^2 - 4.30^2) 28 \times 10^6 / 0.70 = 13.5 \times 10^6$$

$$1/k_{eq} = 1/k_1 + 1/k_2 + 1/k_3$$

$$1/k_{eq} = 1/10^6 (0.0139 + 0.957 + 0.074) = 1/10^6 (1.045)$$

$$k_{eq} = 0.96 \times 10^6 \approx 10^6 \text{ in-lbs.}$$

Using the weight of the hot frame (1.0 lb) the natural frequency of the case in the vertical direction can be calculated.

$$w_n = \sqrt{k/m} = \sqrt{kg/W} = 19,700 \text{ r/s} = 3130 \text{ cps}$$

With a supported weight of 2.3 lbs., the deflection of the case for a 3 g acceleration is 6.9×10^{-6} inches. This is much smaller than the response amplitudes obtained for the internal parts and the assumption of rigidity should have little effect for small accelerations.

The situation is now investigated where the shock is applied horizontally to the generator. The rigidity of the case acting as a cantilever is first analyzed³.

The weight supported by the case is equal to all the internal parts and the hot frame (2.3 lbs.). The deflection for 3 g acceleration is $6.9/6.4 \times 10^5$ which equals 1.08×10^{-5} inches. The bending stress in the thinnest section is as follows.

$$S = Mc/I = PLc/I = 6.9 (2.16)/0.0675 = 221 \text{ psi}$$

The effect of the horizontal shock on the internal parts is considered next. Due to the small clearances between the follower and the cold frame, the couple is assumed to rotate on its spherical cap. The force causing the rotation is inertial and the force opposing is frictional. To be precise, the friction in the spherical cap should be included, but for first calculations it is neglected.

The first case considered is the frictional force between the hot frame and insulator. The centers of gravity are approximated and moments about point 0 are calculated for 3 g maximum acceleration.

Leg moment $3 (0.0129) (0.756/2) (96) = 1.41 \text{ in-lbs.}$

Button moment $3 (0.0006) (0.766) (48) = 0.133 \text{ in-lbs.}$

Electrode moment $3 (0.00187) (0.791) (48) = 0.213 \text{ in-lbs.}$

Insulator moment $0.849 (0.116) = 0.296 \text{ in-lbs.}$

$$F_k (0.907) = M_o = 2.052$$

$$F_k = 2.052/0.907 = 2.26 \text{ lbs.}$$

Since the normal force is 384 lbs. and the friction coefficient is 0.2 (TPM 169)⁴, the friction force present is 77 lbs. This is more than adequate for a 3 g acceleration. The system can be subjected to about 100 g's before the friction force is exceeded.

$$(77/2.26) \times 3 = 102 \text{ g}$$

If the interface between the electrode and insulator has the same coefficient of 0.2, the friction force is $0.2 \times N$, where N is the normal force. Since two springs are involved $N = 2 kx = 2 (7.2) (0.551) = 7.95 \text{ lbs.}$ and the friction force is $0.2 (7.95) = 1.59 \text{ lbs.}$ Using the same procedure used on the first case, the required frictional force is 0.0462 lbs. The system is capable of taking $(1.59/0.0462) 3 = 103 \text{ g's.}$

Assuming a coefficient of 0.15 (steel on steel) for the button-electrode interface, the friction present is $0.15 (7.95) = 1.19$ lbs. Summing the moments as in the previous cases results in a required friction force of 0.042 lb. This system is capable of handling $(1.19/0.042) \times 3 = 85$ g's.

Unless a very small coefficient results between the leg and the button, the same order of magnitude friction force and capability are involved. It is noted that when the parts are heated, some degree of diffusion bonding of the surfaces can occur and under such circumstances, the frictional force can not be described by a coefficient of friction. On the basis of friction coefficients, the generator can take 85-100 g's before slipping of the parts will occur.

Generator Vibration Analysis

The vibratory input to the generator as defined in MIL-STD-810A is between respective amplitudes of 0.018 in. and 0.50 in., acceleration limits of 5 g's and 1.3 g and frequencies of 50 and 5 cps.

