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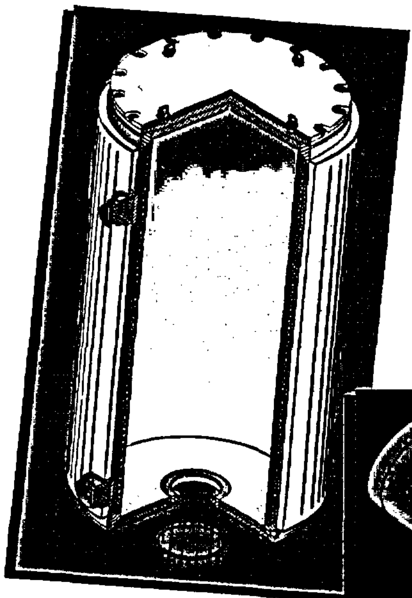
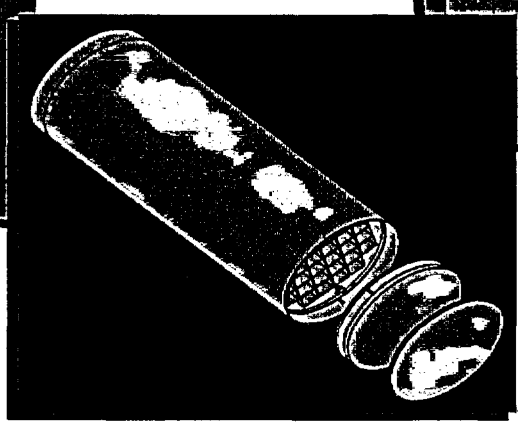
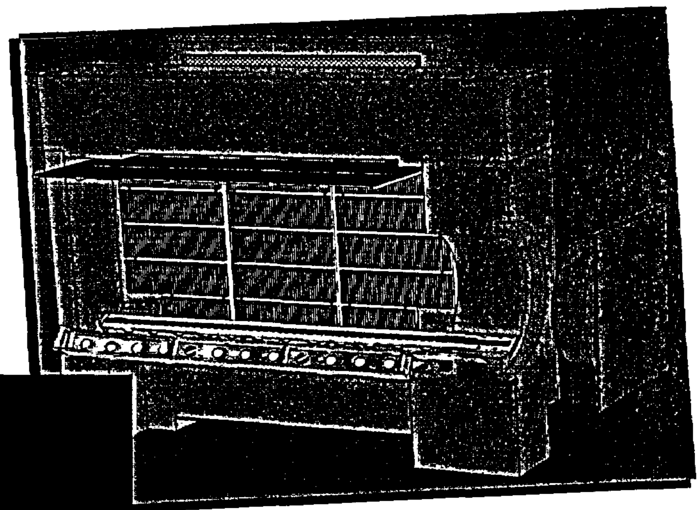
NUHOMS® HD
Horizontal Modular Storage System For Irradiated Nuclear Fuel



OS 187H TC



32PTH DSC



HSM-H

SAFETY ANALYSIS REPORT

Non-Proprietary

TRANSNUCLEAR

NUHOMS® HD
Horizontal Modular Storage System
For Irradiated Nuclear Fuel

SAFETY ANALYSIS REPORT

FOUR SKYLINE DRIVE, HAWTHORNE, NEW YORK 10532

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CHAPTER 1
GENERAL INFORMATION

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1. GENERAL INFORMATION

This Safety Analysis Report (SAR) describes the design and forms the licensing basis for 10CFR 72[1], Subpart L certification of the NUHOMS® HD dry spent fuel storage system. The NUHOMS® HD System provides for the horizontal storage of high burnup spent Pressurized Water Reactor (PWR) fuel assemblies in a dry shielded canister (DSC) that is placed in a Horizontal Storage Module (HSM-H) utilizing a OS187H transfer cask. The NUHOMS® HD System is designed to be installed in an Independent Spent Fuel Storage Installation (ISFSI) at power reactor sites under the provision of a general license in accordance with 10CFR 72, Subpart K. This system has been specifically optimized for high thermal loads, limited space, and needs for superior radiation shielding performance.

The QA program applicable to this design satisfies the requirements of 10CFR 72, Subpart G and is described in Chapter 13. The format of this SAR follows the guidance of NRC Regulatory Guide 3.61[2]. To facilitate NRC review of this application, this SAR has been prepared in compliance with the information and methods defined in NUREG-1536 [3], "Standard Review Plan for Dry Cask Storage Systems" and the associated Interim Staff Guidance (ISGs).

The NUHOMS® HD System is an improved version of the Standardized NUHOMS® System described in Certificate of Compliance (C of C) 72-1004 [4]. The 32PTH DSC included in this application is similar to the 24PTH DSC previously submitted for licensing as amendment No. 8 to the Standard NUHOMS® System [7]. The HSM-H is virtually identical to the HSM-H in the 24PTH amendment. The OS187H transfer cask (TC) is very similar to the previously licensed OS197 transfer cask but with a slightly larger diameter and closures containing seals.

The NUHOMS® HD System has been designed for enhanced heat rejection capabilities, and to permit storage of Non Fuel Assembly Hardware (NFAH) with the fuel and/or damaged spent fuel assemblies. Protection afforded to the public has been increased relative to the earlier HSM designs [5] by substantially reducing radiation dose rates. Details of the system design, analyses, operation, and margins are provided in the remainder of this SAR.

1.1 Introduction

The type of fuel to be stored in the NUHOMS® HD System is Light Water Reactor (LWR) fuel of the PWR type. The NUHOMS® HD System accommodates up to 32 PWR fuel assemblies with zircaloy, (zirlo, M5) cladding, uranium dioxide (UO_2), and Non-Fuel Assembly Hardware (NFAH). Provisions have been made, as discussed in Chapter 2, for storage of up to sixteen damaged fuel assemblies in the 32PTH DSC. The physical and radiological characteristics of these payloads are provided in Chapter 2.

The NUHOMS® HD System consists of the following components as shown in Figure 1-1, Figure 1-2, and Figure 1-6:

- A Horizontal Storage Module (HSM-H) that provides spent fuel decay heat removal, physical and radiological protection for the 32PTH DSC. The HSM-H consists primarily of thick concrete walls, a steel support structure for the 32PTH DSC, and a thick concrete door. Each HSM-H includes provisions for thermal monitoring instrumentation. The HSM-H is virtually identical to the HSM-H for the NUHOMS® 24PTH DSC included in amendment No. 8 to CoC 1004. [7].

- A Dry Shielded Canister (32PTH DSC) that provides confinement, an inert environment, structural support, and criticality control for 32 PWR fuel assemblies. The 32PTH DSC shell is a welded stainless steel pressure vessel that includes thick shield plugs at either end to maintain occupational exposures ALARA. The 32PTH DSC basket consists of stainless steel square tubes and support strips for structural support, and geometry control; and aluminum/borated aluminum for heat transfer and criticality control. The 32PTH DSC is very similar to the 24PTH DSC.

- The OS817H TC provides shielding and protection from potential hazards during the DSC closure operations and transfer to the HSM-H. It also provides a helium environment around the DSC during transfer operations. It is very similar to the previously licensed OS197 transfer cask for the Standardized NUHOMS® System.

- HSM-Hs are arranged in arrays to minimize space and maximize self-shielding. The 32PTH DSC is longitudinally restrained to prevent movement during seismic events. Arrays are fully expandable to permit modular expansion in support of operating power plants.

- The HSM-H provides the bulk of the radiation shielding for the 32PTH DSC. The HSM-Hs can be arranged in either a single-row or a back-to-back arrangement. Thick concrete supplemental shield walls are used at either end of an HSM-H array and along the back wall of single-row arrays to minimize radiation dose rates both onsite and offsite.

Approval of the NUHOMS® HD System components described above is sought under the provisions of 10CFR 72, Subpart L for use under the general license provisions of 10CFR 72, Subpart K. The components are intended for storage on a reinforced concrete pad at a nuclear power plant. In addition to these components, the system requires use of an onsite transfer cask,

transfer trailer, and other auxiliary equipment that is described in this SAR. Similar equipment was previously licensed under C of C 72-1004 [5]. Sufficient information for the transfer system and auxiliary equipment is included in this SAR to demonstrate that means for safe operation of the system are provided.

