**HIGH-LEVEL WASTE
CANISTER ENVELOPE STUDY**

STRUCTURAL ANALYSIS

MASTER

prepared for

ROCKWELL HANFORD COMPANY

CONTRACT NO. CPFF-626

TASK 626-01

UNDER PRIME CONTRACT NO. EY-76-C-06-2130

November 1977

**Kaiser
Engineers**

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KAISER ENGINEERS
DIVISION OF HENRY J. KAISER COMPANY

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MASTER

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TABLE OF CONTENTS

	<u>Page</u>
I. INTRODUCTION	1
II. SUMMARY AND RECOMMENDATIONS	2
III. CANISTER LIFE CYCLE	12
A. Discussion	
B. Design Criteria	
IV. DESCRIPTION OF CANISTER MODELING	22
A. Canister Configuration	
B. Material	
C. Finite Element Model	
V. BASIC MODEL CANISTER STRESS DUE TO NORMAL LOADING CONDITIONS	30
A. Mechanical Load - Dead Load, Live Load & Internal Pressure	
B. Heat Transfer Analysis	
C. Thermal-Elastic Analysis	
D. Combination of Loadings	
E. Welding Thermal Effects	
F. External Pressure	
VI. BASIC MODEL CANISTER STRESS DUE TO ABNORMAL AND IMPROBABLE EVENTS	37
A. Maximum Temperature Excursion	
B. Maximum Pressure Excursion	
C. Drop	
D. Earthquake	
F. Quenching in Water	

	<u>Page</u>
VII. BASIC MODEL ANALYTICAL RESULTS FOR CANISTER LIFE CYCLE STAGES	42
A. Waste Solidification	
B. Interim Storage at Fuel Reprocessing Plant Stage	
C. Shipping Stage	
D. Geological Storage Stage	
VIII. PARAMETRIC STUDIES	48
A. Effects of Varying Canister Diameter - Normal Loading Conditions	
B. Effects of Canister Length - Normal Loading Conditions	
C. Effects of Canister Diameter and Length - Abnormal and Improbable Events	
D. Effects of Different Materials (Type 304L and Type 410)	
E. Canister Handling Limitations	
IX. APPENDIX	
APPENDIX A. IMPACT STUDY OF HIGH-LEVEL WASTE CANISTER	
APPENDIX B. STUDY OF THE EFFECT OF EARTHQUAKE ON A FREELY SUSPENDED CANISTER	
APPENDIX C. DISPOSAL HOLE LINERS	

LIST OF TABLES

	<u>Page</u>
Table 1 - Structural Integrity of High-Level Waste Canisters Under Life Cycle Loading	3
Table 2 - Life Cycle of High-Level Waste Canister	13
Table 3 - Stress Categories and Allowable Stress	30
Table 4 - Maximum Stress vs. Allowable Stress by Stress Category	32
Table 5 - Canister Thermal Elastic Analysis Results Maximum Stress at 100°F Thermal Gradient	36
Table 6 - Maximum Stresses During Solidification	45
Table 7 - Maximum Stress in Various Diameter Canisters - Normal Loading Conditions	49
Table 8 - Maximum Thermal Stress in Canisters - Using Different Materials	52

LIST OF FIGURES

<u>Figure</u>		<u>Page</u>
1	Twelve (12)-Inch Diameter Canister	23
2	Twenty-Four (24)-Inch Diameter Canister	24
3	Finite Element Model 1 - Coarse Mesh - HLW Canister	26
4	Finite Element Model 2 - Fine Mesh - HLW Canister	27
5	Finite Element Model 3 - One Segment of Canister Wall	28
6	HEATING5 Model - HLW Canister	28
	(a) Top Head	
	(b) Bottom Head	
7	Steady State Temperature Distribution of 100°F Gradient Across Top and Bottom Heads of Canister	34
	(a) Section A-A of Top Head	
	(b) Section B-B of Top Head	
	(c) Section C-C of Bottom Head	
8	Transient Canister Wall Temperatures at 3 Seconds	41
9	Circumferential Thermal Stress in Canister Wall During Transient State at 3 Seconds	43

Appendix A A1 - A12

Appendix B B1 - B9

STRUCTURAL ANALYSIS OF HIGH-LEVEL WASTE CANISTER

I. INTRODUCTION

This report presents the results of a study of the structural integrity of high-level waste canisters subjected to loadings resulting from normal, abnormal, and improbable events occurring during the canister life cycle. The study was performed by Kaiser Engineers for Atlantic Richfield Hanford Company (now Rockwell Hanford Company) as Task 626-01 under subcontract number CPFF-626 during the period from June through September 1977. This work is part of the U.S. Government's overall program for the management and ultimate storage of high-level solidified radioactive wastes from the commercial nuclear power industry. Technical direction for the overall program is provided by the Office of Waste Isolation in Oak Ridge, Tennessee for the Department of Energy.

The canister is assumed to be filled with vitrified high-level nuclear waste, stored temporarily at a fuel reprocessing plant, then transported for storage in an underground salt bed or other geologic storage. The basic or reference canister model for the study is 12-inches in diameter by 10-feet long with a 3/8-inch thick wall. Alternative designs of canisters with diameters of 8-inches and 24-inches and lengths of 10 and 16 feet were also examined.

In general, the study was based upon accepted structural and finite element analytical techniques, and conformance with the allowable stresses and requirements of the ASME Boiler and Pressure Vessel Code, Section VIII - Division 2. The potential contribution of the solidified waste in the canister to the structural integrity of the canister was not considered in the study.

II. SUMMARY AND RECOMMENDATIONS

A. Summary

In the High-Level Waste Canister Study, the structural integrity of waste canisters, fabricated from standard weight Type 304L stainless steel pipe, was analyzed for sizes ranging from 8 to 24-inches in diameter and 10 to 16-feet in length under normal, abnormal, and improbable life cycle loading conditions.

The improbable events which might cause major damage to the canister such as aircraft impact, flood, sabotage or a tornado are not considered in the study. Canister exposure to such events is assumed to be prevented by protective measures in the facility and transport hardware design.

A canister is considered acceptable for containment of high-level waste if the analysis finds that the structural integrity of the canister is retained under all life cycle loading conditions.

The results of the structural analyses are summarized below and in Table I.

1. Basic Model Canister - Normal Loading Conditions

- a. The stresses imposed on the basic model canister made of stainless steel, 12-inch diameter by 10-feet long with a 3/8-inch wall, by normal loading conditions occurring during the canister life cycle will not exceed the permissible limits stipulated in the ASME Boiler and Pressure Vessel Code, Section VIII - Division 2 except for direct burial in salt.

TABLE I

STRUCTURAL INTEGRITY OF HIGH LEVEL WASTE CANISTERS UNDER LIFE CYCLE LOADING

LIFE CYCLE STAGES	INCIDENT CONFIGURATION (1)		NORMAL CONDITIONS						ABNORMAL EVENTS					IMPROBABLE EVENTS		
			COMBINATION OF LOADINGS (DEAD + LIVE + THERMAL + PRESSURE)						THERMAL EXCURSION	OPERATING BASIS EARTHQUAKE	THERMAL SHOCK	DROP	PRESSURE EXCURSION	SAFE SHUTDOWN EARTHQUAKE	MINE SHAFT DROP	
	C1	C2														C3
WASTE SOLIDIFICATION	DURING CANISTER FILL AND SOLIDIFICATION		A	A	A	A	A	NA								
	DURING CLOSURE		A	A	A	A	A	NA								
	DURING TRANSFER		A	A	A	A	A	NA	A				A	NA		
				A								NA			A	
INTERIM STORAGE AT FUELS REPROCESSING PLANT	INTERIM STORAGE	WET	A	A	A	A	A	NA								
		AIR	A	A	A	A	A	NA	A				A	NA		
SHIPPING	SHIPPING	WET	A	A	A	A	A	NA	A				A	NA		
		DRY	A	A	A	A	A	NA	A				A	NA		
GEOLOGIC STORAGE	RECEIVING AT STORAGE		A	A	A	A	A	NA		A					A	
	TRANSFER TO MINE											NA				
			A	A	A	A	A	NA	A				A	NA		
	STORAGE	BASALT	A	A	A	A	A	NA		A					A	NA
		SHALE	A	A	A	A	A	NA								
SALT		NA	NA	NA	NA	NA	NA									

NOTES:

1. CONFIGURATIONS: C1: 8-IN DIAMETER X 10 FT LONG X 0.322-IN WALL (304L SS)
C2: 8-IN DIAMETER X 16 FT LONG X 0.322-IN WALL (304L SS)
C3: 12-IN DIAMETER X 10 FT LONG X 0.375-IN WALL (304L SS)
C4: 12-IN DIAMETER X 16 FT LONG X 0.375-IN WALL (304L SS)
C5: 24-IN DIAMETER X 10 FT LONG X 0.375-IN WALL (304L SS)
C6: 24-IN DIAMETER X 16 FT LONG X 0.375-IN WALL (304L SS)

LEGEND:

A = ACCEPTABLE, NA = NOT ACCEPTABLE

The estimated external pressure which can be exerted on the canister while buried in salt at the assumed depth of 2000 feet is 3500 psi. The maximum external pressure which a 12-inch diameter, 3/8-inch thick steel canister can resist is 900 psi; therefore, the shell thickness would need to be substantially increased to withstand direct burial in salt. An alternative approach, to maintain the structural integrity of the canister after burial, would be to provide thick-wall sleeves in the salt into which the canister could be placed to resist the salt pressure.

The canister exterior pressure design limits established by the ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, are:

- o 8-inch diameter - 1150 psi
- o 12-inch diameter - 900 psi
- o 24-inch diameter - 450 psi

Stainless steel is susceptible to pitting corrosion and chloride stress cracking if buried in salt in the presence of moisture. (See Appendix C) Although stainless steel may be a suitable corrosion-resistant material for canisters buried in basalt or shale, it is not an acceptable material for canisters buried in salt.

- b. The threads of the canister closure plug would be less subject to galling if fabricated of Type 410 instead of Type 304L stainless steel. However, the use of materials with different coefficients of thermal expansion for the top head and the

shell of the canister produce excessive stresses in the transition zone between the thin shell and the top head; whereas, the thermal stresses are within allowable code limits if the same materials are used for both the cylinder shell and the top head of the canister.

- c. During waste solidification the canister is batch-filled with a waste-frit mix (molten glass) at a temperature of approximately 1000°C (1832°F). Since the solidifying mass of vitrified waste has approximately half the coefficient of thermal expansion of the stainless steel, the canister can develop residual hoop stresses during the cooling process in excess of the allowable code limits. Therefore, it is necessary to provide an external mechanical cooling system to keep the canister temperature below 800°F and the associated stresses within code limits.

2. Basic Model Canister - Abnormal Event Loading

- a. Thermal Excursions - Type 304L stainless steel is limited by the ASME Boiler and Pressure Vessel Code, Section VIII - Division 2, to a maximum service temperature of 800°F . Although there is not, as yet, complete agreement between investigators as to the maximum temperature excursions to expect from high-level waste, such as may be caused by loss of shipping cask cooling, no maximum temperatures higher than 690°F have been predicted. Therefore, thermal excursions do not appear to constitute a threat to the canister.
- b. Operating Basis Earthquake (OBE) - The limit method of analy-

sis used to assess the structural integrity of the canister during its various life cycle stages indicates that an operating basis earthquake intensity of 0.125g may cause the canister to be slightly dented (approximately 0.01 inches) but it would retain its structural integrity.

- c. Thermal Shock - The basic model canister at the maximum study temperature of 650°F would be overstressed if subjected to the thermal shock of sudden quenching in a pool of 120°F water. A limiting canister temperature of 350°F would be required to avoid overstressing the canister by quenching. A controlled cooling system must provide a means of slow cooling where a sudden temperature change is possible during the canister life cycle. However, a non-linear time history analysis of the canister quenching would allow further investigation of the structural integrity of the canister.

The non-linear analysis utilizing the MARC-CDC computer program will permit quantitative evaluation of the residual strain and stress in the canister after quenching.

- d. Canister Drop - If the basic model canister is dropped a distance of 20 feet onto a hard surface, the limit analysis used in this study indicates that the impact force will be far greater than the elastic limit of the canister material and result in severe structural damage (buckling) to the canister. However, modifying the energy-absorbing skirt design will mitigate the damage.

A comprehensive finite element analysis of the canister utilizing a sophisticated computer program such as the MARC-CDC is required to ascertain whether the canister will buckle without rupturing or the canister skirt will buckle first and protect the canister.

- e. Pressure Excursion - The maximum internal pressure which the basic model canister can resist without exceeding its allowable code stress limit is 800 psi at the study temperature of 650°F. Therefore, the assumed excursion pressure of 100 psi considered in the study is well below the code limit.

3. Basic Model - Improbable Event Loading

- a. Safe Shutdown Earthquake (SSE) - The results of the limit analysis of the basic model canister, if subjected to the intensity of a 0.25g safe shutdown earthquake, indicate that the wall of the canister may be slightly dented but it should survive the earthquake without structural failure.
- b. Mine Shaft Drop - The force of the impact resulting from accidental dropping of the canister down the mine shaft to a geologic storage horizon is greater than the elastic limit of stainless steel and would cause buckling of the canister and brittle rupture in the impact zone.

Structural failure can be assumed in the improbable event that a waste canister free-falls down the mine shaft. Therefore, supplemental methods of cushioning, cleanup and confinement of the accident should be required in the design of the geologic storage facility.

*How the debris
is contained?*

4. Parametric Studies

In addition to analyzing the basic model 12-inch diameter by 10-feet long canister, a range of other canister sizes were studied under life cycle loading conditions. Canister diameters of 8 and 24-inch and lengths of 10 and 16-feet were analyzed. The 12-inch diameter basic model canister with its length increased from 10 to 16-feet was also included in the analysis. The results of these parametric studies are summarized below and in Table I.

a. Normal Loading Conditions

The results of the structural analyses of the 12-inch diameter by 10-feet long basic model canister, under normal loading conditions, as summarized in Section II.A.1, which found the basic model to withstand all combinations of normal loading except burial in salt, may also be applied to the following canister configurations:

1. 8-inch diameter x 10 ft long x 0.322 inch wall
2. 8-inch diameter x 16 ft long x 0.322 inch wall
3. 12-inch diameter x 10 ft long x 0.375 inch wall
4. 12-inch diameter x 16 ft long x 0.375 inch wall
5. 24-inch diameter x 10 ft long x 0.375 inch wall.