A determination of the natural frequencies of the generator and its components is required to see if any resonances with the input frequencies occur. Natural frequencies of the internal parts and the case have been calculated in the section on shock. Frequencies of 72, 76, 80 and 124 cps have been found for the masses in contact with the follower spring. The case has a natural frequency of 3130 cps in the vertical direction and 1650 cps as a cantilever.

$$w_n = \sqrt{k_{eq}/m} = \sqrt{640,000 (386)/2.3} = 10350 \text{ r/s} = 1650 \text{ cps}$$

The hot frame plate has a natural frequency given by the following³.

$$w_n = B \sqrt{\frac{Et^2}{R^4 (1 - u^2)}} = 5.9 \sqrt{\frac{28 \times 10^6 (0.25)^2}{0.29 (2.15)^4 (0.91)}} = 3290 \text{ r/s}$$

$$w_n = 525 \text{ cps}$$

The frequency of 525 cps as well as those for the case are well above the 50 cps maximum input. The internal parts however, are only slightly above 50 cps. Using the steady state response equation for an undamped system, the effect of near resonance can be examined⁵.

$$A = x/x_0 = 1/(1-r^2) = 1/(1-w^2/w_n^2)$$

where:

A is the magnification factor for zero damping

$$\text{For 72 cps} \quad A = 1/1-(72/50)^2 = 1/1.07 = 0.935$$

$$\text{For 76 cps} \quad A = 1/1-(76/50)^2 = 1/1.31 = 0.765$$

$$\text{For 80 cps} \quad A = 1/1-(80/50)^2 = 1/1.56 = 0.641$$

$$\text{For 124 cps} \quad A = 1/1-(124/50)^2 = 1/5.15 = 0.194$$

It is seen no amplification results (A is less than 1). For 50 cps the acceleration input is 5 g. The amplitudes developed for each of the above cases is determined as follows:

$$x_0 = F/k = \ddot{u}/k = (W/g)(5 g)/k = 5W/k$$

$$\text{For 72 cps, } W = 1.30 \text{ lbs., } k = 6901 \text{ lbs./in., } x_0 = 6.5/690 = 0.0094 \text{ in.}$$

$$x = Ax_0 = 0.0094 (0.935) = 0.0088 \text{ in.}$$

$$\text{For 76 cps, } W = 1.18 \text{ lbs., } k = 690 \text{ lbs./in., } x_0 = 5.90/690 = 0.00855 \text{ in.}$$

$$x = Ax_0 = 0.00855 (0.765) = 0.00655 \text{ in.}$$

$$\text{For 80 cps, } W = 0.0221 \text{ lbs., } k = 7.2 \text{ lbs./in., } x_0 = 0.1105/14.4 = 0.0077 \text{ in.}$$

$$x = Ax_0 = 0.0077 (0.641) = 0.00492 \text{ in.}$$

$$\text{For 124 cps, } W = 0.0046 \text{ lbs., } k = 7.2 \text{ lbs./in., } x_0 = 0.023/7.2 = 0.0032 \text{ in.}$$

$$x = Ax_0 = 0.0032 (0.194) = 0.00062 \text{ in.}$$

The steady state response equation is also independent of initial conditions. Since the excitation is sinusoidal, the response is sinusoidal. Also, since downward displacements are restrained, the probable response is a series of sinusoidal pulses with maximum amplitude x.

Effect of Dynamic Load on Existing Generator

The weakest point of the generator from a structural viewpoint is the case. The membrane stress initially present in the axial direction is 7350 psi for the thinnest section. For each g of acceleration, the added load is 2.3 lbs. The area of the thin section is 0.0675 in^2 , so that for each g, stress is increased by $2.3/0.0675 = 34 \text{ psi}$.

The yield point stress for 347 stainless at 1200°F is 24,000 psi. Since the operation time is long and the material drops in rupture strength over periods of time, a stress level of 10,000 psi is used. Consequently, the maximum acceleration that can be applied is found:

$$10,000 - 7350 = 2650 \text{ psi}$$

$$2650/34 = 78 \text{ g max.}$$

The similar approach is used for the horizontal direction:

$$S = PL/(I/c) \quad 2.3 (2.16)/0.0675 = 73 \text{ psi}$$

$$2650/73 = 36 \text{ g max.}$$

Another effect of dynamic loading is movement of the generator parts. For loads applied horizontally, frictional forces between the buttons, electrodes, insulator and hot frame are sufficient to allow accelerations of 85 to 100 g's with no resulting movement of parts. In the vertical direction however, the response amplitudes to shock and vibration indicate a 7 g limitation.

The accelerations that can be applied to the generator are thereby limited by structural strength horizontally and by amplitude vertically.