1.2 General Description of the NUHOMS® HD System

The NUHOMS® HD System provides for the horizontal, dry storage of canisterized Spent Fuel Assemblies (SFAs) in a concrete HSM-H. The storage system components consist of a reinforced concrete HSM-H and a stainless steel 32PTH DSC confinement vessel which holds the SFAs. The general arrangement of the NUHOMS® HD System components is shown in Figure 1-3 and Figure 1-4. The defined ASME Code Boundaries for the 32PTH DSC are shown in Figure 1-5. This SAR addresses the design and analysis of the storage system components, including the 32PTH DSC, the OS187H TC, and the HSM-H, which are important to safety in accordance with 10CFR 72.

In addition to these storage system components, the NUHOMS® HD System also utilizes transfer equipment to move the 32PTH DSCs from the plant's fuel/reactor building, where they are loaded with SFAs and readied for storage, to the HSM-Hs where they are stored. This transfer system consists of a transfer cask, a lifting yoke, a hydraulic ram system, a prime mover for towing, a transfer trailer, a cask support skid, and a skid positioning system. This transfer system interfaces with the existing plant fuel pool, the cask handling crane, the site infrastructure (i.e. roadways and topography) and other site specific conditions and procedural requirements. Auxiliary equipment such as a cask/canister annulus seal, a vacuum drying system and a welding system are also used to facilitate canister loading, draining, drying, inerting, and sealing operations. Similar transfer system and auxiliary equipment have been previously licensed under C of C 72-1004 [5].

During dry storage of the spent fuel, no active systems are required for the removal and dissipation of the decay heat from the fuel. The NUHOMS® HD System is designed to transfer the decay heat from the fuel to the canister and from the canister to the surrounding air by conduction, radiation and natural convection.

Each canister is identified by a Model Number, XXX-32PTH-YYY-Z, where XXX identifies the site for which the 32PTH DSC was fabricated, Z designates the basket type, and YYY is a sequential number corresponding to a specific canister. The basket types are described in SAR drawing no. 10494-72-10.

The NUHOMS® HD System components do not include receptacles, valves, sampling ports, impact limiters, protrusions, or pressure relief systems.

1.2.1 NUHOMS® HD System Characteristics

1.2.1.1 Dry Shielded Canister (32PTH DSC)

The key design parameters of the 32PTH DSC are listed in Table 1-1. The cylindrical shell, the inner top cover/shield plug, and shell bottom form the pressure retaining confinement boundary for the spent fuel. The inner top cover/shield plug and shell bottom provide shielding for the 32PTH DSC so that occupational doses at the ends are minimized during drying, sealing, handling, and transfer operations.

The 32PTH DSC has redundant welds which join the shell and the top cover plate and shield plug to form the confinement boundary. The cylindrical shell and inner bottom cover plate confinement boundary welds are fully compliant to Subsection NB of the ASME Code and are made during fabrication. The top closure confinement welds are made after fuel loading.

Both inner top cover/shield plug penetrations (siphon and vent ports) are welded after drying operations are complete. There are no credible accidents which could breach the confinement boundary of the 32PTH DSC as documented in Chapters 3 and 11.

The 32PTH DSC is designed for a maximum heat load of 34.8 kW. The internal basket assembly contains a storage position for each fuel assembly. The criticality analysis credits the fixed borated neutron absorbing material placed between the fuel assemblies. The analysis takes credit for soluble boron during loading operations. Sub-criticality during wet loading, drying, sealing, transfer, and storage operations is maintained through the geometric separation of the fuel assemblies by the basket assembly, the boron loading of the pool water, and the neutron absorbing capability of the 32PTH DSC materials, as applicable. Based on poison material and boron loading, several basket types are provided, as shown on drawing 10494-70-11 and described in Chapter 6.

Structural support for the PWR fuel is provided by the basket fuel compartments and support strips. The support strips are located periodically over the full length of the basket with allowance provided for thermal growth. Stainless steel transition rails are provided at the basket periphery for support and heat transfer.

Dimensions of the 32PTH DSC components described in the text and provided in figures and tables of this SAR are nominal dimensions for general system description purposes. Actual design dimensions are contained in the drawings in Section 1.5.2 of this SAR. For a discussion of the contents authorized to be stored in this DSC, see Section 2.1.1 of this SAR.

1.2.1.2 Horizontal Storage Module (HSM-H)

Each HSM-H provides a self-contained modular structure for storage of spent fuel canisterized in a 32PTH DSC. The HSM-H is constructed from reinforced concrete and structural steel. The thick concrete roof and walls provide substantial neutron and gamma shielding. Contact doses for the HSM-H are designed to be ALARA. The key design parameters of the HSM-H are listed in Table 1-1.

The nominal thickness of the HSM-H roof is four feet for biological shielding. Separate shield walls at the end of a module row in conjunction with the module wall, provide a minimum thickness of four feet for shielding. Similarly, an additional shield wall is used at the rear of the module if the ISFSI is configured as single module arrays. Sufficient shielding is provided by thick concrete side walls between HSM-Hs in an array to minimize doses in adjacent HSMs during loading and retrieval operations.

The HSM-Hs provide an independent, passive system with substantial structural capacity to ensure the safe dry storage of SFAs. To this end, the HSM-Hs are designed to ensure that normal

transfer operations and postulated accidents or natural phenomena do not impair the 32PTH DSC or pose a hazard to the public or plant personnel.

The HSM-H provides a means of removing spent fuel decay heat by a combination of radiation, conduction and convection. Ambient air enters the HSM-H through ventilation inlet openings located on both sides of the lower front wall of the HSM-H and circulates around the 32PTH DSC and the heat shields. Air exits through air outlet openings located on each side of the top of the HSM-H. The HSM-H is designed to remove up to 34.8 kW of decay heat from the 32PTH DSC.

Decay heat is rejected from the 32PTH DSC to the HSM-H air space by convection and then removed from the HSM-H by natural circulation air flow. Heat is also radiated from the 32PTH DSC surface to the heat shields and HSM-H walls where the natural convection air flow and conduction through the walls aids in the removal of the decay heat. The passive cooling system for the HSM-H is designed to assure that SFA peak cladding temperatures during long term storage remain below acceptable limits to ensure fuel cladding integrity.

The HSM-Hs are installed on a load bearing foundation which consists of a reinforced concrete basemat on a subgrade suitable to support the loads. The HSM-Hs are not tied to the basemat.

Dimensions of the HSM-H components described in the text and provided in figures and tables of this SAR are nominal dimensions for general system description purposes. Actual design dimensions are contained in the drawings in Section 1.5.2 of this SAR.

1.2.1.3 Transfer Systems

1.2.1.3.1 OS187H On-Site Transfer Cask

The OS187H transfer cask (TC) used in the NUHOMS® HD System provides shielding and protection from potential hazards during 32PTH DSC loading and closure operations and transfer to the HSM-H. The key design parameters of the TC are listed in Table 1-1. The TC included in this SAR is the NUHOMS® cask which is limited to on-site use under 10CFR 72. The OS-187H transfer cask is very similar to the OS197 and OS 197H transfer casks described in the FSAR for the Standard NUHOMS® Storage System [5].

The OS-187H TC has a 186.6 inch cavity length, a 70.5 inch inside diameter and a payload capacity of 121,000 pounds (wet) and 109,000 pounds (dry). The TC is designed to meet the requirements of 10CFR72 for on-site transfer of the DSC from the plant's fuel pool to the HSM-H. As shown in Figure 1-6, the TC is constructed from two concentric stainless steel shells with a bolted and gasketed top cover plate and a welded bottom end assembly. The TC also includes an outer steel jacket which is filled with water to provide neutron shielding. The top and bottom end assemblies also incorporate a solid neutron shield material.