Although the maximum combined stress under the combination of normal loadings indicated in Table I for 16-feet long canisters is within the allowable code limits for both the 8 and 12-inch

diameter canisters, the allowable stress is exceeded by 13% in the 3/8-inch (0.375 inches) thick bottom head of the 24-inch by 16-feet long canister. Therefore, this canister configuration is not acceptable without reinforcement or an increase in the bottom head thickness.

b. Abnormal and Improbable Event Loading Conditions

Additional parametric investigations of 8 and 24-inch canisters, and a 12-inch diameter by 16-feet long canister, indicate that stress levels resulting from the abnormal event of thermal excursions are acceptable (within code allowable stress) for all canister configurations studied.

In the event of a pressure excursion up to 100 psi, only the 24-inch diameter by 16-foot canister will experience stress in the bottom head exceeding the allowable code limit.

In the event of either an operating basis earthquake (0.125g) or a safe shutdown earthquake (0.25g), all canister configurations analyzed in the study will survive without failure.

Stress levels resulting from the abnormal and improbable events of thermal shock, a drop of 20 feet onto a hard surface or a mine shaft drop, as discussed in Section II.A.2 and 3 for the basic model canister, exceed allowable code and material limits and are not acceptable for any of the configurations studied.

c. Canister Handling Limitations

The handling of canisters up to and including 24-inch diameter

and 16-feet in length in the terminal storage facility has been reviewed. In general, the technical problems of handling large canisters appear to be resolvable during the conceptual design of each handling facility. However, the primary consideration with increasing canister size is the associated increase in fixed capital investment of the handling facilities and hardware, and the cost-benefit impact on the long-term waste management system.

d. Selection of Optimum Canister

Selection of an optimum diameter and length, wall thickness, and material for a high-level waste canister does not appear to be restrained by the technical problems of engineering and designing adequate handling, transport systems, and hardware. The selection of an optimum canister must rely on an economic evaluation of the long-term waste management system.

B. Recommendations

- a. In each of the three impact conditions studies, the resulting impact force is far greater than the elastic limit capacity of the material used. Therefore, additional study of the canister and skirt design using a sophisticated computer program (e.g., MARC-CDC) is recommended for further examination of the buckling behavior to determine whether the canister will buckle without failing or the skirt will buckle and protect the canister.
- b. A canister drop test of 20-feet onto a hard surface is recommended to verify results of the structural analysis.

- c. Since it was concluded in the impact analysis that a properly designed canister skirt could be effective in mitigation of free drop (20 ft.) damage, further study of skirt configuration is recommended.
- d. Since the thermal stress level exceeds the code limit for elastic analysis, non-linear time history analysis of canister quenching in a water pool is recommended to permit quantitative evaluation of the residual strain and stress in the canister after quenching.
- e. Refinement of temperature and internal pressure values are required as criteria for further analysis of high-level waste canisters.
- f. Further canister and/or protective sleeve concept development in materials other than stainless steel is required for canisters to be stored in salt.
- g. The non-structural canister seal weld integrity should be verified by 20-foot drop tests of a prototype canister(s).

III. CANISTER LIFE CYCLE

A. Discussion

The postulated loading conditions imposed on a canister during its life cycle are indicated in Table 2 for each of the four major stages of the canister life cycle. The first stage is filling the canister with waste in the form of either calcine or molten glass at a reprocessing plant. The second stage consists of interim storage of the canister at a fuel reprocessing plant for a minimum of five years while radioactivity decays and heat generation is reduced to a level acceptable for waste shipment (the third stage) to geologic storage. The fourth life cycle stage involves handling and burial of the canister in a geologic formation of basalt, shale, or salt.

During the life cycle of the waste canister, it will be exposed to stresses associated with routine and normal events such as canister fabrication, elevated temperatures during filling, sealing by welding, high temperatures generated by the radioactive decay of the waste, and both internal and external pressures. Abnormal and improbable events are those which have been postulated as accidental occurrences to which the canister could be subjected during its life cycle as indicated in Table 2.

Of all the improbable events considered, mine shaft drop and earthquake are considered to represent the most severe threats to the canister. For the purpose of this study, it is assumed that the waste canister will never be directly exposed to flood, sabotage or the impact of aircraft or missiles, because adequate protection would be provided by site location, administrative controls, and secondary

TABLE 2
LIFE CYCLE OF HIGH LEVEL WASTE CANISTER

LIFE CYCLE STAGES	INCIDENT CANISTER POSTURE		NORMAL CONDITIONS					ABNORMAL EVENTS (5)					IMPROBABLE EVENTS (5)						
			DEAD LOAD	LIVE LOAD	THERMAL	INTERNAL OR EXTERNAL PRESSURE	WELDING THERMAL	THERMAL EXCURSION	OPERATING BASIS EARTHQUAKE (OBE)	PRESSURE EXCURSION	THERMAL SHOCK	DROP	(1) TORNADO	SAFE SHUTDOWN EARTHQUAKE (SSE)	(2) AIRCRAFT MISSILE	(3) FLOOD	(4) SABOTAGE	MINE SHAFT DROP	TORNADO
WASTE SOLIDIFICATION	DURING CANISTER FILL AND SOLIDIFICATION		●	●	●														
	DURING CLOSURE		●	●	●		●												
	DURING TRANSFER		●	●	●	●		●	●			●		●					
INTERIM STORAGE AT FUELS REPROCESSING PLANT	INTERIM STORAGE	WET	●	●	●	●													
		AIR	●	●	●	●		●		●		●							
SHIPPING	SHIPPING	WET	●	●	●	●		●		●									
		DRY	●	●	●	●		●		●									
GEOLOGIC STORAGE	RECEIVING AT STORAGE		●	●	●	●			●				●						
	TRANSFER TO MINE		●	●	●	●		●		●								●	
	STORAGE	BASALT	●	●	●	●													
		SHALE	●	●	●	●													
		SALT	●	●	●	●													

NOTES:

1. CANISTER NOT EXPOSED TO TORNADO.
2. BARE CANISTER PROTECTED FROM AIRCRAFT MISSILES BY SECONDARY CONTAINMENT.
3. SITE DEPENDENT.
4. FACILITY TO BE SECURITY PROTECTED IN ACCORDANCE WITH ANSI/N 18.17 - 1974 AND 10 CFR 73.
5. INDIVIDUAL AND COMBINATIONS OF LOADS ANALYZED ARE ON THE SAME HORIZONTAL LINE.

confinement barriers.

The following paragraphs discuss the processing and the assumed loading conditions imposed on the canister during each life cycle stage:

1. Waste Solidification Stage

Although several alternative waste forms have been examined over the past several years, for this study it was assumed that the waste would be in a powdered calcine form which has been further stabilized into a glass-like ceramic by adding glass frit, melting the waste frit mix in a glass melter, followed by pouring the "molten" glass into the canister and allowing the mix to solidify into an impervious mass in the canister.

While the canister is being filled with waste-frit mix (molten glass) at a temperature of about 1832°F (1000°C)*, it is assumed that the canister will be cooled by external mechanical systems to maintain it well below the maximum temperature of 800°F allowed by the ASME Boiler and Pressure Vessel Code Section VIII - Division 2.

In-can melting of the waste-frit mixture was not considered in this study. An earlier study** indicates that unacceptably high residual stresses would be induced in the waste canister by the

* "Calcine Conversion Facility: Alternative Concepts Engineering Studies" by Atlantic Richfield Hanford Company, Kaiser Engineers, and Pacific Northwest Laboratory. ARH-LD-108, February 1975; pp 23.

** "Vitrification of Nuclear Waste Calcines by In-Can Melting" by H.T. Blair, Battelle Pacific Northwest Laboratories. BNWL-2061, UC70, May 1976.

in-can method of melting. Since in-can melting takes place with the canister in a furnace, a feasible method of limiting the temperature rise of the canister is not apparent and requires further investigation.

During canister closure welding, residual stresses can be created in the welding zone. Therefore, the welding thermal stress is considered as one of the normal loading conditions.

2. Interim Storage at Fuel Reprocessing Plant Stage

During transfer operations in the reprocessing plant, the canister will be passed through relatively small openings or ports in hot cell floors. One of the assumed abnormal events is that an earthquake causes the canister to swing and strike the side of an opening.

Another abnormal event assumes that during transfer of the canister from the waste solidification operations to interim storage, the canister could be subjected to an impact loading condition if it was dropped (no credit assumed for canister reinforcement by contents). Therefore, it is assumed in this study that the canister could fall 20 feet, land directly on its base on a concrete floor and then fall to a horizontal position.

It is also assumed that a transfer cask cooling system may fail with the canister contained in a non-ventilated shielded transfer cask. If this abnormal event occurs, the waste canister would experience a temperature increase (thermal excursion) and an accompanying pressure increase (pressure excursion), until cooling

could be restored.

Two modes of interim storage are considered:

1. Wet - Canisters arrayed vertically in water storage pools
2. Dry - Canisters arrayed vertically in shielded vaults or containers which are cooled by either natural or forced ventilation.

No risks to the canister structural integrity during interim wet storage have been considered in the study because it is assumed that any loss of coolant could be prevented by providing a secondary water supply system or other emergency water-makeup and cooling systems. However, if the canister is suddenly immersed in water to provide emergency cooling, the canister would be subjected to thermal shock and a severe thermal gradient would produce high stresses in the canister shell. Therefore, the possibility of thermal shock to the canister while at the fuel reprocessing facility was analyzed.

Dry canister interim storage in air presents the possibility of cooling air supply reduction or loss due to a restriction or blockage in the ventilation air entrance or exhaust passages. It is also possible that a period of time could elapse before discovery of the air blockage. The resultant loss of air cooling is assumed to produce the same temperature and pressure excursions that would be experienced due to loss of the cooling system during waste canister transfer in a transfer cask.

3. Shipping Stage

It is assumed that high level waste canisters will initially be shipped to the geologic storage facilities using shipping casks whose structural integrity has been verified by design and test to meet Department of Transportation Regulations. The hazard to the waste canisters during shipment is assumed to be the loss of coolant due to an accident or other mishap during transport. The loss of shipping cask coolant would result in a canister thermal excursion and an accompanying pressure excursion. The shock loading to the canisters will depend on the specific design of the cask intervals and their ability to withstand the loading and protect the canisters.

4. Geologic Storage Stage

It is assumed the waste canisters will be transported to the terminal geologic waste storage facility by truck or rail. The canisters will be removed from the shipping cask into a hot cell and prepared for transfer to the underground mine for permanent burial.

As the canisters are processed through the hot cell, the abnormal hazards they could encounter are assumed to be similar to those postulated during transfer at the waste solidification facility (i.e. earthquake and drop). The most severe potential accident at the terminal facility would be a drop of the canister several thousand feet down the mine shaft. While this event is highly improbable, it is considered as the extreme accident condition at the terminal storage facility for the purpose of this study.

Three types of geologic structures are being considered for geologic storage of high-level radioactive waste canisters: salt, basalt and shale. Because basalt and shale formation are characterized by extremely high compressive strength, it is assumed that they will not exert any loads of crushing magnitude on the buried canisters. However, salt being highly plastic can produce excessive external pressures on the waste canisters. Earlier studies indicate that this pressure could reach as high as 3500 psi at depths of 2000 ft*.

B. Design Criteria

1. Dead Load

Density of stainless steel: 500 lb/ft^3

2. Live Load

Density of waste: 187 lb/ft^3

3. Thermal Effect of Normal Life Cycle Condition

The temperature drop across the canister wall is a function of heat generation of the waste within the canister ($Kw/\text{canister}$). For a 12-inch diameter by 10 feet long canister with a 3/8-inch thick wall the temperature drop is indicated below:

* Callahan, G. D., and Ratigan J. L., "Thermo/Viscoelastic Analysis of the Waste-Container Sleeve: Time Dependent Salt Loading on the Sleeve and Closure of the Drillhole. ORNL-SUB-4269-4, October 1975.

<u>Temperature Drop</u>	<u>Kw/Canister</u>
36°F	10
1.8°F	5
0.4°F	1

The amount of heat generation depends on the type and age of the waste in the canister. For this study, a heat generation level of 5 Kw per canister and a 1.8°F temperature drop across the wall at steady state is assumed as design criteria for the analysis purposes.

The steady state temperature of the outside canister shell or wall surface in air is assumed to be approximately 650°F (343°C) during all life cycle stages except for final storage in sale. The design temperature of 650°F and design internal pressure of 100 psi for a canister in air are based upon assumed extreme values for use in the structural analysis. These critical numbers were established as criteria during a meeting in Richland, Washington on June 8, 1977, based on extrapolation from past studies and will be refined after the waste form and age have been finalized.

The steady state temperature of the outside canister surface in water during interim storage is assumed to be approximately 120°F (49°C)**.

** "Retrievable Surface Storage Facility Water Basin Concept: Conceptual System Design Description", by Atlantic Richfield Hanford Company and Kaiser Engineers, ARH-2799, June 1973, pp 3.2-4

The steady state temperature of the outside canister surface during geological storage in salt is limited to 482°F (250°C); salt temperatures above 250°C may result in localized fracture of the salt (Appendix C).

4. Internal Pressure

The maximum internal canister pressure is assumed to be equal to or less than 100 psi under normal life cycle conditions at a wall temperature of 650°F for analysis purposes. An atmospheric pressure of 14.7 psia at 80°F during closure, by Boyle's Law on behavior of gases, will rise to 30 psia at 650°F. Therefore, the value of 100 psi used for analysis is considered to be conservative.

5. Welding Thermal Load

The high thermal gradients emanating from the weld zones during canister closure welding cause yielding in the surrounding material. After the canister has cooled, a high residual stress is formed near the closure seal weld. The yield strength of the material will limit the maximum residual stress in the vicinity of the weld; however, subsequent creep of the material tends to relieve this residual stress.

6. Thermal and Pressure Excursion

Rather than postulating a specific pre-determined pressure and thermal excursion criteria, reasonable limits for excursions which the canister can withstand without loss of structural integrity will be determined.

7. Earthquakes

The maximum horizontal ground acceleration is 0.25g for the Safe Shutdown Earthquake (SSE), and 0.125g for the Operating Basis Earthquake (OBE). The design response spectra used in this study is taken from NRC Regulatory Guide 1.60. No soil amplification factors are included.

8. Thermal Shock

When quenching the canister in water, as discussed in Section III A.2, an initial canister surface temperature of 650°F and a water temperature of 120°F are assumed for reference temperatures.

9. Drop

The following accident drops of this canister during its life cycle were considered: (1) during transfer from reprocessing to interim storage, (2) during handling in the surface facility at terminal geologic storage complex, and (3) during transfer down the mine shaft. In the first two incidents, a free fall of 20 feet to a concrete floor is assumed. During transfer down the mine shaft, a 2,000-feet free fall and hard surface impact are assumed.