It is felt that the horizontal value is reasonable. The vertical limitation of 7 g's however, is considered small and probably conservative since it assumes that the only parts that move are the couples and followers. In reality the buttons and electrodes as well as the hot frame can displace somewhat, allowing a larger g force to be applied before the seating depth of the couple in the button is exceeded. The relative movement between the parts as well as that of the couple is very much dependent upon the damping that the packing material has. Since this is to a large degree undeterminable, the actual acceleration to cause unseating can not be calculated. The structural limitation in the vertical direction is 78 g's, so that it is possible to state that a vertical acceleration somewhere between 7 and 78 g's could be applied vertically.

APPENDIX

Derivation of the transient response for a spring-mass system subjected to a sawtooth pulse²:

$$m\ddot{x}/k + x = m\ddot{u}(t)/k = e(t)$$

The solution of the differential equation in LaPlace coordinates is as follows:

$$x(t) = \mathcal{L}^{-1} \frac{sx_0 + \dot{x}_0 + w_n^2 \mathcal{L}e(t)}{s^2 + w_n^2}$$

$$x(t) = \mathcal{L}^{-1} \frac{sx_0}{s^2 + w_n^2} + \mathcal{L}^{-1} \frac{\dot{x}_0}{s^2 + w_n^2} + \mathcal{L}^{-1} \frac{w_n^2 \mathcal{L}e(t)}{s^2 + w_n^2}$$

x_0 and $\dot{x}_0$ are initial conditions. Since $\dot{x}_0$ is zero and x_0 is a constant, interest is in the transient.

$$x(t) = \mathcal{L}^{-1} \frac{w_n^2 \mathcal{L}e(t)}{s^2 + w_n^2}$$

To determine the inverse of $x(t)$, $e(t)$ must be expressed in LaPlace coordinates. $E(t)$ is expressed as a ramp function of duration T and maximum acceleration of $u(t)_{\max}$.

For $0 < t < T$:

$$\mathcal{L} e(t) = \mathcal{L} \frac{\ddot{u}(t)_{\max}}{k} \frac{t}{T} = \frac{\ddot{u}(t)_{\max}}{kT} \frac{1}{s^2}$$

For $t > T$:

$$\mathcal{L} e(t) = \frac{\ddot{u}(t)_{\max}}{kT} \frac{1}{s^2} - \frac{\ddot{u}(t)_{\max}}{kT} \frac{e^{-Ts}}{s^2} - \frac{\ddot{u}(t)_{\max}}{k} \frac{e^{-Ts}}{s}$$

Substitution of the above into the response equation gives the following:

For $0 < t < T$:

$$x(t) = \frac{w_n^2 \ddot{u}(t)_{\max}}{kT} \mathcal{L}^{-1} \frac{w_n^2}{s^2 (s^2 + w_n^2)}$$

The inverse transform can be found in a handbook and the solution for $x(t)$ becomes:

$$x(t) = \frac{\ddot{u}(t)_{\max}}{k} \left(\frac{t}{T} - \frac{\sin(w_n t)}{w_n T} \right)$$

A similar substitution and applying the proper transform gives the result for $t > T$.

$$x(t) = \frac{\ddot{u}(t)_{\max}}{k} \left(-\frac{\sin w_n t}{w_n T} + \frac{\sin(w_n(t-T))}{w_n T} + \cos(w_n(t-T)) \right)$$

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2. Harris & Crede, Shock and Vibration Handbook, Vol. I., McGraw-Hill, 1961. Chapter 8.
3. Crandall & Dahl, Mechanics of Solids, McGraw-Hill, 1959. Chapter 8.
4. 3M Specification TPM 169.
5. Morse, Tse, Hinkle, Mechanical Vibrations. Allyn & Bacon, 1963. Chapter 2.

APPENDIX VII
DYNAMIC ANALYSIS OF 10-WATT GENERATOR
BY DR. R. PLUNKETT

APPENDIX VII
DYNAMIC ANALYSIS OF 10-WATT GENERATOR
BY DR. R. PLUNKETT

December 8, 1966

Mr. Fred Fox
3M Company Space Center
St. Paul, Minnesota

Dear Mr. Fox:

In accordance with our conversation of December 6, I have made a rough survey of the expected response of the SNAP-21B thermoelectric generator to the shock and vibration loadings to be expected during transport by common carrier as specified by MIL-STD-810A. I have looked only at the response of the 3M-37 generator itself and have not considered the effects on the rest of the assembly, casing, heat source, electronics, etc. However, in applying the specifications, I have assumed that the total shipping weight will exceed 500 lb. The appropriate test curves then become (Ref. MIL-STD-810A).