The TC is designed to provide sufficient shielding to ensure dose rates are ALARA. Two top lifting trunnions are provided for handling the TC using a lifting yoke and overhead crane. Lower trunnions are provided for rotating the cask from/to the vertical and horizontal positions on the support skid/transport trailer. A gasketed cover plate is provided to seal the bottom hydraulic ram

access penetration of the cask during loading. The TC lid is also provided with gaskets so that a helium environment can be maintained during DSC transfer operations.

1.2.1.3.2 Transfer Equipment

Transfer Trailer: The typical transfer trailer for the NUHOMS® HD System consists of a heavy industrial trailer used to transfer the empty cask, support skid and the loaded transfer cask between the plant's fuel/reactor building and the ISFSI. The trailer is designed to ride as low to the ground as possible to minimize the overall HSM-H height and the transfer cask height during 32PTH DSC transfer operations. The trailer is equipped with four hydraulic leveling jacks to provide vertical alignment of the cask with the HSM-H. The trailer is towed by a conventional heavy haul truck tractor or other suitable prime mover. Figure 1-7 shows the typical trailer.

Cask Support Skid: The cask support skid for the NUHOMS® HD System is shown in Figure 1-8 and is essentially the same as described in the FSAR [5] for the standard NUHOMS® System. Key design features include:

The skid is mounted on a surface with sliding support bearings and hydraulic positioners to provide alignment of the cask with the HSM-H. Brackets with locking bolts are provided to prevent movement during trailer towing.

The hydraulic ram may be mounted on the skid or, as an option, the ram can be set-up using a frame structure bolted to the cask bottom and a rear support tripod.

The cask support skid is mounted on a low profile heavy haul industrial trailer.

The plant's fuel/reactor building crane or other suitable lifting device is used to lower the cask onto the support skid which is secured to the transfer trailer. Specific details of this operation and the fuel/reactor building arrangement are covered by the provisions of the plant's 10CFR 50 operating license.

Hydraulic Ram: The hydraulic ram system consists of a hydraulic cylinder with a capacity and a reach sufficient for 32PTH DSC insertion into and retrieval from the HSM-H. The design of the ram support system provides a direct load path for the hydraulic ram reaction forces during 32PTH DSC insertion and retrieval. The system uses a rear ram support for alignment of the ram to the 32PTH DSC, and trunnions as the front support. The design provides positive alignment of the major components during 32PTH DSC insertion and retrieval.

1.2.2 Operational Features

This section provides a discussion of the sequence of operations involving the NUHOMS® HD System components.

1.2.2.1 Dry Run Operations

A dry run utilizing a 32PTH DSC loaded with mock-up fuel assemblies will be performed prior to loading the first canister by each licensee to demonstrate the adequacy of training, familiarity

of system components and operational procedures. Mock-up fuel assemblies shall provide a representation of the maximum fuel assembly cross sectional envelope and provide a reasonable approximation of fuel assembly length and weight. The licensee shall determine the quantity of mock-up fuel assemblies required for the dry run to demonstrate that the loading and unloading processes are sound and the operations personnel are adequately trained.

The loading and unloading operations which have an impact on safety will be verified and recorded according to the requirements detailed in Chapter 8. The operations include loading and identifying fuel assemblies, ensuring the fuel assemblies meet the fuel acceptance criteria, drying, backfilling and pressurizing the canister, gas sampling and transferring the loaded canister to the HSM-H. Additionally, the ability to weld the top cover plates and open a sealed canister shall be demonstrated.

1.2.2.2 SFA Loading Operations

The primary operations (in sequence of occurrence) for the NUHOMS® HD System are:

1. Transfer Cask Preparation
2. 32PTH DSC Preparation
3. Place 32PTH DSC in Transfer Cask
4. Fill Transfer Cask/32PTH DSC Annulus with Clean Water and Seal
5. Fill 32PTH DSC Cavity with Fuel Pool Water (may be accomplished in step 6)
6. Lift Transfer Cask and Place in Fuel Pool
7. Spent Fuel Loading
8. Top Shield Plug Placement
9. Lifting Transfer Cask from Pool (DSC water may be drained)
10. Top Shield Plug Sealing
11. Vacuum Drying and Backfilling
12. Outer Top Cover Plate Sealing
13. Transfer Cask/32PTH DSC Annulus Draining and Transfer Cask Top Cover Plate Placement
14. Backfill Transfer Cask Cavity with Helium
15. Place Loaded Transfer Cask on Transfer Skid/Trailer

16. Move Loaded Transfer Cask to HSM-H

17. Transfer Cask/HSM-H Preparation and Alignment

18. Insertion of 32PTH DSC into HSM-H

19. HSM-H Closure

These operations are described in the following paragraphs. The descriptions are intended to be generic and are described in greater detail in Chapter 8. Plant specific requirements may affect these operations and are to be addressed by the licensee.

Transfer Cask Preparation: Transfer cask preparation includes exterior washdown and interior decontamination. These operations are performed on the decontamination pad/pit outside the fuel pool area. The operations are similar to those for a shipping cask which are performed by plant personnel using existing procedures.

32PTH DSC Preparation: The internals and externals of the 32PTH DSC are inspected and cleaned if necessary. This ensures that the 32PTH DSC will meet plant cleanliness requirements for placement in the spent fuel pool.

Place 32PTH DSC in Transfer Cask: The empty 32PTH DSC is inserted into the transfer cask.

Fill Transfer Cask/32PTH DSC Annulus with Water and Seal: The transfer cask/32PTH DSC annulus is filled with uncontaminated water and is then sealed prior to placement in the pool. This prevents contamination of the 32PTH DSC outer surface and the transfer cask inner surface by the pool water.

Fill 32PTH DSC Cavity with Water: The 32PTH DSC cavity is filled with pool water to prevent an in-rush of water as the transfer cask is lowered into the pool.

Lift Transfer Cask and Place in Fuel Pool: The transfer cask, with the water-filled 32PTH DSC inside, is then lowered into the fuel pool. The transfer cask liquid neutron shield, if provided, may be left unfilled to meet hook weight limitations.

Spent Fuel Loading: Spent fuel assemblies are placed into the 32PTH DSC. This operation is identical to that presently used at plants for shipping cask loading.

Top Shield Plug Placement: This operation consists of placing the top shield plug into the 32PTH DSC using the plant's crane or other suitable lifting device.

Lifting Transfer Cask from Pool: The loaded transfer cask is lifted out of the pool and placed (in the vertical position) on the drying pad in the decon pit. This operation is similar to that used for shipping cask handling operations.

Top Shield Plug Sealing: The water contained in the space above the top shield plug is drained. The top shield plug is welded to the shell. This weld provides the inner (confinement) seal for the 32PTH DSC.

Vacuum Drying and Backfilling: The initial blowdown of the 32PTH DSC is accomplished by pressurizing the vent port with air or helium. The remaining liquid water in the cavity is forced out of the siphon tube and routed back to the fuel pool or to the plant's liquid radwaste processing system via appropriate size flexible hose or pipe, as appropriate. The cavity water may also be removed by pumping out the water using the siphon port/tube. The 32PTH DSC is then evacuated to remove the residual liquid water and water vapor in the cavity. When the system pressure has stabilized, the 32PTH DSC is backfilled with helium and re-evacuated. After the second evacuation, the 32PTH DSC is again backfilled with helium and slightly pressurized.

The helium lines removed, and the siphon and vent port penetrations are closed. The vent and siphon cover plates are installed and welded to the shield plug.

Top Cover Plate Sealing: After helium backfilling, the 32PTH DSC outer top cover plate is installed by using a partial penetration weld between the outer top cover plate and the shell.

The outer cover plate or shell weld and shield plug weld provide redundant seals at the upper end of the 32PTH DSC.

Transfer Cask/32PTH DSC Annulus Draining and Transfer Cask Top Cover Plate Placement: The transfer cask/32PTH DSC annulus is drained. A swipe is then taken over the 32PTH DSC exterior at the top cover plate and the upper portion of the shell. Demineralized water is flushed through the transfer cask/32PTH DSC annulus, as required, to remove any contamination left on the 32PTH DSC exterior. The transfer cask top cover plate is installed, using the plant's crane or other suitable lifting device, and bolted closed.

Backfill Transfer Cask Cavity with Helium: The TC cavity is evacuated and the cavity/annulus is backfilled to a positive pressure with helium.