IV. DESCRIPTION OF CANISTER MODELING

A. Canister Configuration

The basic model for the structural analysis of the waste canister is a 12-inch diameter Schedule 40 pipe wall canister (equivalent to ASTM designation "Standard Pipe"). Standard weight pipe was also used for models of 8-inch and 24-inch diameter canisters.

The canister basically consists of the following three structural parts:

1. Thin wall cylinder.
2. Bottom semi-ellipsodal head with skirt and base plate.
3. Top semi-ellipsodal head with threaded opening and plug.

Three different canister diameters, 8-inch, 12-inch and 24-inch, and two different lengths, 10-foot and 16-foot have been studied. The dimensions of the basic model 12-inch canister used for the analysis are shown in Figure 1. The ratio of the minor to the major axes of the ellipsodal heads is 0.5. The 8-inch and 24-inch diameter canisters have shapes similar to that of the 12-inch canister; however, the ratios of the minor to the major axes of the heads are 0.5 and 0.27 respectively. The 24-inch canister is shown in Figure 2.

B. Material

During the study, two canister materials were analyzed. One analysis assumed the canister was fabricated entirely of type 304L stainless steel, and in the other analysis, a combination of materials with a type 304L body and type 410 stainless steel head. The latter material was postulated because the threads for the canister closure

Figure 1 - Twelve (12)-Inch Diameter Canister

CANISTER ELEVATION

SCALE: 1 1/2" = 1'-0"

SKIRT DETAIL (2)

SCALE: 1 1/2" = 1'-0"

BUTTRESS THREADS (TYPICAL)

DETAIL (1) NTS

PLUG DETAIL (3)

HALF FULL SIZE SCALE

Figure 2 - Twenty-Four (24)-Inch Diameter Canister

plug in a type 410 material would be less subject to galling than those machined in type 304L stainless steel.

C. Finite Element Model

The SAPIV finite element computer program was used for stress analysis of three finite element models of the canister(s). The canister model geometry was idealized as axisymmetric by neglecting the reinforcement at the bottom of the shell. The first model, shown in Figure 3, was used for parametric studies of varying diameters (8-inch, 12-inch and 24-inch) and canister head materials (Type 304L and 410). Relative coarse meshes including only one element across the shell wall were used, since only the primary general membrane stress, i.e. the average value across the thickness of the shell was of interest for comparison purposes. Primarily, a quadrilateral element was used for modeling; however, triangular elements were used at transition zones.

The second model for stress analysis, shown in Figure 4, consists of a rather fine mesh with three equal layers across the shell wall. Local stresses due to discontinuity effects are evaluated. Bending stresses at the surface of the shell are extrapolated by using the stresses at the center of the three elements across the thickness.

The third model, an isolated segment of the cylindrical wall, with ten layers across the thickness, is shown in Figure 5. This model was developed for use in determining the general behavior of thermal stresses when the hot canister is quenched in a water pool.

FIGURE 3 - FINITE ELEMENT MODEL 1 - COARSE MESH - HLW CANISTER

FIGURE 4 - FINITE ELEMENT MODEL 2-FINE MESH - HIW CANISTER

NOTE:

* For Section A, B and C see Fig. 7

FIGURE 6 - HEATING5 MODEL - HLW CANISTER

FIGURE 5 - FINITE ELEMENT MODEL 3-ONE SEGMENT OF CANISTER WALL

For heat transfer analysis, the HEATING5 computer program is used. The canister is modeled, as shown in Figure 6a and 6b, to calculate the steady temperature distribution across the head and bottom portions. Since only rectangular element arrays are acceptable in this computer program, a certain degree of approximation was used in interpreting the results for subsequent thermal stress analysis in the finite element model.

V. BASIC MODEL CANISTER STRESS DUE TO
NORMAL LOADING CONDITIONS

The structural analyses of a basic model 12-inch diameter by 10-feet long canister subjected to normal life cycle loading conditions are discussed in the following pages. Table 3 below shows the normal canister life cycle loads, load combinations, stress category, and the associated level of allowable code stress considered in the structural analysis of the canister.

TABLE 3
STRESS CATEGORIES AND ALLOWABLE STRESS

Load Combination	Stress Category	Allowable Stress
DL+LL+P = ML	Pm	Sm
	P1	1.5 Sm
	P1+Pb	1.5 Sm
ML+T	P1+Pb+Q	3 Sm
ML+T+WT	P1+Pb+Q+F	Sa

Where: DL = Dead Load
 LL = Live Load
 P = Internal pressure
 ML = Mechanical loads which include DL+LL+P
 T = Thermal load
 WT = Welding thermal load
 Pm = Primary general membrane stress

P_l = Primary local membrane stress

P_b = Component of primary bending stress

Q = Secondary membrane plus bending stress by
differential thermal expansion

F = Thermal stresses which may cause fatigue

S_m = Allowable stress at certain temperature

S_a = Allowable fatigue stress intensity

A. Mechanical Load - Dead Load, Live Load & Internal Pressure

The live load of the vitrified waste material was calculated based upon the canister being 80 percent full. This load is uniformly applied at the bottom shell.

The resulting stresses with the canister suspended from its lifting device or resting on its base are shown in line 1 to 3 of Table 4 in a format accommodating various stress categories of the code. The local membrane stresses, P_l , are the computer stress outputs for finite elements in a region of geometrical discontinuity. Under dead plus live, plus pressure loads, the maximum membrane plus bending stress, $P_l + P_b$, occurs in the bottom shell when the canister is at rest on its base, while the maximum stress occurs in the top shell when the canister is in a suspended mode. The overall maximum membrane stress, P_m , occurs at the bottom shell in either position because the live load consistently loads the bottom shell whether the canister is supported from above or below.

B. Heat Transfer Analysis

As discussed in Section III B, there is a temperature drop across the

TABLE 4

MAXIMUM STRESS VS. ALLOWABLE STRESS BY STRESS CATEGORY

Diameter: 12-Inch
 Length: 10 Feet
 Temperature: Gradient 1.8F
 Surface Temperature: 482°F
 Material: Type 304 Stainless Steel

Loading Condition	Stress Category	Canister Stress - KSI	
		Resting on Base	Suspended Position
1. DL + LL	P _l + P _b	1.82	3.13
2. P, 100psi	P _l + P _b	2.53	2.53
3. DL + LL + P	P _m	2.59/14.8 ⁽¹⁾	2.59/14.8
	P _l + P _b	4.04/22.2 ⁽²⁾	5.66/22.2
4. T	Q	0.62	0.62
5. DL + LL + P + T	P _l + P _b + Q	4.66/44.4 ⁽³⁾	6.28/44.4

NOTE: (1) Maximum stress/allowable stress for P_m at 482°F

(2) Maximum stress/allowable stress for P_l+P_b at 482°F

(3) Maximum stress/allowable stress for P_l+P_b+Q at 482°F

wall of the canister due to the normal internal heat generation caused by the decaying waste. At steady state conditions, the temperature distribution across the thin canister cylinder wall is a straight line. Due to the irregularity of the top and bottom heads, computer analyses were used to calculate the temperature distribution at these two locations. The finite element models of the top and bottom heads are shown in Figure 6a and 6b.

The temperature at the outside surface of the canister is 482°F (250°C) which is the maximum temperature permitted in salt storage. The temperature distribution across Section A-A and B-B of the top head portion of the canister is shown in Figures 7a and 7b. Figure 7c shows the temperature distribution at Section C-C across the bottom head. It should be noted that there is a large temperature drop across the air gap at the bottom of the canister due to the insulation of the air space provided by the skirt and base plate on the bottom of the canister. However, all three of the figures indicate that the temperature distribution does not vary linearly.

The temperature distributions determined by heat transfer computer analysis were used to perform thermal elastic analysis of the canister as discussed in the following paragraphs.

C. Thermal-Elastic Analysis

The results of the heat transfer analysis are used as input to the thermal-elastic finite element stress analysis performed by SAPIV. The finite element model uses refined meshes similar to those used for the heat transfer analysis. The modulus of elasticity is tem-

(a) Temperature Distribution Across Section A-A Of Top Head

(b) Temperature Distribution Across Section B-B Of Top Head

(c) Temperature Distribution Across Section C-C Of Bottom Head

FIGURE 7 - STEADY STATE TEMPERATURE DISTRIBUTION
OF 100°F GRADIENT ACROSS TOP AND BOTTOM
HEADS OF CANISTER

perature dependent while Poisson's ratio is assumed to be 0.3. The resulting thermal stresses caused by an assumed 100°F temperature gradient are shown in Table 5 in terms of maximum stresses at different sections of the canister and their respective locations. The absolute maximum stress occurs at the transition of the thin shell wall to the thick wall of the top semi-ellipsoidal head. The results of these analysis were then used to interpolate the thermal stresses of steady state temperature gradients other than 100°F.

D. Combination of Loadings

The resulting thermal stresses in the canister created by a 100°F temperature gradient across the wall are used to interpolate for the thermal stress of a 1.8°F steady state temperature gradient. The resulting thermal stresses are tabulated in line 4 of Table 4. The thermal stresses during this steady state condition are so small as to be negligible.

For combinations of mechanical loadings and thermal load shown in line 5, it should be noted that the stresses are well below allowable limits.

E. Welding Thermal Effects

It is assumed that the procedure and inspection of the welding will follow all applicable code requirements hence the thermal stresses due to the welding process in the canister material adjacent to the seal weld are assumed to be within allowable limits of the code. This type of thermal stress would cause cracking and subsequent failure of the seal weld only under numerous repeated loading cycles. It is

TABLE 5

CANISTER THERMAL ELASTIC ANALYSIS RESULTS
MAXIMUM STRESS AT 100°F THERMAL GRADIENT

Diameter: 12-Inches
Length: 10 Feet
Material: Type 304 L Whole Canister

CANISTER SECTION	MAXIMUM STRESS(1) KSI	LOCATION OF MAXIMUM STRESS
Top Semi-Ellipsodal Head	34.6	At transition from thick to thin wall
Straight Cylinder Wall	19.1	At inner and outside face of wall
Bottom Semi-Ellipsodal Head	20.7	At transition from straight cylinder to bottom head

NOTE:

(1) At a temperature range of 482°F to 582°F.

not expected that the canister will be subjected to repeated thermal stresses of this kind during its life cycle.

F. External Pressure

External pressure on the canister during storage in the salt is expected to be as high as 3500 psi. The allowable external pressure established in the ASME Boiler and Pressure Vessel Code, Section VIII - Division 2, which the 3/8-inch canister wall can withstand is 900 psi. The stiffening effects of the vitrified waste are neglected since the canister is only 80% filled. Since it is necessary to avoid failure of the canister during salt storage, alternative solutions to this problem would include increasing the wall thickness of the canister or providing a thick-wall sleeve into which the canister could be placed.

VI. BASIC MODEL CANISTER STRESS DUE TO
ABNORMAL AND IMPROBABLE EVENTS

A. Maximum Temperature Excursion

Canisters fabricated of Type 304L stainless steel are generally limited to a service temperature of 800°F by the ASME Boiler and Pressure Vessel Code, Section VIII, Division 2. For temperatures higher than 800°F, other types of stainless steel must be used.

Thermal excursions up to 690°F have been predicted if the canister mechanical cooling system(s) malfunctions. Further investigation of canister and shipping cask heat transfer mechanisms is required to verify this upper limit for thermal excursions.

B. Maximum Pressure Excursion

At the study design temperature of 650°F, the 3/8-inch wall of the basis model 12-inch diameter canister has a maximum permissible internal pressure of 800 psi. At the maximum allowable code temperature of 800°F for Type 304L stainless steel, the permissible internal pressure is approximately 750 psi.

The atmospheric pressure existing in the canister following closure may rise due to thermal excursions associated with the decay heat of the wastes in the canister and loss of canister cooling. At the design temperature of 650°F or an excursion temperature of 690°F, the absolute pressure in the canister would double assuming Boyle's Law for behavior of gases prevails. For purposes of the analysis, a conservative internal pressure of 100 psi at 650°F was assumed. This assumed pressure is above the actual two atmospheres of pressure expected to exist in the canister and well below the canister permissible pressure of 800 psi.

C. Drop

Accidental dropping of the canister is most likely to occur at three points during its life cycle: (1) during transfer from reprocessing to interim stage; (2) during surface handling at the terminal storage facility; and (3) during transfer down the mine shaft.

Two different approaches were used in analyzing canister drop in this study. First, the limit analysis method was used to evaluate plastic deformation of the canister; and a second, the impact force method was used to determine the maximum stress level (See Appendix A).

In the impact analyses (Appendix A), a simple model of the canister utilizing a continuous vertical shell was examined by limit analysis methods. For purposes of simplification, certain energy absorbing features of the canister design such as the bottom head, the joint between the skirt and canister, and axial deformation were not included in the model. The results of the model analysis revealed that a 1.23-inch canister wall is required to survive a 20 foot fall. However, a 1.23-inch wall would considerably increase the cost of each canister and, therefore, justifies further conceptual development of a more economical canister design.

The analysis of the simplified model indicates that buckling of the canister will occur so a more comprehensive analysis of the canister utilizing a sophisticated computer program, i.e. MARC-CDC, is required to ascertain if the canister may buckle without rupturing or whether the canister skirt would buckle first and protect the canister. The analytical model would include such energy absorbing features as the bottom head, the joint between the skirt and canister, and axial deformation.

A parametric study of various skirt configurations is also required to develop a skirt design that absorbs most of the kinetic energy of the impact and minimizes canister damage.

Both of the approaches used in this study show that the basic model canister (3/8-inch wall) will suffer severe damage from a free fall of 20 ft (See Appendix A for detailed analysis). In the case of the postulated 2000 feet free fall down the mine shaft and impact on a

hard surface, the canister can be expected to rupture.

D. Earthquake

The limit analysis method was used to assess the structural integrity of the canister during an earthquake accident (See Appendix B). Earthquake damage to the canister during its life cycle could occur during: (1) transfer and handling while suspended during interim storage at the reprocessing plant; and (2) surface facility handling at the terminal storage facility.

The results of the analysis indicate that the canister can survive horizontal impact against a hard surface while suspended due to the intensities of both the OBE at 0.125g and the SSE of 0.25g, without structural failure; although, the wall of the canister may be slightly dented.

F. Quenching in Water

The transient temperature-time history of the canister after plunging it into 120°F water was studied. The heat transfer coefficient at the exterior canister wall is expected to vary from 180 to 1000 BTU/hr/ft²/°F. Three different transient temperature-time histories have been calculated based on heat transfer coefficients of 180 BTU/hr/ft²/°F, and 1000 BTU/hr/ft²/°F, and variation with surface temperature. The transit temperature distribution across the canister wall for these three coefficients at the arbitrarily chosen instant of 3 seconds after immersion in the water are shown in Figure 8.