Vibration-Sinusoidal	1.3 G	5 - 26 cps
	0.036"	26 - 50 cps
no testing above 50 cps because total weight is over 500 lb.		

Shock. Sawtooth 3g peak 6 ms duration (check at 6g and 10g).

The enclosed rough calculations show that the lowest natural frequency should be about 150 cps. In order to get a stress of 10,000 psi, an equivalent static acceleration of 100g is necessary transverse to the axis. Even allowing some room for dynamic amplification factor, which is hardly necessary in view of the frequency, this is a very high shock loading.

The calculations indicate that this is a remarkably sturdy piece of equipment and should take handling and shipping dynamic loads successfully with little protection assuming that there are no locally fragile points, i. e., very high stress concentrations in brittle materials.

I am enclosing my original computations and am keeping no file copy. While these are quite crude and involve many assumptions, I do not feel it worthwhile to attempt more refined calculations in view of the complex structure and the unknown characteristics of your insulating filler. If you have any further questions or feel I have overlooked some aspects of the problem, please do not hesitate to get in touch with me again.

Very truly yours,

Robert Plunkett

RP/nb
Enclosure

VII-1

Original Signed by Robert Plunkett

1. AXIAL VIBRATION

Long case

k eff. springs

96 springs at 7.2 lb/in.

k eff = 96 k = 691 lb/in.

$$\text{Long case } k = \frac{AE}{L} = \frac{\pi D t E}{L} = \frac{\pi \times 4.3 \times 5 \times 10^{-3} \times 30 \times 10^6}{1.70}$$

$$= 1.2 \times 10^6 \text{ lb/in.}$$

Total end wt.

1. Hot frame	1.01
2. Retainer	0.16
3. Insulator	0.120
4. Electrodes	0.090
5. Buttons	0.029
6. Couples	0.619
7. Follower	0.442
8. Springs 1.01	0.500 half effective
9. Long case - part	0.10
	<hr/>
	2.92 lb.

$$\omega = \sqrt{\frac{kg}{W}} = \sqrt{\frac{1.2 \times 10^6 \times 386}{2.92}} = 12.6 \times 10^3 \text{ rad/sec}$$

F = 2000 cps.

6

This is unrealistically high and shows that resonance will be controlled by bending in the hot frame.

$$\omega \approx 4.09 \sqrt{\frac{Eh^3 g}{a^2 \times 2.2 \times W}}$$

$$\frac{Eh^3}{a^2 \times 2.2} = \frac{30 \times 10^6 \times 0.25^{-3}}{2.1^{-2} \times 2.2} = 4.83 \times 10^3$$

$$\omega = 4.09 \sqrt{\frac{-4.83 \times 10^3 \times 386}{2.5}} = 3.54 \times 10^3 \text{ rad/sec}$$

$$F = 565 \text{ cps.}$$

This is undoubtedly high - 250-300 cps would be more realistic.

If this were excited at 5g with an assumed resonance amplification of 10 at 250 cps, there would be a double amplitude of 0.080 inch. This would be troublesome if the vibration were that high. There is no requirement to test at over 50 cps and the heavy assembly will attenuate the actual vibration level in any case.

Conclusion: There does not appear to be a serious vibration problem caused by axial excitation.

II. TRANSVERSE VIBRATION

The long case will be more flexible than if all deflection were due to transverse shear.

$$k_s = \frac{AG}{2l} \approx \frac{1}{6} k_x$$

$$k_b = \frac{3EI}{L^3} = \frac{3}{2} \frac{EA}{L} \frac{r^2}{L^2}$$

$$= \frac{3}{2} k_x \frac{r^2}{L^2}$$

$$\text{at worst } \frac{r}{L} = \frac{2.1}{2.8}$$

$$k_b \sim k_x$$

In any case this will give a much higher frequency than the above.

III. UNDER WHAT CONDITIONS WILL THE ELECTRODES CHATTER ON THE INSULATOR?

Mass on spring

8. 1/2 spring	0.005
7. Follower	0.005
6. 1/2 couple	0.0064
5. Button	0.0003
4. 1/2 electrode	0.0009
	<hr/>
	0.0176 lb

$$\omega = \sqrt{\frac{kg}{W}} = \sqrt{\frac{7.2 \times 386}{1.76 \times 10^{-2}}}$$

$$= 1.3 \times 10^3 \text{ rad/sec}$$

$$F = 64 \text{ cps}$$

$$\frac{3.97}{1.76 \times 10^{-2}} \sim 225 \text{ g}$$

There appears to be no difficulty.