Place Loaded Transfer Cask on Transfer Skid/Trailer: The transfer cask is lifted onto the transfer cask support skid and downended onto the transfer trailer from the vertical to horizontal position. The transfer cask is secured to the skid.

Move Loaded Transfer Cask to HSM: Once loaded and secured, the transfer trailer is towed to the ISFSI along a predetermined route on a prepared road surface. Upon entering the ISFSI the cask is positioned and aligned with the designated HSM-H into which the 32PTH DSC is to be transferred.

Transfer Cask/HSM Preparation and Alignment: At the ISFSI with the cask positioned in front of the HSM-H, the transfer cask top cover plate is removed. The HSM-H door is removed and the transfer trailer is then backed into close proximity with the HSM-H. The skid positioning system is then used for the final alignment and docking of the transfer cask with the HSM-H and the cask restraint installed.

Insertion of 32PTH DSC into HSM: After final alignment of the transfer cask, HSM-H, and hydraulic ram, the 32PTH DSC is pushed into the HSM-H by the hydraulic ram.

HSM Closure: Install 32PTH DSC axial retainer and install HSM-H door.

1.2.2.3 Identification of Subjects for Safety and Reliability Analysis

1.2.2.3.1 Criticality Prevention

Criticality is controlled by utilizing the fixed borated neutron absorbing material in the 32PTH DSC basket and the pool water boron loading. During storage, with the cavity dry and sealed from the environment, criticality control measures within the installation are not necessary because water cannot enter the canister during storage.

1.2.2.3.2 Chemical Safety

There are no chemical safety hazards associated with operations of the NUHOMS® HD System. The coating materials used in the design of the 32PTH DSC are chosen to minimize hydrogen generation. Hydrogen monitoring is required during sealing operations to ensure hydrogen concentration levels remain within acceptable limits.

1.2.2.3.3 Operation Shutdown Modes

The NUHOMS® HD System is a totally passive system so that consideration of operation shutdown modes is unnecessary.

1.2.2.3.4 Instrumentation

The NUHOMS® HD System is a totally passive system. No safety-related instrumentation is necessary. The maximum temperatures and pressures are conservatively bounded by analyses. Therefore, there is no need for monitoring the internal cavity of the 32PTH DSC for pressure or temperature during normal operations. The 32PTH DSC is conservatively designed to perform its confinement function during all worst case normal, off-normal, and accident conditions.

1.2.2.3.5 Maintenance and Surveillance

All maintenance and surveillance tasks are described in Chapter 9.

1.2.3 32PTH DSC Contents

The 32PTH DSC is designed to store up to 32 intact PWR Westinghouse 15x15 (WE 15x15), Westinghouse 17x17 (WE 17x17), and/or Framatome ANP Advanced MK BW 17x17 fuel assemblies (Fr 17x17) with or without NFHAs likeVibration Suppressor Inserts (VSI), Burnable Poison Rod Assemblies (BPRAs), or Thimble Plug Assemblies (TPAs). The 32PTH DSC is also designed for storage of up to 16 damaged fuel assemblies, and remaining intact assemblies, utilizing top and bottom end caps. A description of the fuel assemblies including the damaged fuel assemblies is provided in Chapter 2. The maximum allowable initial enrichment of the fuel

to be stored is 5.00 weight % U-235 and the maximum burnup is 60,000 MWd/MTU. The fuel must be cooled at least 5 years prior to storage.

The criticality control features of the NUHOMS® HD System are designed to maintain the neutron multiplication factor k-effective (including uncertainties and calculational bias) at less than 0.95 under normal, off-normal, and accident conditions.

The quantity and type of radionuclides in the SFAs are described and tabulated in Chapter 5. Chapter 6 covers the criticality safety of the NUHOMS® HD System and its parameters. These parameters include rod pitch, rod outside diameter, material densities, moderator ratios, and geometric configurations. The maximum pressure buildup in the 32PTH DSC cavity is addressed in Chapter 4.

1.3 Identification of Agents and Contractors

The prime contractor for design and procurement of the NUHOMS® HD System components is Transnuclear, Inc. (TN). TN will subcontract the fabrication, testing, on-site construction, and QA services as necessary to qualified firms on a project specific basis in accordance with TN QA program requirements.

The design activities for the NUHOMS® HD Safety Analysis Report were performed by TN and subcontractors in accordance with TN QA program requirements. TN is responsible for the design and analysis of the 32PTH DSC, the HSM-H, the on-site TC, and the associated transfer equipment.

Closure activities associated with welding the top cover plates on the 32PTH DSC following fuel loading are typically performed by the licensee under the licensee's NRC approved QA program.

1.4 Generic Cask Arrays

The 32PTH DSC containing the SFAs is transferred to, and stored in a HSM-H in the horizontal position. Multiple HSM-Hs are grouped together to form arrays whose size is determined to meet plant-specific needs. Arrays of HSM-Hs are arranged within the ISFSI site on a concrete basemat(s) with the entire area enclosed by a security fence. Individual HSM-Hs are arranged adjacent to each other. The decay heat for each HSM-H is primarily removed by internal natural circulation flow and conduction through the HSM-H walls. Figures 1-9, 1-10, and 1-11 show typical layouts for NUHOMS® 32PTH ISFSIs which are capable of modular expansion to any capacity. These are typical layouts only and do not represent limitations in number of modules, number of rows, and orientation of modules in rows. An empty module is required at the end of an array to allow for future expansion. Back to back module configurations require expansion in pairs. Expansion can be accomplished as necessary by the licensee provided the criteria of 10CFR 72.104, 10CFR 72.106 and Chapter 12 are met. The parameters of interest in planning the installation layout are the configuration of the HSM-H array and an area in front of each HSM-H to provide adequate space for backing and aligning the transfer trailer.

1.5 Supplemental Data

1.5.1 References

1. Title 10, Code of Federal Regulations, Part 72, "Licensing Requirements for the Storage of Spent Fuel in an Independent Spent Fuel Storage Installation".

2. U.S. Nuclear Regulatory Commission, Regulatory Guide 3.61, Standard Format and Content for a Topical Safety Analysis Report for a Spent Fuel Dry Storage Cask, February 1989.

3. U.S. Nuclear Regulatory Commission, "Standard Review Plan for Dry Cask Storage Systems," NUREG 1536, U.S. NRC, January 1997.

4. NRC Certificate of Compliance 72-1004, NUHOMS® General License Spent Fuel Storage System, Amendment No. 7, March, 2004.

5. TN West, Final Safety Analysis Report for the Standardized NUHOMS® Horizontal Modular Storage System for Irradiated Nuclear Fuel, Revision 7, November 2003, File NUH003.0103, USNRC Docket No. 72-1004.

6. Title 10, Code of Federal Regulations, Part 50, "Domestic Licensing of Production and Utilization Facilities".

7. Application for Amendment No. 8 of the NUHOMS® Certificate of Compliance 72-1004, Revision 0, September 2003.

1.5.2 Drawings

- 32PTH DSC: 10494-72-(1 to 12), Rev 0 (PROPRIETARY)
- OS187H: 10494-72-(15 to 21), Rev 0 (PROPRIETARY)
- HSM-H: 10494-72-(100 to 109) Rev. 0 (PROPRIETARY)
- Damaged Fuel End Caps: 10494-72-30, Rev 0 (PROPRIETARY)

Table 1-1
Key Design Parameters of the NUHOMS® HD System Components

Dry Shielded Canister (32PTH DSC)	
Overall Length (in)	185.75 (max)
Outside Diameter (in)	69.75
Cavity Length (in)	164.5 (min)
Shell Thickness (in)	0.5
Design Weight of Loaded 32PTH DSC (lbs.)	82,000
Materials of Construction	Stainless Steel Shell Assembly and Internals, Carbon Steel and/or Stainless Steel Shield Plugs, Aluminum
Neutron Absorbing Material	Boral™, borated aluminum, metal matrix composite (MMC)
Internal Atmosphere	Helium