The circumferential stress in the canister with a heat transfer coefficient of 1000 BTU/hr/ft²/°F as determined by thermal elastic analysis

FIGURE 8 - TRANSIENT CANISTER WALL TEMPERATURES AT 3 SECONDS

is shown in Figure 9. The maximum thermal stress is 90 Ksi at the exterior wall surface which exceeds the code limits of 44.4 Ksi.

Therefore, quenching as a method of canister cooling is not acceptable.

According to the ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, 1971, Appendix 4, Article 4-1:

"If the thermal stress exceeds twice the yield strength of the material, the elastic analysis may be invalid and successive thermal cycles may produce incremental distortion".

In the case of quenching the canister in a pool of water, the results of the elastic analysis show that the maximum stress does exceed twice the yield strength and code allowable limit. These results suggest that the elastic analysis method is not acceptable. However, since the thermal stress is essentially transient, a detail non-linear analysis may prove that the procedure is applicable if (1) the residual stress is negligible, and (2) the distortion is not significant.

VII. BASIC MODEL ANALYTICAL RESULTS FOR CANISTER LIFE CYCLE STAGES

The structural integrity of the basic model 12-inch diameter by 10 ft. long canister is assessed below in the successive stages of its life cycle.

A. Waste Solidification

1. Normal Conditions

In this stage, three different postures of the canister are included, i.e. during filling with waste, closure seal welding, and transfer to interim storage. During filling with "molten glass" waste at approximately 1832°F, it is assumed that the canister is cooled by an external cooling system to keep the canister tempera-

FIGURE 9 - CIRCUMFERENTIAL THERMAL STRESS IN CANISTER WALL DURING TRANSIENT STATE AT 3 SECONDS

ture below the code allowable of 800°F and to preclude excessive residual hoop stress in the canister during cooling caused by the different coefficients of expansion of the stainless steel canister and the waste.

The stresses resulting from dead plus live load, thermal load, internal pressure and their combinations are shown in Table 6. These stress levels are all well below the allowable code limit. During the seal welding operations some regions near the weldment are expected to reach the yield stress of the material. Nevertheless, this type of high stress is only vulnerable to cyclic loading which is not expected to occur during the life of the canister. If the procedures of welding and inspection conform to the code requirements, cracking or failure of the weld joint is not expected.

2. Abnormal and Improbable Events

Loss of transfer cask cooling* is not expected to cause the canister temperature to exceed the maximum service temperature of 800°F allowed by the code for Type 304L stainless steel.

Neither the operating basis earthquake (OBE) nor the safe shutdown earthquake (SSE) will damage the canister shell beyond its ultimate strength. Cracking at the non-structural seal weld due to earthquake is not expected but may require verification by prototype demonstration testing.

* Telecon w/SAI 9-23-77, preliminary heat transfer calculations indicate canister temperature will not exceed 690°F.

TABLE 6

MAXIMUM STRESSES DURING SOLIDIFICATION

Incident Canister Posture	NORMAL LOADING CONDITIONS - STRESS IN KSI					
	(1) Dead & Live Load	(2) Thermal Load	(3) Internal Pressure	(4) Welding Thermal	(1)+(3)	(1)+(2)+(3)
During Fill	1.82	0.62			$\frac{1.82}{20.6^*}$	$\frac{2.44}{41.1^{**}}$
During Closure	1.82	0.62	2.53	Residual Yield Stress	$\frac{4.04}{20.6}$	$\frac{4.66}{41.1}$
During Transfer	3.13	0.62	2.53		$\frac{5.66}{20.6}$	$\frac{6.28}{41.1}$

NOTE: *Allowable Stress 1.5 Sm for Pl+Pb at 650°F

**Allowable Stress 3 Sm for Pl+Pb+Q at 650°F

The maximum pressure excursion will not exceed the allowable code limits. (See Section VI B) However, a 20 ft. drop of the canister to the floor can severely damage the canister (See Appendix A).

B. Interim Storage at Fuel Reprocessing Plant Stage

1. Normal Conditions

The stresses in the canister under normal loading conditions are found to be within code allowable limits. The stress level is similar to that during waste solidification with the canister resting on its base.

2. Abnormal Events

Plunging the canister into a water pool to cool it following loss of the canister cooling system would result in a thermal stress buildup exceeding the allowable code limit.

The thermal and pressure excursion associated with loss of an external cooling system will not exceed the allowable stress limits of the canister.

C. Shipping Stage

1. Normal Conditions

The integrity of the canister structure is not jeopardized by the normal loadings imposed on the canister during shipping and the stress levels are similar to those previously mentioned in Table 6.

2. Abnormal Events

The canister temperature due to loss of cask cooling is expected to rise as high as 690°F* during shipping but is still within the service temperature. The maximum pressure excursion is not expected to exceed the design pressure limit of the canister discussed in Section VI B.

D. Geological Storage Stage

1. Normal Conditions

During receipt of the canister at terminal storage facility and transfer into the mine, the stress levels in the canister are similar to those in other normal life cycle handling operations and well below the allowable code limit. When the canister is stored in the salt mine, the external pressure of the salt on the canister is reported to be as high as 3500 psi and, therefore, beyond the 900 psi limit of this canister. Furthermore, stainless steel is susceptible to pitting corrosion and chloride stress cracking if buried in salt in the presence of moisture.

2. Abnormal and Improbable Events

The thermal and pressure excursions during transfer of the canister to the mine pose the same problems as described for the shipping stage.

The drop impact of 20 feet to the concrete floor will damage the canister. A special design to mitigate canister damage due to the

* Telecon w/SAI 9-23-77, preliminary heat transfer calculations indicate canister temperature will not exceed 690°F following loss of shipping cask cooling.

20 feet drop accident should be developed. The drop impact of 2000 feet through the mine shaft is expected to cause failure of the canister. Protective measures in the design of the shaft and canister handling system will be required to preclude this improbable event.

Earthquake intensities of 0.125g and 0.25g for the OBE and SSE, respectively, should not damage the canister beyond its ultimate strength.

VIII. PARAMETRIC STUDIES

The effects of changing canister diameters, lengths, and materials are discussed in this section. In addition to the previously discussed 12-inch diameter canister, 8-inch and 24-inch diameter canisters are also analyzed for the normal loading conditions. Two different lengths, 10 feet and 16 feet, for each diameter are studied. The materials evaluated in determining the effects of different materials of construction were Types 304L and 410 stainless steel.

A. Effects of Varying Canister Diameter - Normal Loading Conditions

The analysis of various diameters of 10 feet long canisters under normal loading conditions is summarized in Table 7 in terms of maximum stress vs. allowable stress. The material is assumed to be Type 304L for the entire canister.

The canister posture is assumed to be in either the handling mode (suspended by its plug) or resting on its base. The maximum primary membrane stress P_m , which is the average value across the wall, is found to be the same for both positions.

TABLE 7

MAXIMUM STRESS IN VARIOUS DIAMETER CANISTERS - NORMAL LOADING CONDITIONS**

Diameter: 8-Inch, 12-Inch and 24-Inch
 Length: 10 Feet
 Material: Type 304L
 Temperature: 650°F

Loadings Conditions	Maximum Stress in KIPS Per Square Inch - KSI		
	Canister Diameter - Inches		
	8"	12"	24"
1. Temperature Q	0.0	0.0	0.0
2. Pressure, Pm 100 psi	1.04	1.63	6.4
3. Gravity Load (DL.+L.L.) Pm	0.94	1.55	5.7
4. - Combination (2) + (3)	$\frac{1.98}{13.7}^*$	$\frac{3.18}{13.7}^*$	$\frac{12.2}{13.7}^*$

NOTE: *Maximum stress/allowable stress, Sm, for Pm at 650°F

**Canister either resting on base or in suspended handling position

For ease of comparison by elimination of variables, the wall temperature is assumed to be 650°F for equilibrium temperature across the wall. The uniform expansion of the material exerts virtually no stress in the canister. The stress levels under combined loadings are small in comparison with code allowable limits, except in the case of the 24-inch diameter canister which reaches about 90% of the allowable ASME code limit.

B. Effects of Canister Length - Normal Loading Conditions

The stress in the canister due to temperature and internal pressure is not affected by the length of canister. However, the maximum stress under gravity loading is directly proportional to the length of canister. Although, the maximum combined stress for the 16-foot long canister is still within allowable limits for both the 8-inch and 12-inch diameter canisters, the allowable stress is exceeded by 13% in the bottom head of the 24-inch diameter by 16-foot long canister under the combination of pressure and gravity loading at the temperature of 650°F.

C. Effects of Canister Diameter and Length - Abnormal and Improbable Events

Additional parametric investigations of 8-inch and 24-inch canisters indicate that the stress levels resulting from abnormal and improbable events of thermal and pressure excursions are acceptable (within code allowable stress) for all six canister configurations studied except for a 24-inch diameter by 16-foot long canister in the case of a pressure excursion. The analysis indicates that the 24-inch diameter by 16-foot long bottom head is overstressed by the combination of

loadings indicated in Table II, using a standard pipe model. However, the canister stress would be within code limits up to a pressure of 70 psi. In the event of an operating basis earthquake (OBE) (0.125g), a calculated permanent deformation or denting of the canister of 0.01-inches would occur at the point of impact. The deformation caused by a safe shutdown earthquake of 0.25g is calculated to be 0.02-inches. Assuming a single impact occurrence at these maximum accelerations, the integrity of the canister should remain intact when subjected to either an OBE or SSE.

Stress levels resulting from abnormal and improbable events of thermal shock, a drop of 20 feet onto a hard surface, and a mine shaft drop are not acceptable for any of the six configurations.

D. Effects of Different Materials (Type 304L and Type 410)

In order to avoid possible galling at the threaded plug opening, using Type 410 stainless steel for the head of the canister while using Type 304L for the rest of the canister was evaluated. The results of the evaluation show that the stresses under mechanical loads are virtually the same for the two materials due to the closeness of their modulus of elasticity. However, high thermal stresses are generated because of the significant difference in the thermal expansion coefficient of each material. The analysis assumes 650°F across the canister wall without temperature gradients. The maximum thermal stresses for three different diameters are shown on Table 8. The maximum thermal stress is a function of the geometry from the thin cylindrical shell to the thick wall head. Among the arbitrarily chosen geometries of the transition curves, the 12-inch diameter canister has the highest

TABLE 8

MAXIMUM THERMAL STRESS IN CANISTERS - USING DIFFERENT MATERIALS

Head: 410SS
Shell: Type 304LSS
Temperature: 650°F

T H E R M A L S T R E S S * Q, Ksi		
8-Inch	12-Inch	24-Inch
27.7 30.4	32.9 30.4	24.3 30.4

NOTE: * Maximum thermal stress/code limitation for
elastic analysis at 650°F

thermal stress. The high thermal stress is near or exceeds the code limit and if combined with stresses imposed by other loadings may induce possible distortion of the structure; therefore, using different materials at the top portion and shell is not considered to be practical.

E. Canister Handling Limitations

The diameter and length of canisters considered in this study do not impose prohibitive restraints on the handling, transport, storage systems and hardware utilized in each of the canister life cycle stages. As the diameter, length, and weight of the canister increase, the number of canisters to be handled will decrease. However, the facilities and equipment provided in each life cycle stage for handling the canister must be sized accordingly.

As the canister dimensions increase, the fixed capital investment will also increase to accommodate the larger canisters. For example, the problems of transporting a 16-foot long high-level waste canister through the corridors of an underground mine to a canister burial room may be compared with those currently under conceptual development by DOE-OWI for transporting a 16-foot long spent unprocessed fuel canister to a burial room. This conceptual design concludes that the technical problems may be resolved in the mine shaft and transporter design and in the dimensions required for traffic in the mine corridors and burial rooms. However, the cost of the additional excavation to accommodate larger canisters is considerable and must be evaluated in the final sizing of waste canisters.

APPENDIX A
IMPACT STUDY OF THE HIGH-LEVEL WASTE CANISTER

I. INTRODUCTION	1
II. PROBLEM BY DEFINITION	1
III. REVIEW OF PREVIOUS WORKS	2
IV. STUDY APPROACH	3
V. LIMIT ANALYSIS	5
VI. ALTERNATE ANALYSIS	6
VII. OBSERVATIONS AND DISCUSSIONS	7
VIII. CONCLUSIONS AND RECOMMENDATIONS	10
IX. APPENDICES	
A-1. LIMIT ANALYSIS (CALCULATIONS)	13
A-2. ALTERNATIVE STRESS ANALYSIS (CALCULATIONS)	19
A-2-1 NORMAL IMPACT	19
A-2-2 ROTATING IMPACT	23
A-3 SUMMARY FORMULAS USED IN DENG'S REPORT	28
A-4-1 DYNAMIC LOAD FACTOR	29
A-4-2 NATURAL MODE SHAPES AND FREQUENCIES	30
A-5 REFERENCES	31

Table of Figures

<u>Figure</u>		<u>Page</u>
A-1	Possible Drop Postures	35
A-2	Probable Failure Mechanism (Normal Impact) (Case 1)	36
A-3	Probable Failure Mechanism (Oblique Impact) (Case 2)	37
A-4	Probable Failure Mechanism (Rotating Impact) (Case 2B)	38
A-5	Probable Failure Mechanism (Parallel Impact) (Cases (3) & (4))	39
A-6	Impact Pressure Time History for 207 lb, 8 in., Schedule 40 Pipe Missile Impacting 5000 lb/in ² Concrete at 320 ft/sec (After Kennedy, Ref. (12))	40
A-7	Comparison of Numerical and Analytical Results in the Case of Impact of a Homogeneous Linear-elastic Bar on a Rigid Wall. (After Drittle, Ref. (10))	40
A-8	Typical Stress-strain Diagram for Ductile Stainless Steel	41
A-9	Postulated Deformed Circular Ring (Parallel Impact)	42
A-10	Idealized Impact Forcing Function (Normal Impact)	43
A-11	Rotating Impact on Another Canister	44
A-12	Postulated Failure Mechanism, Limit Analysis (Normal Impact)	45

APPENDIX A

"IMPACT STUDY OF THE HIGH-LEVEL WASTE CANISTER"

I. INTRODUCTION

This study considers the integrity of a 12-inch diameter by 10 feet long by 3/8-inch thick shell high-level waste canister following various drop-impact conditions during its life cycle. A drop height of 20 feet with the canister landing on a concrete floor was first studied. Four possible drop postures have been identified; of these, three extreme cases were studied. An abnormal drop height of 2,000 feet (encountered during the transfer at salt mine) was then evaluated.