IV. STRESS LEVEL DUE TO AXIAL SHOCK

$$\sigma = \frac{F}{A}$$

$$\begin{aligned} A &= \pi D t = \pi \times 4.3 \times 5 \times 10^{-3} \\ &= 6.7 \times 10^{-2} \text{ in.}^2 \end{aligned}$$

$$F = W \times a \text{ (in g's)}$$

$$\sigma = \frac{2.92 \text{ lb} \times a}{6.7 \times 10^{-2}} = 44a \text{ psi}$$

'a' must exceed 200 g static equivalent for σ to equal 10,000 psi.

V. STRESS LEVEL DUE TO TRANSVERSE SHOCK

$$\sigma = \frac{PLr}{I} = \frac{2P}{A} \frac{L}{r}$$

$$\tau = \frac{3P}{A}$$

These are of the same order and making allowance for the numerical factors reduces a to 100 g static equivalent.

APPENDIX VIII
DYNAMIC ANALYSIS OF 10-WATT GENERATOR
BY DR. J. HANLEY

APPENDIX VIII **DYNAMIC ANALYSIS OF 10-WATT GENERATOR** **BY DR. J. HANLEY**

Title: Dynamic Analysis of SNAP-21 Thermoelectric Generator
For: 3M
By: J. T. Hanley, Ph.D., P.E.

Structural System

The structural system is detailed in the sketch below

	<u>Item</u>	<u>Wt (lbs)</u>	<u>Material</u>	<u>Qty.</u>	<u>Wt (Total)</u>
①	Hot Frame	1.01	347SS	1	1.01
②	Retainer	0.0162	302SS	1	0.0162
③	Insulator	0.120	Boron Nitride	1	0.120
④	Electrode	0.00187	Copper	48	0.08976
⑤	Button	0.0003	1020 Steel	96	0.02880
⑥	Couple	0.0129	PbTe	48	0.6192

	<u>Item</u>	<u>Wt (lbs)</u>	<u>Material</u>	<u>Qty.</u>	<u>Wt (Total)</u>
⑦	Follower	0.0046	6061T6	96	0.442
⑧	Spring	0.0105	302SS	96	1.008
⑨	Long Case	0.2057	347SS	1	0.2057
⑩	Short Case	0.327	347SS	1	0.327
⑪	Cold Frame	15.0	Copper	1	15.0
Total Weight					18.87 lbs.

Material Properties

Aside from the weights listed above, the material properties of primary concern in the dynamic analysis are the modulus of elasticity and proportional limit of the long case, Item 9, and the stiffness of the springs, Item 8.

For 347SS; $E = 30 \times 10^6$

For the springs $k = 7.2 \text{ lbs/in}$

(spring stiffness furnished by 3M Company)

Loading Conditions

- A. Shock: See MIL-STD-810A (USAF) 23 June 1964.
Method 516.1

- B. Vibration: See MIL-STD-810A (USAF)
Method 514.1 and Figure 514-6.
Base vibration is assumed to be sinusoidal.

For both shock and vibration analyses, the load will be assumed to act;

- (1) in the direction of the axis of the long case, and
- (2) in the direction orthogonal to the axis of the long case.

Possible Failure Modes

For both shock and vibration, the possible failure modes are:

A. Axial Load Application

- (1) Tension
- (2) Compression
 - (a) Elastic buckling
 - (b) Inelastic compression

B. Lateral Load Application

Combined flexure and shear

Analysis

A. Axial Load Application - Shock

Since the button (Item 5) is not connected to the electrode (Item 4), this system must be treated as a two-degree-of-freedom system.

M_1 = mass of hot frame, retainer, insulators, electrodes, + 1/3 of the weight of the long case (see Reference 1, page 70)

M_2 = mass of buttons, couples, followers, + 1/3 of the weight of the springs

$$k_1 = \text{stiffness of long case in axial compression or tension } \frac{AE}{L}$$

$$k_2 = \text{sum of spring stiffnesses (96k)}$$

If the initial application of the force (base motion) is vertically upward (see sketch), the two masses may act independently; that is, the system may be treated as two independent single degree-of-freedom systems. The displacement of M_1 is resisted by both k_1 and k_2 if the masses M_1 and M_2 are in contact. The displacement of M_2 is resisted only by k_2 if moving downward but by k_1 and k_2 if M_1 and M_2 are in contact and moving upward.