Horizontal Storage Module (HSM-H):	
Overall length (without back shield wall)	20'-8"
Overall width (without end shield walls)	9'-8"
Overall height	18' 6"
Total Weight not including 32PTH DSC (lbs.)	306,000
Materials of Construction	Reinforced Concrete and Structural Steel
Heat Removal	Conduction, Convection, and Radiation

On-Site Transfer Cask (OS187H)	
Overall Length (in)	197.1
Outside Diameter (in)	92.2
Cavity Length (in)	186.6
Lead Thickness (in)	3.60 (nom)
Gross Weight (including 32PTH DSC) (tons)	114.5
Materials of Construction	Stainless Steel Shell Assemblies and closures with lead shielding
Internal Atmosphere	Helium

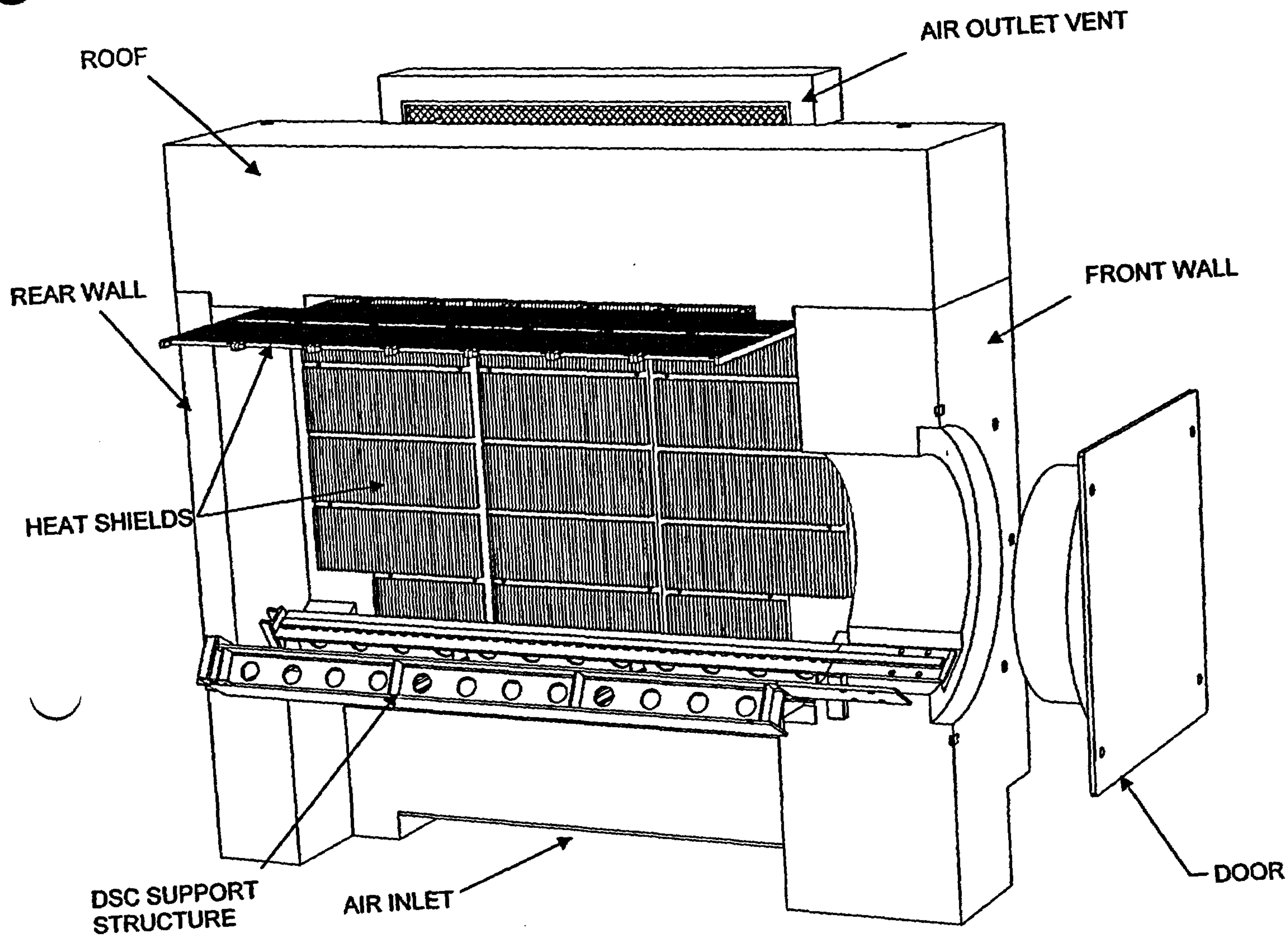
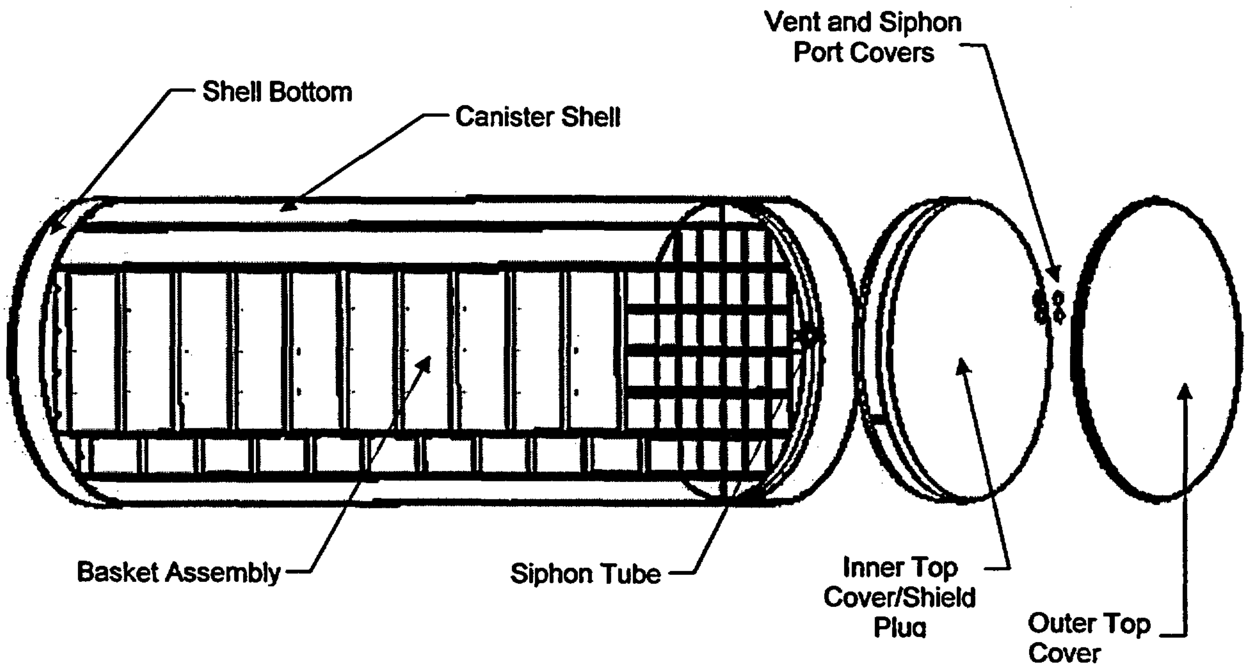


Figure 1-1
NUHOMS® HD System Horizontal Storage Module (HSM-H)



Note: Bottom end of 32PTH DSC not shown.