Two different analytical approaches were employed in this study. First the limit analysis or energy conservation method as proposed in the preliminary study of the project (Ref. 24) was applied, then the impact force method in association with the stress and deformation analysis was used as an alternative approach. Conclusions based upon observations of calculations have been drawn and are presented in Section VIII together with recommendations for mitigation of the abnormal-impact disasters.

II. PROBLEM DEFINITION

The problem of accidental dropping of the canister during its life cycle involves several possible impact situations. The life cycle stages considered in this study include the following: (1) during transfer from a fuel reprocessing plant to an interim storage facility; (2) during handling while in interim storage; (3) during surface reception and handling at terminal storage; and (4) during transfer into the mine.

Possible drop incidents include the following: Case 1 - Dropping the canister in a position perpendicular to the ground from an arbitrary height, subsequent rebounding (the amount of rebounding depends upon the coefficient of restitution of the colliding materials) followed by the canister falling onto its side on the ground surface; Case 2 - Dropping the canister in a position oblique to the ground from an arbitrary height followed by the canister toppling over and striking another canister with its tip; Case 3 - Dropping the canister in a position parallel to the ground from an arbitrary height with the canister landing flat on the ground; and Case 4 - Dropping the canister in a position parallel to the ground from an arbitrary height with the canister landing on, and perpendicular to, another canister resting on its side. (Figure A-1). The possible failure mechanisms resulting from these situations are illustrated in Figures A-2 through A-5.

In reviewing the possible drop cases and their corresponding failure mechanisms as described above, it is apparent that each case represents a different impact problem. However, Cases 1, 2, and 4 have been identified as extreme situations, and selected for further study.

III. REVIEW OF PREVIOUS WORK

The problem of impact loading on a structure has long been of interest in the field of aircraft design and the structural safety assessment of nuclear reactor containment facilities. Theoretical developments on the impact loading function and experimental correlation study on structural responses, have been well established (Refs. 2 - 21). Although the basic purpose was to determine the structural response of the containment, the concept of generating the impact forcing function can be employed in this study. Among these, a simple yet reasonable approach to estimate the impact force given by Kennedy (Ref (12)) is of most interest. Accordingly,

the impact force can be estimated by a NDRC empirical formula which is a function of impact velocity, projectile size and shape, concrete penetrability factor, as well as impact contact area. Note that the impact forces generated for the non-deformable missile and for the deformable missile, Figure A-6, are essentially the upper and lower bounds that a finite-deformable missile will expect to experience.

Another simple approach in estimating the impact force is by the principle of conservation of momentum. A rather conservative estimate of the impact period can be determined from twice the required travelling time of the compressive stress wave along the total length of the canister. The approximate constant impact pressure can then be obtained by the total change of momentum. Drittle (Ref. (10)) substantiated this simple concept in his study by drop testing a solid elastic bar (Fig. A-7).

The preliminary impact study (Ref. 24) made by Dr. Daniel Deng employed the general concept of limit analysis in association with the theorem of conservation of energy. A simple model developed to examine the safety of the canister subjected to impact loading was based on the admissible plastic strain energy of the postulated failing canister being at least equal to the input kinetic energy. Although this rational analysis is simple and rough, it is believed to be an adequate approach for study purposes. However, further impact experiments are needed to substantiate this concept.

IV. STUDY APPROACH

It is possible to generate a reasonable estimate of forcing function by rigorous mathematical manipulations (Refs. (9), (10), (11), (12), (13)).

However, the experimental results demonstrated by several investigations (Refs. (12), (10), (16)) reveal that the envelope of the impact time history can be well defined by simple mechanical assumptions. For example, the two simple approaches mentioned in Section III have been reasonably justified (Refs. (12), (10), (16)) and used extensively in preliminary study of the impact problem.

Our primary interest is the investigation of the waste canister's integrity under the abnormal condition of a free fall. The two approaches mentioned above were used in order to examine the order of magnitude of the impact force and the resulting deformation under different types of free fall. Note that the calculations based on these approaches will yield a reasonable estimate bounded by the ideally perfect elastic and plastic cases (Fig. A-6)).

Once the impact force (either an extreme value or average pressure) has been generated, the typical closed-form solutions for circular rings can be used to determine the resulting stresses and deformations. Unless the solutions are within the elastic limit, the postulates behind these formulas are not valid. Further study of inelastic buckling (or large deformation analysis) is necessary.

The second approach used in this study is the application of Deng's formulas. Normal impact and parallel impact are studied by limit analysis in conjunction with the theory of conservation of energy. However, caution should be used in assessing and interpreting the results since the method involves some simplifying assumptions.

Observations and conclusions based on the results of the calculations have been drawn and are reported in Sections VII and VIII.

V. LIMIT ANALYSIS (24) (20 Ft. Drop)

The complete calculations are provided in Appendix A-1. An abstract of the results for a canister with a 3/8" thick wall follows:

A. Case 1 - Normal Impact: (longitudinal direction)

1. Kinetic energy input = 756.96^{k-11} > Plastic energy
= 177.24^{k-11}

2. Maximum plastic deformation along the postulated plastic hinge zones $\theta = 235^\circ$; i.e., the compressed zone at the bottom of the canister is completely flattened. In other words, the buckled half-wave length is zero.

3. Minimum wall thickness of the canister to sustain the normal impact of 20' drop height

$$t_{\min} \doteq 1.233''$$

B. Case 4 - Parallel Impact: (transverse direction)

1. Total plastic deformation of the circular ring $\theta = 205.6^\circ$
Physical meaning = the circular ring is completely flattened.

C. Case 2 - Oblique Impact: No calculation

VI. ALTERNATE ANALYSIS

Two impact postures have been studied by the alternative approach in which the approximate impact forces and impact times are estimated. The resulting stresses and deformations are estimated based on the elastic analysis of circular ring. Abstracts of the calculations in Appendix A-2 follow:

A. Case 1 - Normal Impact

1. Impact forces

NDRC approach: Total impact force, $P \doteq 5,380$ Kips

Average impact pressure $p_i \doteq 364.5$ Ksi

Rational approach:

Non-deformable canister - Extreme impact force
 $\doteq 8,256$ Kips

Deformable canister - Average impact force
 $\doteq 5,336.2$ Kips

Both NDRC and rational approaches to determine the impact force result in a very close estimation.

2. Impact Period: $t_1 \doteq 0.001317$ sec.

3. Elastic buckling load: $P'_{cr} \doteq 2,086.8$ kips; $p'_{cr} \doteq 141.4$ ksi
 $> f_y = 25$ ksi

4. Inelastic buckling load: $P'_{cr} \doteq 1,164.3$ kips; $p'_{cr} \doteq 78.89$ Ksi
 $> f_y = 25$ Ksi

5. Observation:

($P_{\text{impact}} = 5380$ kips) $>$ (Elastic buckling load, $P'_{cr} = 2,086.8$ Kips)

$>$ (Inelastic buckling load, $P'_{cr} = 1164.3$ Kips)

B. Case 4 - Rotational Impact and Striking Another Canister

1. Impact force

Extreme (deformable), $P_I \doteq 329.55$ kips

2. Resulting stresses

Max. principal stress = 260.85 ksi

Mix. principal stress = 74.85 ksi

Max. shear stress = 167.85 ksi

3. Observation

Max. shear stress = 167.85 ksi can be critical to a ductile material such as 304L stainless steel.

VII. OBSERVATIONS AND DISCUSSIONS

Based on the study and calculations performed, the following observations are made:

1. The limit analysis seems to be an appropriate short-cut approach to examine the structural integrity of the canister under impact loading. The stresses and deformations resulting from the estimated impact forces appear to have substantiated the validity of this approach.
2. The impact force and impact time estimated by two approaches, namely, the NDRC empirical method and the conservation of momentum method, are in fair agreement. Thus, either approach gives a reasonable estimate of impact force and time history which would be suitable for use in future inelastic computer analysis.

3. The basic assumption made in limit analysis is a maximum allowable strain (0.1 in/in) of hoop extension.

A typical stress-strain curve for steel (see Fig. A-8) can be used to examine the validity of this assumption. Typically, the yield strain of the steel ϵ_y equals $1300 \mu\epsilon$ (0.00130 in/in) and the hardening strain ϵ_{sh} roughly equals 0.13 in/in (100 times ϵ_y). Hence, it is conservative to assume a maximum allowable strain equal to (0.1 in/in) for the purpose of maintaining canister integrity.

For each of the drop situations, the resulting strain has been found to be greater than the rupture strain ($\epsilon_f \doteq 0.5 \text{ in/in}$). Hence, the impact force causes buckling of the canister and which, in turn, can result in brittle rupture at the impact zone.

4. In each case of the three impact conditions, the resulting impact force is greater than the elastic limit capacity of the material used ($f_y = 25 \text{ ksi}$, per ASME Boiler & Pressure Vessel Code Section VIII Division 2, 1971). Additional study using a sophisticated computer program (e.g. MARC-CDC) is required for further examination of the buckling behavior to determine whether buckling will also result in canister failure.
5. The above discussions and calculations are based on a quasi-static loading condition. The dynamic effect due to impact should be amplified by a dynamic load factor (DLF) (See Appendix A-4). For example, in the case of normal impact (triangular or half sine wave loading function), $2 > \text{DLF} > 1$. Hence, the dynamic effects of impact

on the canister will be worse than the pseudo-static ones used in elastic analysis.

6. The influence of the solidified waste material inside the canister as structural reinforcement and resistance to impact damage has not been considered in this study. Moreover, the secondary effects (relatively speaking) of pressure excursion, thermal excursion, as well as the thermal effect on the mechanical properties (E , ν , f_y , f_y etc.) have not been considered in this impact analysis.
7. A rotational impact caused by canister toppling may result in two critical conditions:

- a. Local damage of the canister shell near the head zone
- b. Local damage of the seal weld

Similarly, a normal impact occurring when the canister lands in a vertical position may result in:

- a. Local damage of the skirt at the bottom of the canister
- b. Damage of the skirt as well as buckling of the canister shell

Finally, a parallel impact caused by the canister landing in a horizontal position upon a second canister will damage the canister shell in its mid-point section.

8. The most probable impact case is that of oblique impact. Theoretically, the resulting impact force and stress are both infinite since the contact area during impact is very small. Therefore, only localized damage to the skirt is to be expected. The corresponding

impact effect to the canister shell depends totally upon the design and fabrication (weld) of the skirt (per ASME Section VIII, Page 139, 1971).

9. A minimum wall thickness of 1.23 inches, determined by limit analysis, is required to sustain a normal impact. This may be undesirable for cost reasons.

It is, therefore, desirable to perform a more rigorous analysis to attempt to optimize a skirt configuration which may lessen canister wall thickness requirements.

10. The stiffening effect of the convex bottom of the canister body has not been considered in the study in order to develop a simplified model adaptable to manual calculations and obtain first order of magnitude effect of impact on the canister. Moreover, the plastic strain energy contributed by the axial force for the case of normal impact has not been considered in the limit analysis model.

VIII. CONCLUSIONS AND RECOMMENDATIONS

1. Limit analysis and the alternate impact force analysis indicate the probability of inelastic buckling of the canister if subjected to a free fall of 20 ft to a concrete floor. Thus,
 - a. Probable local damage of the bottom skirt results from normal and oblique impacts (Cases 1 and 2);
 - b. Probable inelastic buckling of the canister shell results from both the normal impact and the oblique impact (Cases 1 and 2);

- c. Probable local buckling or damage at the mid-point results from an impact of a canister falling in a horizontal position upon a second horizontal canister (Case 4);
 - d. Probable local buckling or damage in the head-neck zone results from a rotating impact (toppling) (Case 2,B); and
 - e. Probable damage to or leaking from the seal weld results from a rotating impact (toppling) and striking of another canister (Case 2,B).
2. Large deformation or inelastic buckling may result in each of the impact situations. A further investigation by a sophisticated computer program, for instance, MARC-CDC, is necessary to substantiate these observations.
- Furthermore, a drop experiment to corroborate these theoretical studies is necessary. This is essential to confirm the structural integrity after impact.
3. The result of a free fall of 2,000 ft which might occur during transfer to the salt mine storage horizon can be understood intuitively since the stress of impact is proportional to the terminal velocity. The drop velocity increases to 360 ft./sec. (ten times that of the terminal velocity for a 20 ft. drop) and the kinetic energy is increased 100 times. The canister will

suffer severe or complete damage upon impact at the bottom of the shaft. Canister damage may also occur due to impact on the walls of the shaft during the fall.

4. If the possibility of an accidental drop is unavoidable, a special design of energy absorption devices is necessary. It is possible to design a skirt for absorption of part or all of the input kinetic energy. Also, a removable cushion attached to the canister during the period of handling may be a feasible solution. However, these suggested energy absorption devices would not be adequate in avoiding the possible impact disaster of a 2000 ft. free fall down the mine shaft.
5. The wall thickness of the skirt should be at most equal to that of the canister body. The essence of the skirt design as an energy absorption device should be based on the concept that plastic failure will be forced to take place in the skirt so that the residual kinetic energy transmitted to the canister body is minimum.