$$M_1 = \frac{1.30453}{0.386 \times 10^3} = 3.39 \times 10^{-3}$$

$$k_1 = \frac{AE}{L} = \frac{0 \times 07.82 \times 3 \times 10^5}{2.164} = 10.8 \times 10^5$$

$$M_2 = \frac{1.426}{0.386 \times 10^3} = 3.7 \times 10^{-3}$$

$$k_2 = 96 \times 7.2 = 6.9 \times 10^2$$

$$\text{Note } k_1 \gg k_2$$

If the initial motion of the base is upward

$$M_2 \ddot{X}_2 + k_2 (X_2 - X_0) = 0$$

$$M_2 (\ddot{X}_2 - \ddot{X}_0) + k_2 (X_2 - X_0) = -M_2 \ddot{X}_0$$

$$\text{Let } X_2 - X_0 = U_2$$

$$\ddot{X}_2 - \ddot{X}_0 = \ddot{U}_2$$

Then

$$\ddot{U}_2 + \frac{k_2}{M_2} U_2 = -\ddot{X}_O$$

$$w_2^2 = \frac{k_2}{M_2} = \frac{6.9 \times 10^2}{3.7 \times 10^{-3}}$$

$$w_2^2 = 1.86 \times 10^5$$

$$w_2 = 0.431 \times 10^3 = 431 \text{ rad/sec}$$

Then

$$T_2 = \frac{2\pi}{w_2} = 1.46 \times 10^{-2} \text{ sec}$$

or $T_2 = 0.0146 \text{ sec}$

For a shock duration of 6 ms = τ

$$\frac{\tau}{T_2} = \frac{0.006}{0.0146} = 0.411$$

The maximum distortion then

$$U_2 = \frac{\ddot{X}_O}{w_2^2} \times 1.0 = \frac{3g w_2}{k_2 g} = 0.0062 \text{ in}$$

The change in the force in the spring is then $F = 7.2 \times 0.62 \times 10^{-2} = 0.0448 \text{ lbs.}$ which is negligible. Even at 10 g maximum acceleration, the maximum possible relative displacement between the button and the electrode is about:

$$X = \frac{10 W_2}{k_2} = \frac{10 \times 1.426}{6.9 \times 10^2} = 0.0207 \text{ in.}$$

Thus, the button may be assumed to remain in contact with the electrode.
When the mass M_1 moves downward;

$$M_1 \ddot{X}_1 + (k_1 + k_2) (X_1 - X_0) = 0$$

Since

$$k_1 \gg k_2 \quad k_1 + k_2 = k_1 \left(1 + \frac{k_2}{k_1} \right)$$

$$k_1 + k_2 = k_1 \left(1 + \frac{6.9 \times 10^2}{10.8 \times 10^5} \right) = k_1 (1 + 0.00064) \approx k_1$$

Then

$$M_1 (\ddot{X}_1 - \ddot{X}_0) + k_1 (X_1 - X_0) = -M_1 \ddot{X}_0$$

or

$$\ddot{U}_1 + W_1^2 U_1 = -\ddot{X}_0$$

$$T_1 = \frac{2\pi}{W_1} = \frac{2\pi}{1.78 \times 10^4} = 3.53 \times 10^{-4} \text{ sec}$$

$$\frac{\tau}{T_1} = \frac{0.006}{0.000353} \approx 17 \quad \text{DAF} = 1.0 \text{ (Dynamic Amplification Factor)}$$

$$U_{1 \text{ max}} = \frac{\ddot{X}_0}{W_1^2} \times 1.0 = \frac{3g}{k_1} \frac{W_1}{g} = \frac{3 \times 1.30453}{1.0 \times 8 \times 10^5}$$

$$U_{1 \max} = 3.62 \times 10^{-5} \text{ in.}$$

$$F_m = k_1 U_{1 \max}$$

$$\sigma = E \epsilon$$

$$\sigma_m = \frac{k_1 U_{1 \max}}{A}$$

$$\sigma_m = E \frac{u_{1 \max}}{L}$$

$$\sigma_m = \frac{E}{L} \times \frac{3 W_1 L}{A E} = \frac{3 W_1}{A}$$

$$\sigma_m = 50.2 \text{ psi}$$

Negligible.