Figure 1-2
NUHOMS® HD 32PTH DSC

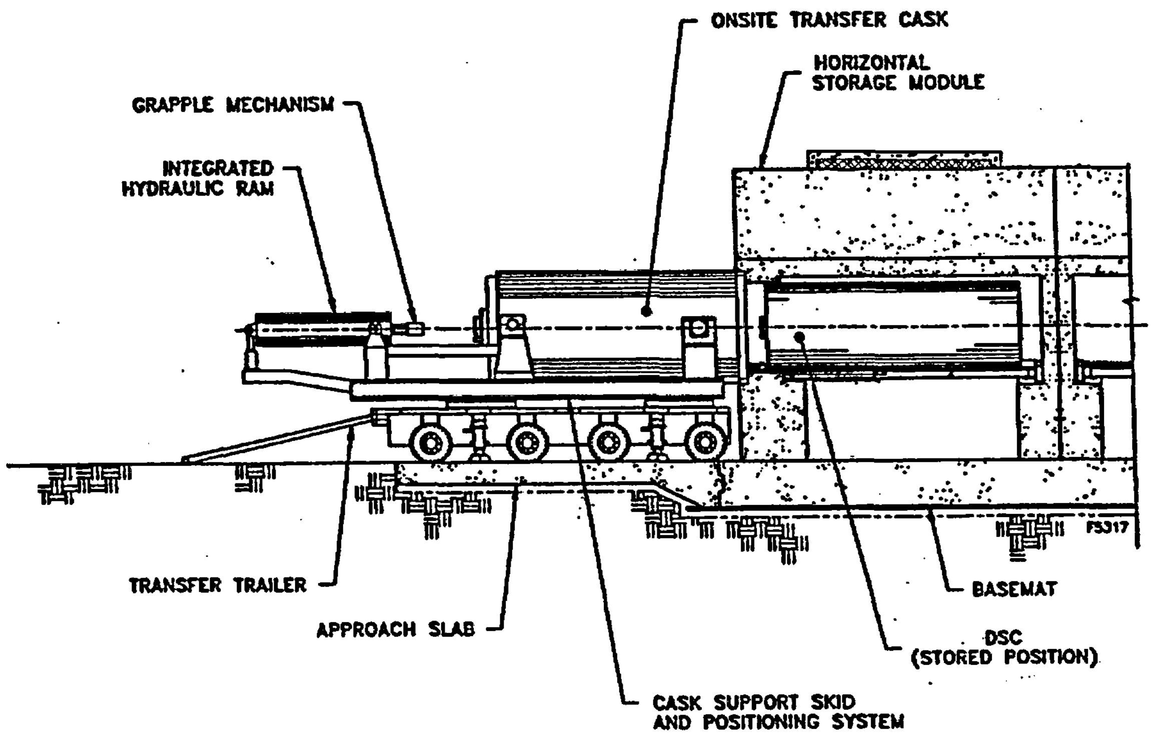


Figure 1-3
NUHOMS® HD System Components, Structures, and Transfer Equipment – Elevation View (Typical)

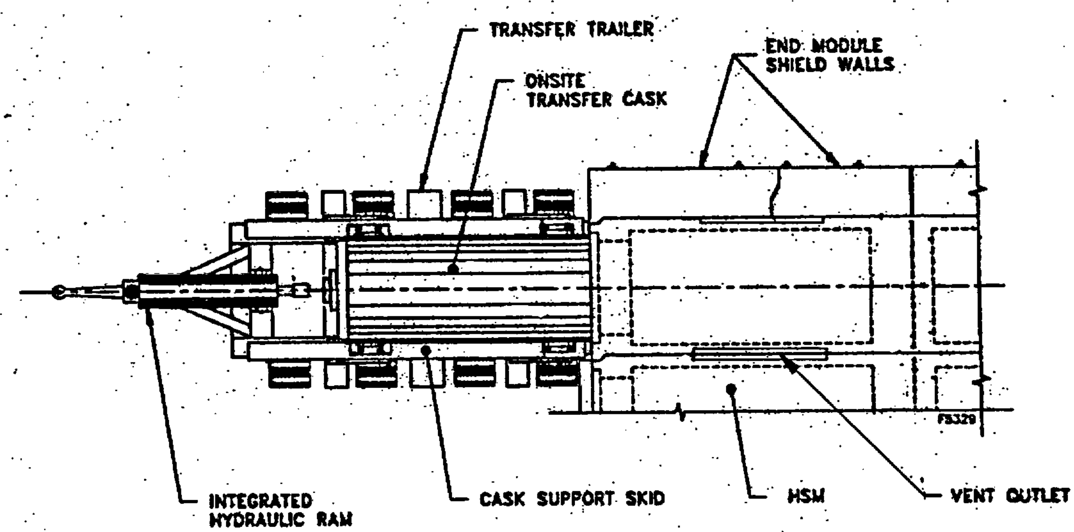


Figure 1-4
NUHOMS® HD System Components, Structures, and Transfer Equipment – Plan View (Typical)