APPENDIX A-1. CALCULATIONS OF LIMIT ANALYSIS

(A.) Design Data

Unit Wt. of steel shell $W_s = 500 \text{ lb/ft}^3 = 0.0002894 \text{ Kips/in}^3$

Unit Wt. of glass $W_g = 187 \text{ lb/ft}^3 = 0.0001824 \text{ Kips/in}^3$

Young's Modulus $E = 25,700 \text{ Ksi}$

Mass density $\rho = 0.000000734 \text{ Kips-sec}^2/\text{inch}$

Canister Dimensions

Outer Diameter $d_o = 12.750''$; Area $A = 14.759 \text{ in}^2$

Inner Diameter $d_i = 12.000''$;

Shell Thickness $t = 3/8'' = 0.375''$

Total length (including skirt) $L = 126.380''$

Steel Yield Strength
(304L Stainless Steel) $f_y = 25.0 \text{ Ksi}$ (per ASME Sec. II 1971)

Steel Tensile Strength $f_t = 70.0 \text{ Ksi}$ (per ASME Sec. II 1971)

Drop height $h = 20'0''$

Total weight of filled canister $= (0.0002894) [(14.759 \text{ in}^2)(126'') + \pi(3'')^2(4.38'')]$
 $+ (0.0001824 * \pi * 6^2) \times 124'' = 3.154 \text{ Kips}$

(B.) Kinetic Energy of Canister Before Impact

Impact Velocity $V_1 = \sqrt{2gh} = \sqrt{2(384)(240)} = 430. \text{ inch/sec}$

Kinetic Energy = Potential Energy = $W * h$

$= 3.154^k \times 240'' = 756.96 \text{ k-''}$

(C) Final Distance Traveled by the Plastic Wave Front

$$\frac{\rho v_1^2}{f_y} = \frac{(0.000000734)(430.7)^2}{(25.0)} = 0.005446$$

Solving the initial condition ϵ_1 from

$$\frac{\epsilon_1^2}{1-\epsilon_1} = \frac{\rho v_1^2}{f_y} \quad (\text{Eq. 7})$$

$$\text{or, } \frac{\epsilon_1^2}{1-\epsilon_1} = 0.005446$$

$$\epsilon_1 = \frac{-5.446 \pm \sqrt{(5.446)^2 + 4(5.446)(1000)}}{2(1000)} = 0.071124$$

The "unstressed" rear portion of the canister can be obtained by solving the equation (6). Thus,

$$\ln \left[\frac{\bar{x}_f}{L} \right]^2 = 1 - \frac{1}{1-\epsilon_1} + \ln (1-\epsilon_1)$$

or,

$$\ln \left[\frac{\bar{x}_f}{126.38} \right]^2 = 1 - \frac{1}{1-0.071124} + \ln (1-0.071124)$$

or,

$$\bar{x}_f = 0.9275373 \quad L = 117.226''$$

The "stressed" portion is the impact contact face to the plastic wave front. Thus,

$$\bar{h}_f = 126.38'' - 117.226'' = 9.15''$$

(D) Maximum Allowable Hoop Elongation

$$\Delta \ell = 0.1 \pi (d) = (0.1)(\pi)(12.375") = 3.8877"$$

(E) Angle of Rotation (θ)

$$\theta = \sin^{-1} \left(\frac{d_A - d}{hf/2} \right)$$

$$\theta = \sin^{-1} \left(\frac{(0.1)(12.375")}{9.150"} \right) = 7.77^\circ = 0.1356 \text{ rad.}$$

(F) Plastic Strain Energy W_1 Contributed by Rotation of Limit Bending Moment (M_L)

$$M_L = f_y \times \frac{t^2}{4} = (25.0) * \frac{(0.375)^2}{4} = 0.878905 \text{ k-in/in}$$

$$\begin{aligned} W_1 &= 2 M_L \theta \pi (d + d_A) \quad \text{Eq. (9)} \\ &= 2 (0.878905)(0.1396)(\pi) (12.375" + 1.2375") \\ &= 10.49 \text{ k-in} \end{aligned}$$

(G) Plastic Strain Energy W_2 contributed by Hoop Tension

$$\begin{aligned} W_2 &= f_y \times t \times \Delta \ell \times \frac{\pi hf}{2} \\ &= (25.0)(0.375") * 3.8877" \times \frac{9.150"}{2} \\ &= 166.75 \text{ K-"} \end{aligned}$$

(H) Determination of Strength of Canister to Resist Impact

Total plastic strain energy capacity of the canister

$$S.E. = W_1 + W_2 = 10.49 + 166.75 = 177.24 \text{ K-"}'$$

$$(K.E. = 756.96 \text{ K-"}') > (S.E. = 177.24 \text{ K-"}')$$

(I) Determination of Maximum Plastic Deformation Due to

NORMAL IMPACT (Case 1)

Let K.E. = S.E.

Where S.E. = $W_1 + W_2$

$$\begin{aligned} &= 2M_L \theta \pi (d+d_A) + f_y \times t \times \Delta l \times \frac{\bar{h}_f}{2} \\ &= 2(0.8789.5)(\theta)(\pi)(12.375 + (1.1)(12.375)) \\ &\quad + (25.0)(0.375) * 3.8877 * \frac{9.154}{2} \\ &= 143.52\theta + 166.819^{\text{k-in}} \end{aligned} \quad \text{----- (a)}$$

$$\text{and K.E.} = 756.96^{\text{k-in}} \quad \text{----- (b)}$$

Equating (a) and (b) yields

$$\begin{aligned} \theta &= \frac{756.96 - 166.819}{143.52} = 4.112 \text{ Radians} \\ &= 235.6^\circ \end{aligned}$$

The result of θ indicates that a complete fold of the compressed zone (see Figure A-12) may result from the excessive kinetic energy input. Consequently, the inelastic buckling of the canister shell along the postulated plastic hinge lines is expected.

(J) Determination of Minimum Thickness of the Canister to Sustain the Normal Impact.

The canister can sustain the normal impact as long as the plastic strain energy generated through the admissible plastic deformation equal to or larger than the input impact Kinetic energy. Thus,

$$\begin{aligned} \text{KE} &= \text{SE} \\ \text{or KE} &= \left\{ \frac{1}{2} (f_y \times t^2) \theta \pi (d+d_A) \right\} + \left\{ f_y \times t \times \Delta l \frac{\bar{h}_f}{2} \right\} \end{aligned} \quad \text{----- (c)}$$

$$\text{or, } 756.96 \text{ k-in} = \left\{ \frac{1}{2} (25.0)(t^2)(0.1356) \pi (12.375 + 12.375 \times 1.1) \right\} \\ + \left\{ 25.0 \times 3.8877 \times \frac{9.150}{2} \right\}$$

$$137.375t^2 + 444.66t^2 - 756.96 = 0$$

$$t = \underline{1.233 \text{ inches}}$$

Parallel Impact - Longitudinal axis parallel to the target surface (Case 3)

(A) Kinematic energy of canister before impact

$$\text{K.E.} = \frac{1}{2} \frac{W}{g} v_c^2 = W \cdot h = 3.154^k \times 240'' = 756.96 \text{ k-in} \\ (\text{per Case (1), p. 13})$$

(B) Plastic Strain Energy W_1 , Contributed by Plastic Rotation of Bending Moment (M_L)

$$M_L = f_y \times \frac{t^2}{4} = (25.0) \times \frac{(0.375)^2}{4} = 0.87890 \text{ k-in/inch}$$

(per Case (1), P. 15)

Total plastic strain energy absorbed by the plastic deformation

$$W_1 = 2M_L L \theta$$

$$\text{Since K.E.} = W_1 = 2M_L L \theta$$

$$\theta = \frac{\text{K.E.}}{2 M_L L} = \frac{756.96}{2(0.878905)(120.0'')} = 3.588 \text{ (Rad)} = 205.6^\circ$$

Assuming impact contact length

The result of $\theta = 205.6^\circ$ indicates that the canister has suffered extremely severe damage such that even the strain energy generated from the complete smash of the canister cannot dissipate the input kinetic energy. The canister is expected to buckle during the parallel impact.

APPENDIX A-2. ALTERNATIVE ANALYSIS

(A-2-1) Normal Impact (Longitudinal Direction) (Case 1)

(A) Design Data

Young's Modulus: Stainless Steel = 25,700 Ksi (304L)

Unit Weight Stainless Steel = 500 lb/ft³ = 0.0002894 Kips/in³

$$\text{Glass} = 187 \text{ lb/ft}^3 = 0.0001824 \text{ Kips/in}^3$$

Unit Mass Stainless Steel = 0.000000734 K-sec²/inch

$$\begin{aligned} \text{Wave Velocity } C &= \sqrt{E/\rho} = \sqrt{25700 / 0.000000734} = 191,620 \text{ inch/sec} \\ &= 15,967 \text{ ft/sec} \end{aligned}$$
$$\text{Impact Duration } t_I = \frac{2L}{C} = \frac{2(126.38")}{191620} = 0.001319 \text{ sec}$$

Cross Sectional Area $A = \pi (R_o^2 - R_i^2) = \pi (6.375^2 - 6^2) = 14.759 \text{ in}^2$

(B) Determine the Impact Force

(1) Approach One Perfect-Plastic Impact

Total mass of the filled canister

$$M = 3.154 \text{ Kips} / 386 = 0.008171$$

Total change of linear momentum = 2 MV

$$2MV = 2 (0.008171)(430.7) = 7.0385$$

Impulse during the impact = $P_I' \Delta t$

$$\text{Hence, the impact force } p' = \frac{2MV}{\Delta t} = \frac{7.0385}{0.001319} = 5,336.2 \text{ Kips}$$

(2) Approach Two: by NDRC Empirical Formulas, pp. 193, Ref (12)

Contact pressure

$$P_I = \frac{263,820}{KN} \left(\frac{V}{12000 d} \right)^{0.2} g(x/a) \quad \text{psi}$$

Let $g(x/a) = 1.0$ for the extreme case, then

$$P_I = \frac{263,820}{KN} \left(\frac{V}{12,000 d} \right)^{0.2}$$

Where $V = 430.7$ inches/sec

$d = 12.375$ inches (mean)

$N = 0.72$ for flat skirt

$$K = 180 / \sqrt{f_c'} = \frac{180}{\sqrt{5,000}} = 2,546$$

for $f_c' = 5,000$ psi RC floor

$$P_I = \frac{26,3820}{(2,546)0.72} \left(\frac{430.7}{12,000 \times 12.375} \right)^{0.2}$$

$$= 44,730.6 \text{ psi} = 44.736 \text{ Ksi}$$

Total Impact Force

$$P_I = P_I * A_c = (44.736) (\pi) \left(\frac{12.375^2}{4} \right) = \underline{\underline{5,380.7 \text{ Kips}}}$$

Comparing the results obtained from the above two approaches,

it reveals that both approaches yield a very close estimate

of the impact force. Hence, it is reasonable to assume that

the maximum impact force is 5380 Kips for the case of deformable

canister, and 8256 Kips (approximately 150% that of the deformable

case, per Ref. (12)) for the non-deformable canister, or,

Non-deformable $P_I = 8,256 \text{ Kips}, p_I = 559.39 \text{ Ksi}$

Deformable $P_I = 5,380 \text{ Kips}, p_I = 364.52 \text{ Ksi}$

(C) Determine the buckling load of the designed canister

Assuming elastic buckling occurs when the impact pressure applies, the theoretical buckling pressure is given by (Ref (22))

$$P_{cr} = \frac{Et^2}{R \sqrt{3(1-\nu^2)}} = \frac{(25,700)(0.375)^2}{(6.188) \sqrt{3(1-0.3^2)}} = 353.93 \text{ Ksi}$$

or,

$$P_{cr} = P_{cr} * A_s = (353.48)(14.759) = 5217 \text{ Kips}$$

Note that the buckling load calculated by the formula is independent on the length of the cylinder.

Actual experimental results show that a buckling load P_{cr} of about 40% of the theoretical buckling load. (Ref. 22).

$$\text{Therefore, } p_{cr} = 353.48 \text{ Ksi} \times 0.40 = 141.39 \text{ Ksi}$$

prediction

$$P_{cr} = 5217 \text{ Kips} \times 0.40 = 2,086.8 \text{ Kips}$$

Comparing the predicted critical buckling axial force and the input impact force, it is obvious that the cylinder will undergo beyond the proportional limit of the material used. Hence, the inelastic buckling may result from the impact.

It should be noted that the above discussion is based on the static loading consideration. The impact effect will amplify the stress (force) level to the amount of static solutions time the dynamic load factor (DLF), see Appendix (A-4)

Inelastic Buckling Load

The inelastic buckling load is obtained from the formula given on Page 457, Ref (22).

$$P'_{cr} = \frac{t^2 \sqrt{E E_t}}{R \sqrt{3(1-\nu^2)}} \text{-----(a)}$$

Where E_t is the tangent modulus of the canister may be determined by trial and error from p_{cr} and the stress-strain relationship of the 304L stainless steel. A conservative estimate of E_t , without the rigorous calculation, can be taken as 50% of the Young's modulus E . Thus, $E_t = 12850$ Ksi

From equation (a) we obtain the inelastic buckling pressure

$$p'_{cr} = \frac{(0.375)^2 \sqrt{(25700)(12850)}}{(6.188 \sqrt{3(1-0.3^2)})} = \underline{\underline{78.89 \text{ Ksi}}}$$

and the corresponding buckling load is

$$P'_{cr} = p'_{cr} * A_s = (78.89)(14.759) = \underline{\underline{1164.3 \text{ Kips}}}$$

(D) Observations:

- (1) The canister will undergo plastic buckling upon the normal impacting. Since the elastic buckling pressure is greater than the yield strength $f_y = 25$ Ksi, the canister will yield before the elastic buckling occurs.
- (2) The assumption of E_t equal to 50% of E reaches a high inelastic buckling pressure. The inelastic buckling will take place at the stage that the inelastic buckling pressure is less than the yield strength. At that stage, the tangent modulus E_t is very small and the axial deformation is large.

(A-2-2) Rotational Impact and Striking Another Canister

(A) Determine the tip velocity at impact upon rotational dropping

Moment of inertia of the canister about the bottom rotating Point B.

$$I_B = \frac{W}{g} \left[\left(\frac{r_o^2}{4} + \frac{L^2}{12} \right) \right] - \frac{W}{g} \left(\frac{r_i}{r_o} \right)^2 \left[\left(\frac{r_i^2}{4} + \frac{L^2}{12} \right) \right]$$
$$= \frac{W}{g} \left[-\frac{1}{4} (r_o^2 - \frac{r_i^4}{r_o^2}) + \frac{L^2}{12} (1 - \frac{r_i^2}{r_o^2}) \right] \text{ ---- (a)}$$

Potential energy at normal position (to the target surface)

$$P.E. = W \frac{L}{2} \text{ ---- (b)}$$

Total (P.E.) transfers to the angular kinetic energy

Thus,

$$\frac{1}{2} I_B \dot{\omega}^2 = \frac{1}{2} WL \text{ ---- (c)}$$

Where $\dot{\omega}$ = angular velocity of the canister at final horizontal position

I_B = Moment of inertia about the rotating (Stationary) point.