$$\text{For } \ddot{X}_O = 6g$$

$$\sigma_m \sim 100 \text{ psi}$$

$$\text{For } \ddot{X}_O = 10g$$

$$\sigma_m \sim 167 \text{ psi}$$

When the mass M_1 moves upward;

$$(M_1 + M_2) \ddot{X}_1 + (k_1 + k_2) (X_1 - X_O) = 0$$

Then

$$(M_1 + M_2) \ddot{U}_1 + k_1 U_1 = -(M_1 + M_2) \ddot{X}_O$$

or

$$\ddot{U}_1 + \frac{k_1}{M_1 + M_2} U_1 = -\ddot{X}_O$$

Let

$$\frac{k_1}{M_1 + M_2} = P_1^2$$

$$T = \frac{2\pi}{P_1} \quad T^2 = \frac{(2\pi)^2 (M_1 + M_2)}{k_1}$$

$$T^2 = \frac{(2\pi)^2 \times 7.09}{10.8 \times 10^5} \times 10^{-3} = 26 \times 10^{-8}$$

$$T = 5.1 \times 10^{-4} = 0.00051$$

and

$$\frac{\tau}{T} = \frac{0.00600}{0.00051} = 11.8$$

Then for

$$\ddot{X}_O = 3g$$

$$\sigma_M = \frac{3(W_1 + W_2)}{A} = \frac{3 \times 2.731}{0.0782} = 116 \text{ psi}$$

For

$$\ddot{X}_O = 6g; \quad \sigma_M \sim 232 \text{ psi}$$

For

$$\ddot{X}_O = 10g; \quad \sigma_M \sim 350 \text{ psi}$$

Again negligible.

Check buckling stress:

$$(1-\nu^2) q_{cr} \frac{R}{Eh} = \frac{(1-\nu^2)}{(n^2-1) \left(1 + \frac{n^2 L^2}{\pi^2 R^2} \right)}$$

$$+ \frac{h^2}{12R^2} \left[n^2 - 1 + \frac{(2n^2 - 1 - \nu)}{\left(1 + \frac{h^2 L^2}{\pi^2 R^2} \right)} \right]$$

Here

$$\nu \approx 0.3$$

$$n = 2$$

$$R = 2.15 \text{ in.}$$

$$E = 30 \times 10^6 \text{ psi}$$

$$h = 0.0058 \text{ in.}$$

$$L = 2.164 \text{ in.}$$

$$q_{cr} = 19,100 \text{ lbs/in}^2 \gg 167 \text{ psi} \quad \text{OK}$$

The long case will not fail in any mode, subjected to axial application of shock loading.

B. Axial Load Application - Vibration

The behavior of the system under sinusoidal base motion is quite complicated. A complete solution would require computer analysis. For this analysis, two possible systems will be considered:

(a) M_2 ; vibrating alone

(b) $M_1 + M_2$; vibrating alone

The assumed frequency of base motion is 400 cps (see Figure 514.7, Reference 2). Then, from Figure 514.6, Reference 2, the maximum acceleration for design is 5g.

It is possible that consideration should be given to placing an absorber of some type between the electrode and the button to absorb the impact under both shock and vibration.

C. Lateral Load Application - Shock

Assume first no interaction with MIN-K 1301.

Because the springs supporting the button, couple, and follower have negligible lateral resistance, it is assumed that the entire mass of the system is restrained only by the long case. Further, because the centroid of the mass is approximately at the end of the long case, there will be no applied moment at the end, only a shear force;

$$F = M\ddot{y}$$

Using the static deflection method of Rayleigh

$$\left(\text{i. e., } p^2 = \frac{g}{y_{st}} \right) ;$$

Deflection due to shear

$$y = \frac{k W L}{A G}, \text{ where } W = W_1 + W_2$$

Deflection due to Moment;

$$y = \frac{W L^3}{3 E I}$$

Total deflection;

$$y_{st} = \frac{k W L}{A G} + \frac{W L^3}{3 E I} = \frac{k W L}{A G} \left(1 + \frac{L^2 A G}{3 E I k} \right)$$