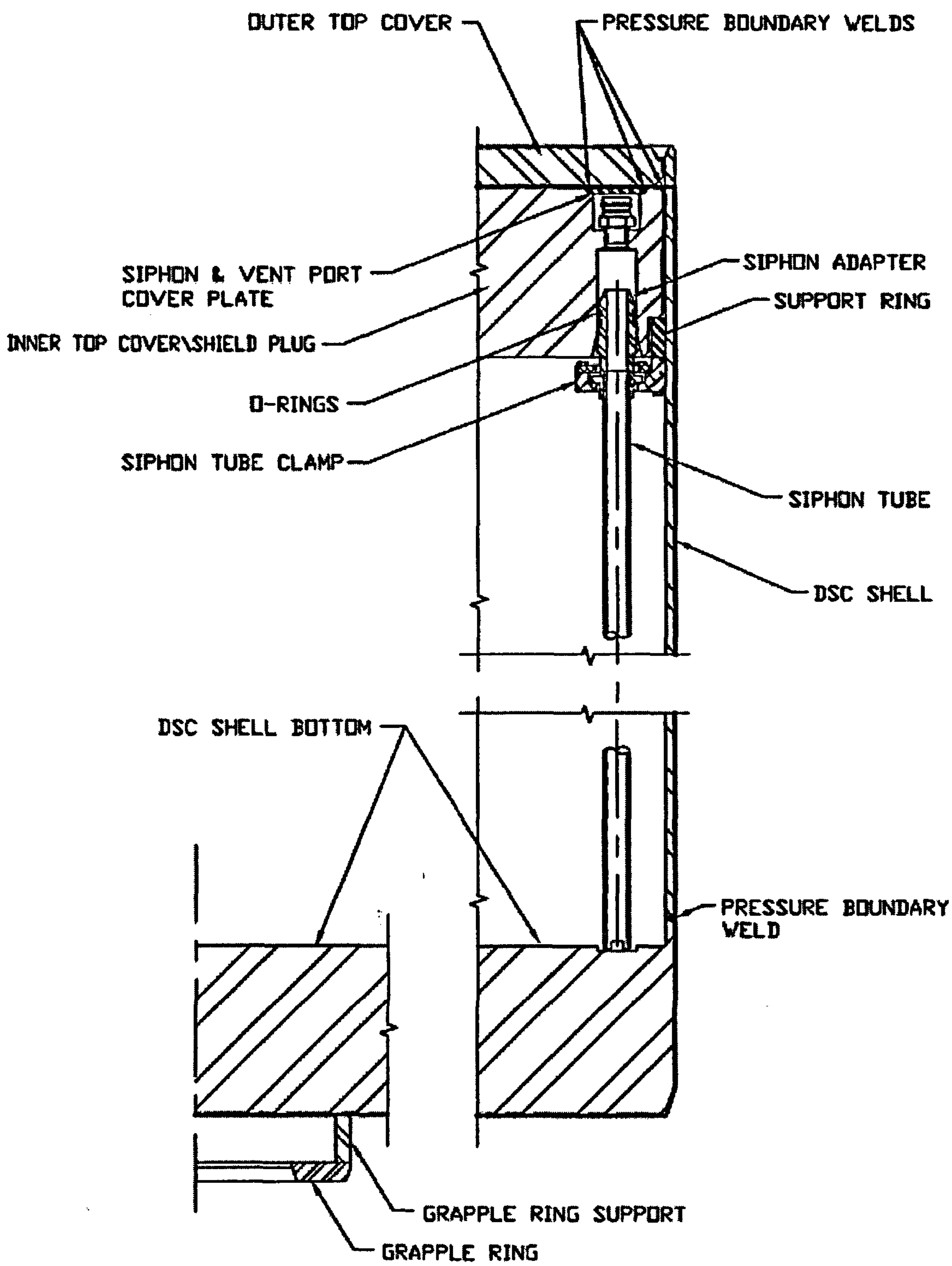


Figure 1-5

32PTH DSC ASME Code Boundaries

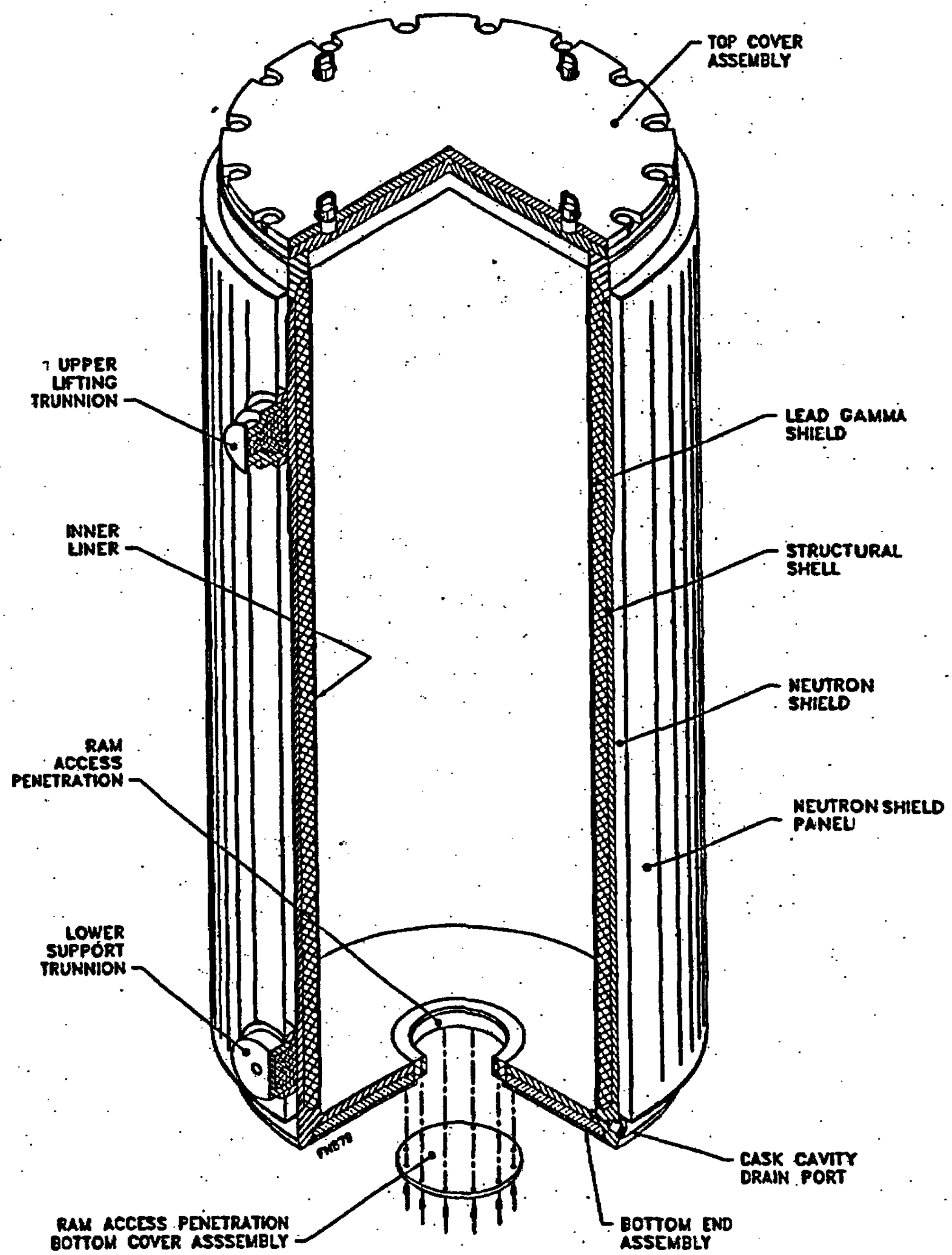


Figure 1-6

OS187H On-Site Transfer Cask

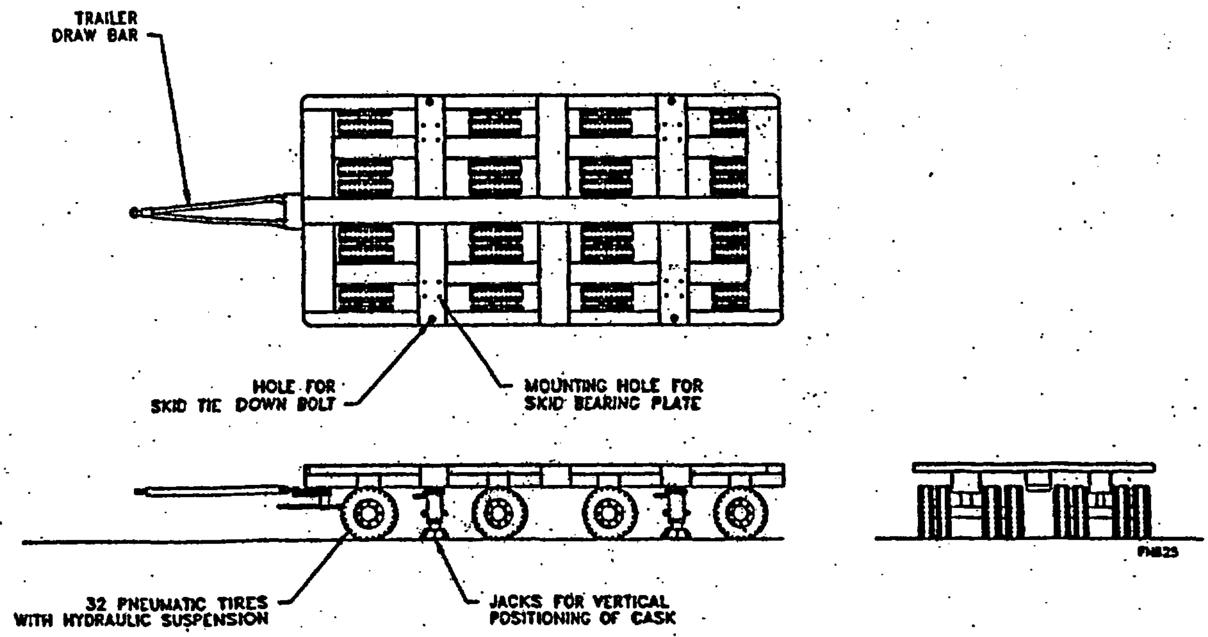


Figure 1-7

Transport Trailer for OS187H Transfer Cask (Typical)