Hence,

$$\frac{I_o}{2} \dot{\omega}^2 = \frac{1}{2} \frac{W}{g} \left[\frac{1}{4} (r_o^2 - \frac{r_i^4}{r_o^2}) + \frac{L^2}{12} (1 - \frac{r_i^2}{r_o^2}) \right] \dot{\omega}^2 = \frac{1}{2} WL$$

or, $\dot{\omega} = \sqrt{\frac{gL}{\frac{1}{4} (r_o^2 - \frac{r_i^4}{r_o^2}) + \frac{L^2}{12} (1 - \frac{r_i^2}{r_o^2})}} \text{ ---- (d)}$

Substituting the design data of the canister into eq. (d) we get

$$\dot{\omega} = 17.788254 \text{ rad/sec}$$

The resulting tip velocity (linear) at A is

$$\begin{aligned} \dot{U}_A &= \dot{\omega} L = (17.788254)(126.38") = 2248.08 \text{ in/sec} \\ &= 187.34 \text{ ft/sec} \end{aligned}$$

(B) Determine the free vibration properties of the canister

$$\text{1st Mode} \quad T_1 = 0.0134227 \text{ sec}$$

$$\omega_1 = 2 \pi f_1 = 468.1$$

$$\text{2nd Mode} \quad T_2 = 0.004873 \text{ sec}$$

$$\omega_2 = 2 \pi f_2 = 1289.4$$

$$\text{3rd Mode} \quad T_3 = 0.002485 \text{ sec}$$

(per calculation p. 30)

(C) Determine the impact equation of motion

The impact free vibration motion is determined by the solution (per

Clough/Penzian Ref. (23) P.331) as follows.

$$(X,t) = \frac{2\dot{U}_A}{\pi} \left[\frac{1}{\omega_1} \sin\left(\frac{\pi X}{L}\right) \sin(\omega_1 t) - \frac{1}{2\omega_2} \sin\left(\frac{2\pi X}{L}\right) \sin(\omega_2 t) + \dots \right] \quad \text{----- (e)}$$

The resulting force due to rotational dropping at any point X of the canister can be determined by the following relationship.

$$P(x,t) = EI \mathcal{U}'''(x,t) \quad \text{----- (f)}$$

Where

$$\mathcal{U}'''(x,t) = \frac{2\dot{v}_A}{\pi} \left(\frac{\pi}{L} \right)^3 \left[\frac{1}{\omega_1} \cos\left(\frac{\pi x}{L}\right) \sin(\omega_1 t) - \frac{1}{2\omega_2} \left(\frac{2}{1} \right)^3 \cos\left(\frac{2\pi x}{L}\right) \sin(\omega_2 t) + \text{-----} \right] \quad \text{----- (g)}$$

$$\begin{aligned} P(x,t) &= EI \mathcal{U}'''(x,t) \\ &= \frac{2\dot{v}_A}{\pi} \left(\frac{\pi}{L} \right)^3 EI \left[\frac{1}{\omega_1} \cos\left(\frac{\pi x}{L}\right) \sin(\omega_1 t) - \frac{4}{\omega_2} \cos\left(\frac{2\pi x}{L}\right) \sin(\omega_2 t) + \text{-----} \right] \quad \text{----- (h)} \end{aligned}$$

(D) Determine the impact force at tip point A

Eg (h) can be used to determine the maximum impact for at the tip point A by letting

$$x = L = 126.38'' ; \sin \omega_1 t, \sin \omega_2 t \text{ -----} = 1$$

$$E = 25,700 \text{ Ksi}$$

$$\begin{aligned} I &= \frac{\pi}{4} (R_o^4 - R_i^4) = \frac{\pi}{4} (6.375^4 - 6^4) = 279.37 \text{ inch}^4 \\ &= 6.188'' \text{ (mean)} \end{aligned}$$

Hence,

$$P(L,t) = \frac{2(2248.08)}{\pi} \left(\frac{\pi}{126.38} \right)^3 (25700)(279.37) \left[\frac{1}{468.1} - \frac{4}{1289.37} + \text{-----} \right]$$

$$P(L,t)_{\max} \doteq 329.55 \text{ Kips} = P_I$$

To calculate the stress induced by the impact force P_I , the formulas of elastic solution for circular ring are used (Ref. (1)).

$$X = 180^\circ \text{ (interested in the contact point B)}$$

$$P_I = 330 \text{ Kips}$$

$$W = \frac{P_I}{\pi d} = 8.49 \text{ Kips}$$

$$k_4 = 0.662$$

$$\begin{aligned} LT_m &= -WR^2(XZ + U-1) = -(8.49)(6.175)^2(\pi + 2) \\ &= 1670.1 \end{aligned}$$

$$\begin{aligned} LT_T &= -WRXZ \\ &= -(8.49)(6.175)(\pi)(0) = 0 \end{aligned}$$

$$\begin{aligned} LT_v &= -WRXU \\ &= -(8.49)(6.175)(-1) = -52.403 \end{aligned}$$

$$\text{At Point A, } M_A = \frac{WR^2}{2} k_4 = \frac{(8.49)(6.175)^2}{2} (0.662) = 107.11 \text{ k-in}$$

$$T_A = \frac{WR}{2} k_4 = \frac{(8.49)(6.175)^2}{2} (0.662) = 17.35 \text{ Kips}$$

$$V_A = 0$$

$$\begin{aligned} \text{At Point B, } M_B &= M_A - T_A R(1-u) + V_A RZ = LT_m \\ &= 107.11 - (17.35)(2) + 1670.1 = 1742.5 \text{ k-in} \end{aligned}$$

$$\begin{aligned} T_B &= T_A U + V_A Z + LT_T \\ &= (17.35)(-1) + (-52.4) = -69.75 \text{ Kips} \end{aligned}$$

$$V_A = -T_A Z + V_A U + L T v$$

$$= - (17.35)(1) + (52.40) = 52.40 \text{ Kips}$$

The resulting stresses are:

$$M_B = 1742.5 \text{ k-in}$$

$$\sigma_T = 69.75 \text{ k} / (1")(0.375") = 186.0 \text{ Ksi} > 25 \text{ Ksi}$$

$$\tau_s = 52.4 \text{ k} / (1")(0.375") = 139.73 \text{ Ksi} > 12.5 \text{ Ksi}$$

Results

$$\text{Max. principal stress} = 260.85 \text{ Ksi} > 25 \text{ Ksi}$$

$$\text{Min. principal stress} = -74.85 \text{ Ksi} > 25 \text{ Ksi}$$

$$\text{Max. shear stress} = 167.85 \text{ Ksi} > 12.5 \text{ Ksi}$$

The results indicate that the stresses caused by the rotational impact exceed the elastic limit. Hence, the linear elastic solutions are not valid in this case. The canister may expect to have inelastic deformation.

APPENDIX A-3

SUMMARY OF FORMULAS USED IN DANIEL DENG'S REPORT

1. Loss of Kinetic Energy (arbitrary impact angle)

$$\Delta (KE) = \frac{1}{2} \frac{W}{g} V_c^2 \left[\frac{1}{1+3\cos^2 \alpha} \right] \text{ ---- Eq. (4)}$$

2. Unstressed rear portion $\bar{X}_f$

$$\ln \left[\frac{\bar{X}_f}{L} \right]^2 = 1 - \frac{1}{1-\epsilon_1} + \ln (1-\epsilon_1) \text{ ----Eq. (16)}$$

3. Initial Condition ϵ_1

$$\frac{\epsilon_1^2}{1-\epsilon_1} = \frac{\rho V_c^2}{f_y} \text{ ----Eq. (7)}$$

4. Maximum allowable elongation of the cylinder

$$\Delta \ell = \pi (d_A - d) = 0.1 \pi d' \text{ ----Eq. (8)}$$

5. Strain Energy contributed by rotation of limit bending moment M_L

$$W_1 = 2M_L \theta \pi (d + d_A) \text{ ----Eq. (1)}$$

$$\text{Where } M_L = f_y \times \frac{t^2}{4}$$

6. Strain energy contributed by hoop tension of the cylinder

$$W_2 = f_y t \Delta \ell \times \left(\frac{hf}{2} \right) \text{ ----Eq. (10)}$$

7. Safety Criteria

$$(W_1 + W_2) \geq \text{Kinetic energy loss ----Eq. (11)}$$

APPENDIX A-4-1 Dynamic Load Factor (per Ref. 23)

$$f_n = \frac{K_n}{2\pi} \sqrt{\frac{EIg}{\omega L^4}}$$

Per Ref. 1, P.577

Uniform Bar

where $K_1 = 22.4$; $K_2 = 61.7$
 $K_3 = 121.0$; $K_4 = 200.0$

$E = 25,700.0$ ksi

$A = 14.579$ sq. in.

$I = (R_o^4 - R_i^4) = 279.335$ in⁴

$L = 126.38$ inches

$w = 0.02490$ kips/inch

1 st
Mode

Therefore,

$$f_1 = \frac{22.4}{2\pi} \sqrt{\frac{(25700)(279.335)(386.4)}{(0.02490)(126.38)^4}}$$

$$T_1 = \frac{1}{f_1} = 0.0134227 \text{ second}$$

$$f_2 = \frac{61.7}{22.4} (74.5) = 25.21 \text{ rad./second}$$

$$T_2 = \frac{1}{f_2} = 0.004873 \text{ second}$$

$$f_3 = 408.1 \text{ rad./second}$$

$$T_3 = 0.002485 \text{ second}$$

2 nd
Mode

3 rd
Mode

Free Vibration Mode Shapes

APPENDIX A-4-2 Natural Vibration Frequencies & Mode Shapes

APPENDIX A-5
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Figure A-1 Possible Drop Postures

(a)

(b)

Figure A-4 Probable Failure Mechanisms (Rotating Impact) (Case 2-B)

(a)

(b)

Case (4)

Case (3)

Figure A-5 Probable Failure Mechanisms (Parallel Impact)

Fig. A6 Impact pressure time history for 207 lb, 8 in., Schedule 40 pipe missile impacting 5000 lb/in.² concrete at 320 ft/sec.
(After Kennedy, Ref. (12))

Fig. A7 Comparison of numerical and analytical results in the case of impact of a homogeneous linear-elastic bar on a rigid wall: (---) = numerical results; (—) = analytical results.

(After Drittle, Ref. (10))

Figure A- 8 Typical Stress- Strain Diagram for Ductile Stainless Steel

Canister Circular Ring

Where

$$r_1 = r_0 (\sin(90^\circ - \theta))$$

$$U = r_0 - r_1$$

Figure A- 9 Postulated Deformed Circular Ring
(Parallel Impact)

Figure A- 10 Idealized Impact Forcing Function (Normal-Impact Case)

Let canister dimensions be

Length $L = 126.38''$

Outer diameter $d_o = 12.75''$

Inner diameter $d_i = 12.00''$

Total wt (filled) $W = 3.154$ Kips

Figure A-11 Rotating Impact on Another Canister

S. W. Chen
9/30/77
Job 77101

APPENDIX B

STUDY OF THE EFFECT OF AN EARTHQUAKE ON A FREELY SUSPENDED
CANISTER

	PAGE
(I) INTRODUCTION	1
(II) CASE STUDIES	3
(III) CONCLUSIONS	5
(IV) APPENDIX (CALCULATIONS)	7

Table of Figures

<u>Figure</u>		<u>Page</u>
B-1	Horizontal Design Response Spectra	14
B-2	Postulated Striking Position (Case 1)	15
B-3	Postulated Striking Force (Case 1)	16
B-4	Postulated Plastic Deformation of Canister (Case 1)	17
B-5	Idealized Stress-strain Relationship	18
B-6	Plastic Stress and Strain (Case 1)	18
B-7	Postulated Plastic Deformation of Canister (Case 2)	19
B-8	Postulated Striking Position and Force Equilibrium (Case 2)	19
B-9	Postulated Striking Position (Case 3)	20

(I) Introduction

The USNRC seismic and geologic siting criteria define two levels of earthquake ground motions for which a nuclear plant and its facilities must be designed: the Safe Shut-down Earthquake (SSE) and the Operating Basis Earthquake (OBE).

The SSE represents the maximum possible ground excitation at a site. For this earthquake, a facility should be designed so that all systems necessary to protect the health and safety of the public will remain functional both during and after the event. An upper limit to the peak horizontal ground acceleration for such an earthquake has been defined in this study as equal to 0.25g.

The Operating Basis Earthquake (OBE), on the other hand, represents the maximum excitation that can be expected to occur at a site during the operating life of a nuclear facility. For this earthquake, a facility should be designed so that those features of the plant necessary for continued operation without undue risk to the health and safety of the public will remain functional. For the OBE, the upper limit to peak horizontal acceleration has been defined as one-half that of the SSE or 0.125g.

Potential threats to the canister are posed by earthquakes principally during two stages in the canister life-cycle: (1) during transfers within the fuel reprocessing plant, and (2) between its reception at the terminal storage surface facility, and burial in the terminal storage mine.

During these stages the canister, suspended from an overhead crane, is lifted from and lowered into transfer carriages through floor ports. Earthquake-induced horizontal movement of the suspended canister during these lifting and lowering operations could cause the canister to strike the sides of the port and/or carriage. As the canister is lowered or raised through the floor ports, into and out of the transfer carriage, different relationships between the canister and its potential striking surface come into existence. This relative change of position must be taken into account when evaluating the performance of the canister in horizontal impacts.

Three possible hanging positions have been considered in this study:

- (1) The case when the canister bottom has left the transfer carriage and is suspended midway through the floor port, (2) the case when only the canister bottom is in the floor port, and
- (3) the case when the canister top is within the floor port and the canister bottom is in the transfer carriage.

Because the vibratory response of the canister to the earthquake excitation depends highly upon the characteristics of the particular earthquake, the analysis based on the design response spectrum suggested by the Regulatory Guide 1.60 for Seismic Design of the Nuclear Power Plants (Figure B-1) has been considered to be appropriate for the purposes of this study. The resultant striking force obtained from this design response spectrum is then used in the stress analysis of the canister.

A limit analysis approach is employed in this study. Any mitigating influence of the internal waste material on the analysis has been ignored. The final results have been assessed by comparison of the resulting stresses and deformations with the AISC code provisions.

(II) Case Studies

Three possible canister situations, each with its associated horizontal impact possibility have been considered in this study. For each case the striking force has been estimated by the upper bound spectral values from the Design Response Spectrum, Regulatory Guide 1.60 for the SSE. Limit analysis has been used to estimate and evaluate deformations which resulted from the postulated impacts. The following results have been abstracted from the detail calculations in Appendix (B-1).

SSE Criteria

- (a) Case (1): When the canister bottom has left the transfer carriage and is suspended midway through the floor port.

Max. spectral velocity = 41.26 in/sec

Max. spectral displacement = 30.0 inches

Yield deformation $\theta_y = 0.00519$ radian

Plastic deformation $e_p = 0.082$ radian

Plastic deformation (change in diameter)

$U = 0.021$ inch

Case (2): When the only canister bottom is in the floor port.