Since

$$G = \frac{E}{2 (1 + \nu)} \quad \text{and} \quad \frac{I}{A} = r^2$$

$$y_{st} = \frac{k W L}{A G} \left(1 + \frac{L^2}{3 \times 2 (1 + \nu) r^2 k} \right)$$

$$r^2 = \sqrt{\frac{r^2 + r_1^2}{2}} \approx \sqrt{\frac{2 r^2}{2}} = \frac{r}{\sqrt{2}} = 1.52$$

$$y_{st} = \frac{1.5 \times 2.73 \times 2.164}{0 \times 0.782 \times 11.5 \times 10^6} \left(1 + \frac{(2.15)^2}{6 (1.3) 1.52 \times 1.5} \right)$$

$$y_{st} = 9.9 \times 10^{-6} (1 + 0.26) = 12.5 \times 10^{-6}$$

$$p^2 = \frac{3.86 \times 10^2}{12.5 \times 10^{-6}} = 0.309 \times 10^8 (\text{rad/sec})^2$$

$$p = 0.555 \times 10^4 \text{ rad/sec}$$

$$T = \frac{2\pi}{p} = 11.3 \times 10^{-4} \text{ sec}$$

$$\frac{\tau}{T} = \frac{6 \times 10^{-3}}{1.13 \times 10^{-3}} = 5.3$$

∴ Dynamic amplification factor ~ 1.0

Maximum flexural stress:

$$\sigma_M = \frac{Mc}{I} = \frac{(\alpha W \cdot L) c}{I}$$

where

$$\alpha = 3, 6, 10$$

$$I = 0.0491 (d^4 - d_1^4) = 0.078$$

$$d = 4.3 d_1 = 4.295$$

$$c = 2.15 \text{ in.}$$

For

$$\alpha = 3; \quad \sigma_M = \frac{3 \times 2.73 \times 2.164 \times 2.15}{7.8 \times 10^{-2}} \sim 490 \text{ psi}$$

For

$$\alpha = 6; \quad \sigma_M \sim 980 \text{ psi}$$

For

$$\alpha = 10; \quad \sigma_M \sim 1630 \text{ psi}$$

Negligible again. Since stresses are negligible without considering interaction, it would be a waste of time to analyze the more complicated problem.

D. Lateral Load Application - Vibration

Here again, for total weight of 2.73 lbs., use frequency of 400 cps (Figure 514-7, Reference 2). From Figure 514-6, the design acceleration is 5g, which produces stresses under 1000 psi.

Conclusions

Neither shock nor vibration loading well beyond 10g will produce significant stresses in the long case. However, if possible, consideration should be given to placing a shock and vibration absorber somewhere in the linkage between the spring and the electrode to prevent repeated impact in that system.

References

1. Jacobsen and Ayre, Engineering Vibrations, McGraw-Hill, 1958.
2. MIL-STD-810A (USAF) 23 June 1964.

APPENDIX IX

**PRELIMINARY DESIGN OF SNAP-21 SHIPPING CONTAINER
SUSPENSION SYSTEM**

APPENDIX IX

PRELIMINARY DESIGN OF SNAP-21 SHIPPING CONTAINER SUSPENSION SYSTEM

KNOWN DATA

Fragility Level - 6 g's

System Frequency less than 5 cps

Metal Container

Subject to 3" flat and 6" corner drop

Baseline Configuration 10-Watt System

Maximum acceleration of supported or isolated unit*

$$\ddot{\delta}_{\max} = (W_n \sqrt{2gh} + g) \text{ in. /sec}^2$$

Maximum deflection of supported unit*

$$\delta_{\max} = \left(\frac{\sqrt{2gh}}{W_n} + \frac{2g}{W_n^2} \right) \text{ in.}$$

*Harris and Crede, Shock and Vibration Handbook. McGraw-Hill, 1961. Vol. 3, Chapter 41.

Figure IX-1. Maximum Acceleration vs Frequency for 3-Inch Drop Height

Figure IX-2. Maximum Deflection vs Frequency for 3-Inch Drop

Figure IX-3. Spring Rate vs Frequency for 600-Pound Generator

Natural frequency is given by*

$$W_n = \sqrt{\frac{k}{m}}$$

The frequency of the suspension is to be 5 cps or smaller to prevent resonance and the curve for acceleration versus frequency for a 3-inch drop indicates that acceleration is under 6 g's for frequencies less than 5 cps. The smaller the frequency however, the larger the deflection becomes, so should the frequency be kept close to 5 cps to minimize excursion.

Shear mountings are proposed as means of isolating the generator. Shipping container mountings are available with rates of 10 to 5000 lbs/in. and deflection capabilities of 22 inches in shear.** The requirements for the generator system are a total spring rate of 1500 lbs/in. with a deflection of about 2.3 inches. For all attitudes of protection, double shear mounts can be used.

*Harris and Crede, Shock and Vibration Handbook. McGraw-Hill, 1961. Vol. 3, Chapter 41.

**General Catalog - Shipping Container Products lb-1600, July 1962, Lord Manufacturing, Erie, Pennsylvania.