Max. yield rotation of the postulated cantilever type

failure mechanism $\theta_y = 0.000153$ radian

Plastic rotation $\theta_p = 0.004873$ radian

Ductility ratio $(\frac{\theta_y}{\theta_p}) = 31.9$

Max. tip end deformation $d = 0.4093$ inch

Max. shear stress $\tau_{max} = 8.16$ Ksi

Allowable shear stress $F_v = 10$ Ksi

Case (3): When the canister top is within the floor port and the
canister bottom is in the transfer carriage

Max. shear stress $= 0.4007$ Ksi

OBE Criterion

Case (1)

Plastic deformation $\theta_p = 0.041$ radian

Yield deformation $\theta_y = 0.00519$ radian

Permanent deformation (change in diameter)

$U = 0.010$ inch

Case (2)

$\theta = 0.002437$ radian $\theta_y = 0.000153$ radian

Tip deformation $d = 0.2047$ inch

Max. shear stress $\tau_{max} = 4.08$ Ksi < 10 Ksi

Case (3)

Max. shear stress $\tau_{max} = 0.2004$ Ksi < 10 Ksi

Observation

In cases (1) and (2), plastic deformation of the canister due to an earthquake exceeds the yield deformation for both the OBE and the SSE. Though permanent plastic deformation (denting) can be expected, the deformation is not expected to be large enough to cause serious structure damage. Because in all cases, shear stress is below AISC specification, no canister indentation will result from the shear force.

(III) Conclusion

Observations drawn from the calculations for the Safe Shut-down Earthquake (SSE) support the conclusion that the canister will safely respond to such earthquake excitation. These observations include: (1) the plastic deformation exceeds the elastic limit of deformation therefore, a permanent deformation (denting) will result upon the occurrence of a SSE. However, the canister will not suffer severe damage. (2) The shear stresses in cases (2) and (3) are far below the maximum allowable shear stress, hence, the canister will not be damaged by shear force. (3) The calculations indicate that canister integrity will not be violated by a Safe Shut-down Earthquake.

The Operating Basis Earthquake (OBE) exposes the canister to forces equal to one-half those presented by the Safe Shut-down Earthquake. Accordingly, although some denting will occur, damage to the canister during an OBE will be less than that occurs during a SSE. Therefore, it can be concluded that the canister's structural integrity will not be violated during a SSE.

Although all cases of the seismic study show that the canister will not rupture due to the OBE and the SSE, a nonlinear time history finite element analysis of the impact problem is recommended.

IV. Appendix

(A) Design Data

Canister

Total weight (filled with glass)	W = 3.154 Kips
Young's modulus (304L Stainless steel)	E = 25,700 Ksi
Steel Yield Strength	Fy = 25.0 Ksi (per ASME Section V, 1971)
Steel Tensile Strength	Ft = 70.0 Ksi (per ASME Section V, 1971)

Canister Dimensions

Outer Diameter	do = 12.750"
Inner Diameter	di = 12.000"
Shell Thickness	t = 3/8"
Total length (including skirt & head)	L = 132"
Cross sectional area	A = 14.759 in ²
Moment of inertia	I = 279.37 in ⁴

- (B) Case 1. When the canister bottom has left the assembly carriage and is suspended midway through the floor port.

- (a) Natural vibration frequency of the cable-canister system

$$f_n = \frac{1}{2\pi} \sqrt{\frac{g}{L}} = \frac{1}{2\pi} \sqrt{\frac{386}{234}} = 0.2044 \text{ CPS}$$

Based on the pre-defined design criteria, the maximum peak accelerations (horizontal) are:

Operating Basis Earthquake (OBE) 0.125g

Safe Shutdown Earthquake (SSE) 0.250g

(per Minutes of 7/8/77 Meeting)

The minimum damping ratio γ min. of 0.5% has been assumed for the system studied. The design response spectra in Regulatory Guide 1.60 gives the following:

OBE = Max. spectral velocity = 20.63 in/sec
 Max. spectral displacement = 15 inches

SSE = Max. spectral velocity = 41.26 in/sec
 Max. spectral displacement = 30 inches

Because the intensity of the SSE is two times that of the OBE, and the calculations will always remain proportional to each other, the following study based on the SSE will easily yield the corresponding results for that of OBE.

It is reasonable and conservative to assume that the surrounding cell structure moves with the maximum velocity of 41.26 in/sec when the earthquake starts exciting. The time required for canister to hit the port opening is approximately equal to 0.235 sec (9.68"/41.26 in/sec) which is much less than the fundamental period of the suspended canister. Hence, the striking velocity can be assumed to be equal to the max. velocity of 41.26 in/sec due to the fact that the canister is slow in responding to the excitation.

The relative kinetic energy (K.E.) of the canister at the striking is obtained as

$$K.E. = \frac{1}{2} MV^2 = \frac{1}{2} \left(\frac{3.154^k}{386} \right) (41.26^2) = 6.96 \text{ kips-inches}$$

Limit Analysis

The plastic bending moment along the postulated yield line is

$$M = f_y \times \frac{t^2}{4} = (25^{ksi}) \left(\frac{0.375^2}{4} \right) = 0.879 \text{ Kip-inch/inch}$$

The plastic strain energy contributed by this plastic bending moment is equal to the moment multiplied by the plastic angle change along the yield line. Assuming a 4' striking contact length, we have

$$S.E. = 2 \times \left(f_y \times \frac{t^2}{4} \right) \times L\theta = 2 \times (0.879) \times 48\theta = 84.384\theta$$

The angle change θ can be obtained by equating the impacting input kinetic energy to the strain energy (S.E.). Thus,

$$\frac{1}{2} MV^2 = S.E.$$

$$\theta = \frac{\frac{1}{2} MV^2}{S.E.} = \frac{6.96}{84.384} = 0.082 \text{ radian} = 4.726^\circ$$

Change in diameter (plastic deformation) is $U = (1 - \cos \theta) R_m$

or $U = (1 - \cos (4.726^\circ)) R_m = 0.021 \text{ inch}$ (See Figure B-4)

Discussion

The small deformation of 0.021 inch in the canister which, from its earthquake generated collision, should be discussed in conjunction with the admissible yield deformation θ_y : (where θ_y is calculated for the section before yield).

By a simple theory of the mechanics of solids, we have

$$\frac{1}{\rho} = \text{Curvature} = \frac{M}{EI} = \theta$$

The yield deformation θ_y can be obtained by the yield moment M_y .

$$M_y = \frac{bd^2}{6} (\sigma_y) = \frac{1(\frac{3}{8})^2}{6} (25) = 0.586 \text{ K-in}$$

$$\text{Hence, } \theta_y = \frac{M_y}{EI} = \frac{0.586}{(25700) \frac{(1)(3/8)^3}{12}} = 0.00519 \text{ radian}$$

Comparing θ_y with θ obtained from the impacting force, it is obvious that the deformation due to impact is beyond the elastic limit (See Fig. B-5).

Hence, a permanent deformation in the amount of 0.021 inch will result from the impact force used.

The plasticized shell section is shown in Figure B-6. Note that the shell section is not fully plasticized due to the impact force, therefore, residual stress will result upon the release of the impact force.

Case (2). When only the canister bottom is in the floor port.

When the canister strikes the concrete port, the canister behaves as a cantilever beam as shown in Figure B-7. Assuming that the plastic hinge takes place at the fixed end point where the maximum bending moment exists, the plastic bending moment of the cylinder section is therefore given by

$$M_p = 1.3 M_y \text{ (per AISI plastic design manual)}$$

Where M_y is the yield bending moment of the cylindrical section and 1.3 is the plastic shape factor for cylinder.

$$\text{Thus, } M_y = f_y \times S_s \times 1.3$$

$$= (\text{yield strength of the canister}) * (\text{Section Modules}) * 1.3$$

$$= 25^{\text{Ksi}} \times I/C \times 1.3$$

$$= 25 \times 279.37/6.375 \times 1.3$$

$$= 1,428.28 \text{ Kips-inch}$$

By equilibrium,

$$M = 24'' \times R_p$$

$$R_p = 24''$$

$$= 1428.3/24'' = 59.50 \text{ Kips}$$

$$= \text{Striking force}$$

By equating the work done by the plastic moment M to the input kinetic energy, we have

$$M \theta = \text{K.E.}$$

$$\theta = 6.96/1428.28 = \underline{\underline{0.004873}} \text{ radian}$$

$$\text{and } \theta = \frac{d}{L'}$$

$$d = \theta L' = (0.004873) (84'') = \underline{\underline{0.409332''}}$$

$$= \text{Tip deformation}$$

Discussion

- (1) The yield deformation at the assumed location of the plastic hinge can be obtained as follows.

$$\frac{M_y}{EI} = \frac{1}{S} = \theta_y, \text{ where } M_y \text{ is the yield moment capacity of the section,}$$

$$\text{thus, } M_y = f_y \cdot S = 25 \text{ Ksi} \times \frac{279.37}{6375} = 1095.6 \text{ K-inch}$$

$$\theta_y = \frac{M_y}{EI} = \frac{1095.6}{(25,700)(27937)} = \underline{\underline{0.000153}} \text{ radian}$$

$$\text{hence, the ductility ratio} = \frac{\theta}{\theta_y} = \frac{0.004873}{0.000153} = 31.9$$

The result indicates that the canister is within the plastic. Note that the deformation θ of the postulated failure was calculated based on the assumption that the input strain energy was completely dissipated through the cantilever type θ . The energy adsorbed by other sources of deformation (for instance, the deformation of the contact area BC) has been ignored. Therefore, the actual amount of deformation θ should be less than that calculated.

- (2) The maximum shear stress $\tau_{\max}$ that the canister experiences during the second case earthquake can be expressed as

$$\tau_{\max} = \frac{R_p}{\pi d t} = \frac{2 \times 59151}{\pi (12.375)(3/8)} = 8.16 \text{ Ksi}$$

The allowable shear stress is, per AISC specification

$$F_v = 0.4 f_y = (0.4)(25) = 10 \text{ Ksi} > 8.16 \text{ Ksi}$$

Hence, the canister can sustain the shear force without failure.

Case 3. The third case of interest is the investigation of shear stress when the top of the canister is within the floor port and the canister bottom is in the transfer carriage at the time of earthquake occurrence, (shown in Fig. B-9.)

The carriage is on wheels and is assumed free to move along the rails. It has also been assumed that the surrounding structure (cell) moves relatively fast and that the canister and carriage do not have time to respond. Therefore, a relative acceleration is developed between the surrounding structure and the canister and carriage. For the case of SSE ($\ddot{v}_g = 0.25g$) and assuming the minimum ^pdamping factor of 0.5%, the upper bound acceleration that can be developed by the surrounding structure is 1.25g. (Design Spectrum of Regulatory Guide 1.60).

Therefore, the maximum possible shear force that can be developed between the canister and the surrounding structure is

$$V_{\max} = M \ddot{v}_g = (1.25)(3.154) \text{ Kips} = 3.943 \text{ Kips} < 59.5 \text{ Kips}$$

$$\tau_{\max} = 0.4007 \text{ ksi}$$

The maximum shear force developed in this case is less than that already determined to be safe in the second case. Therefore, the canister can be considered safe in this case also.

(C) OBE CASES

Case (1).

Since the acceleration of the OBE (0.125g) is one-half that of the SSE (0.25g), the maximum pseudo velocity and the kinetic energy are proportionally one-half that of the SSE cases. Thus -

$$K.E. = \frac{1}{2} MV^2 = 3.48 \text{ Kip-in.}$$

The plastic deformation due to an earthquake is

$$\theta = 4.726^\circ/2 = 2.363^\circ = 0.041 \text{ radians} > \theta_y = 0.00519 \text{ radians}$$

Again, the deformation exceeds the yield limit and permanent post earthquake deformation can be expected.

Case (2)

$$\theta = 0.004873/2 = 0.002437 \text{ radian} > \theta_y = 0.000153 \text{ radian}$$

$$d = 0.40933/2 = 0.2047 \text{ inch}$$

$$\tau_{\max} = 8.16/2 = 4.08 \text{ Ksi} < 10 \text{ Ksi, AISC specification}$$

$< 12.5 \text{ Ksi, Maximum shearing stress}$
 in elastic range

Case (3)

$$\tau_{\max} = 0.4007/2 = 0.2004 \text{ Ksi} < 10 \text{ Ksi}$$

$< 12.5 \text{ Ksi}$

FIGURE B1 HORIZONTAL DESIGN RESPONSE SPECTRA – SCALED TO 1g HORIZONTAL
GROUND ACCELERATION

Figure B- 3 Postulated Earthquake Striking Force (Case 1)

Figure B-4 Postulated Plastic Deformation of Canister
(1st Case)

Figure B- 6 Plastic Stress and Plastic Strain, Case 1

Figure B- 5 Idealized Stress- strain Relationship

(b) Force Equilibrium

(a) Postulated Striking Position

Figure B-8 Postulated Striking Position and Force Equilibrium, 2nd Case

Figure B-7

Postulated Plastic Deformation of Canister (2nd Case)

Figure B- 9

Postulated Striking Position
(3rd Case)

APPENDIX C
DISPOSAL HOLE LINERS
(1972)

1. Review of liner materials by ORNL

Corrosion studies conducted during project Salt Vault are documented in CF-70-7-42 (draft of ORNL-4555). These studies were initiated following thermocouple failures during high-temperature heat experiments as a result of scaling and corrosion of the type 304 stainless steel sheath. The object of the studies was to determine the most economical material for all critical components in project Salt Vault where contact with salt could cause problems. Candidate materials investigated included stainless steel (types 304L, 310 and 347), Hastelloy C, Haynes Alloy No. 25, Titanium, Carbon Steel, Aluminum, Nickel, Monel, Inconel, Ni-o-nel, and protective coatings designed to allow use of a cheaper material, such as carbon steel, as in the base metal. A portion of the laboratory tests were conducted at 300°C, the "design" temperature for the disposal hole liners, with the various materials in contact with natural salt representative of the deposit underlying Lyons, Kansas. The laboratory tests indicated that type 304L stainless steel appeared to be the most economical and satisfactory material for use in natural salt at 300°C. Hastelloy, C, Hayes Alloy No. 25, and titanium would be satisfactory at 300°C but were more expensive and difficult to fabricate. Type 310 stainless steel and carbon steel were considered unsuitable due to pitting corrosion.

As the project Salt Vault demonstration progressed, a "modified pillar heater" experiment was included to further evaluate the qualities of stainless steel (type 304L) and carbon steel as waste container materials when in contact with salt at elevated temperature. After several months of operation, the experiment was expanded by the addition of the simulated waste container test (SWCT). The objective of the SWCT was to secure additional data on the effect of moisture on stainless steel and carbon steel containers. In both the "modified pillar heater" and the SWCT, the containers, fitted with electric heaters, were placed in holes in the salt and the holes backfilled with crushed salt as in an actual disposal operation. The tests, and results, are discussed in CF-70-7-42 and the results, briefly, are as follows:

- a. Examination of the 304L stainless steel containers revealed gross cracking due to chloride stress-corrosion.
- b. Examination of the carbon steel containers revealed a corroded outer surface (generalized rusting) but no evidence of penetration. There was apparently little, if any, loss of wall thickness to corrosion and container integrity should be maintained for an indefinite period of years.

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