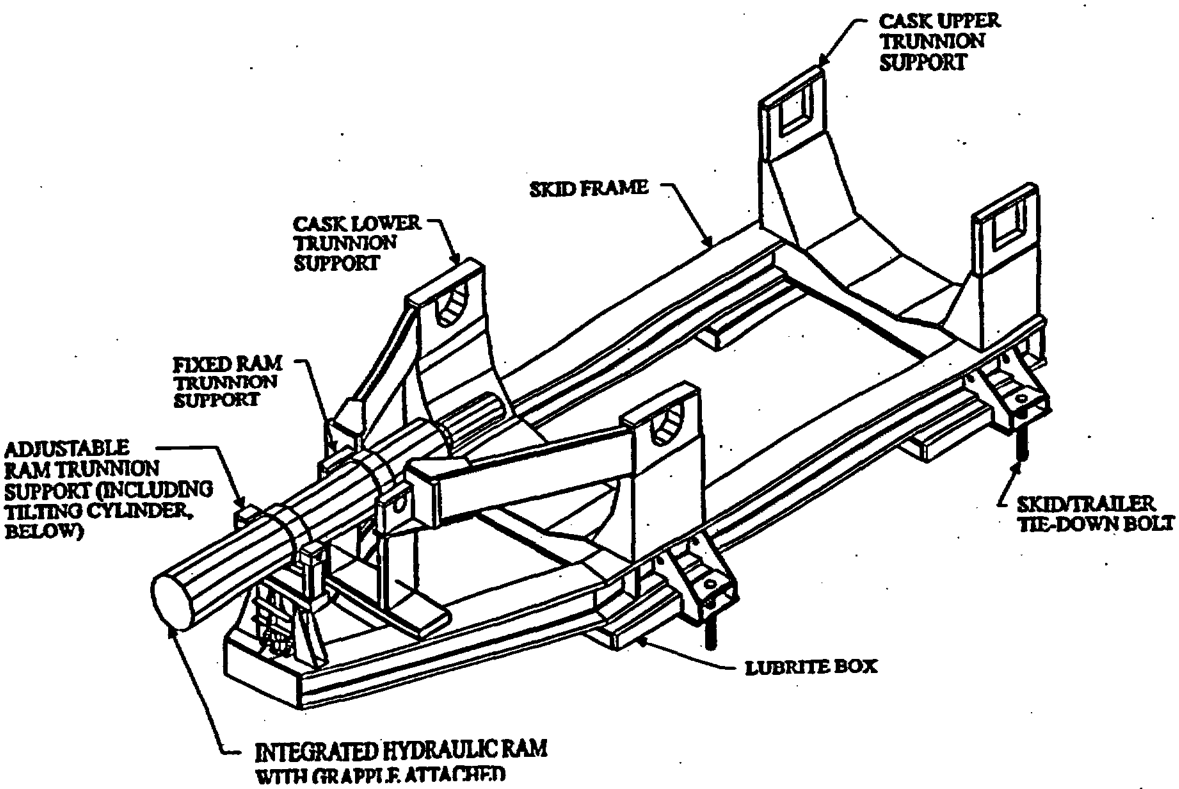
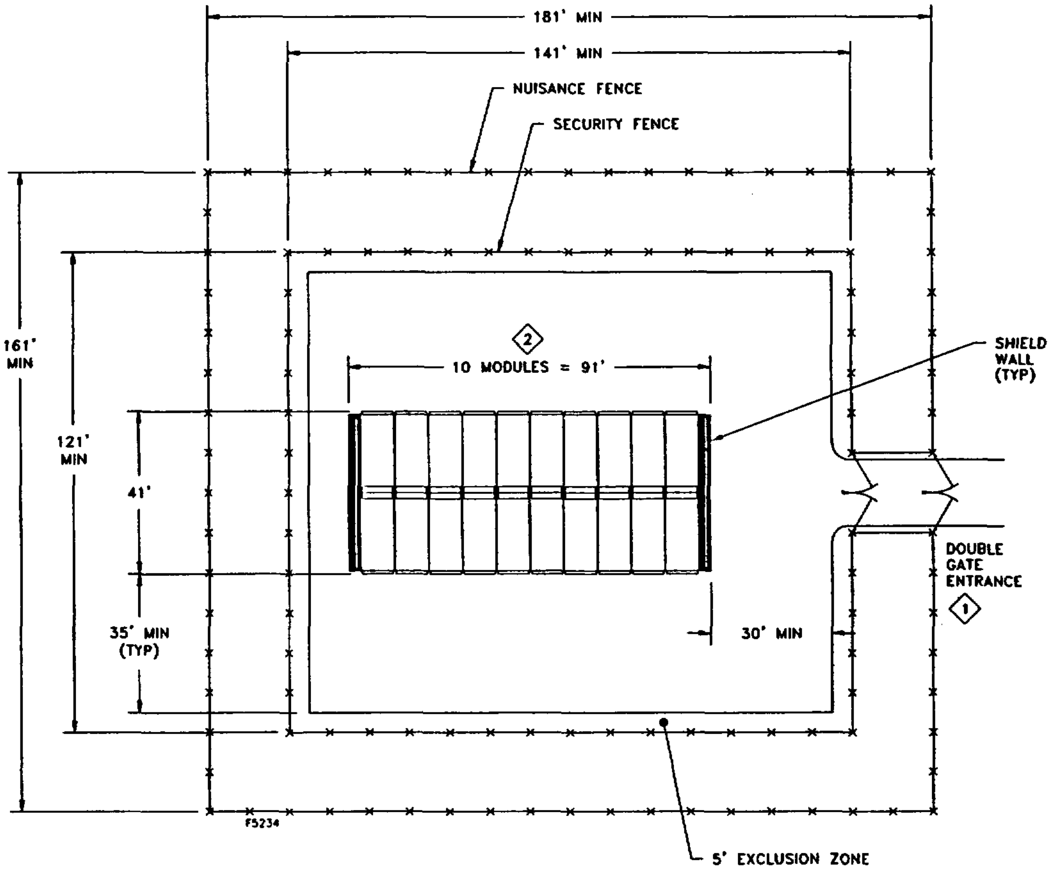


Figure 1-8

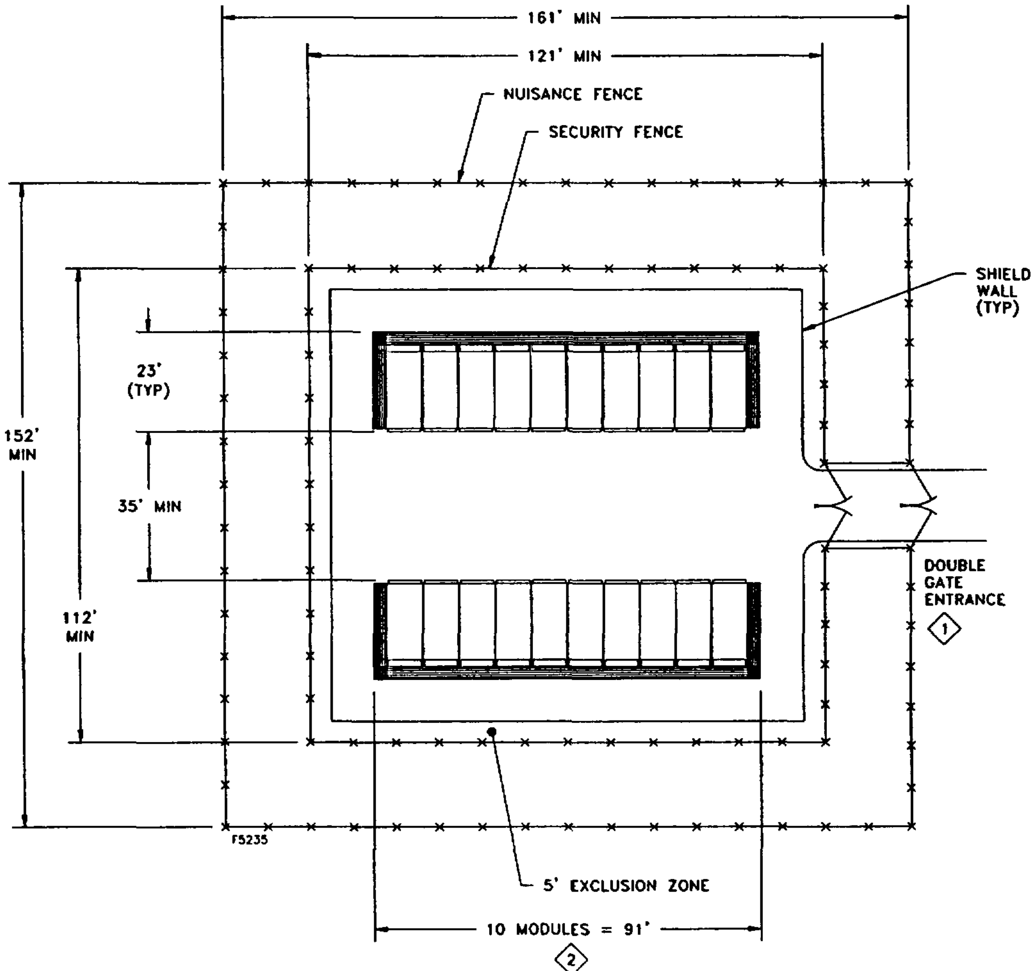
Cask Support Skid for OS187H Transfer Cask (Typical)



NOTES:

1. LOCATION OF ENTRANCE TO ISFSI TO BE COMPATIBLE WITH PLANT SITE ROADS.
2. NUMBER OF MODULES DETERMINED BY USER BASED ON PLANT DISCHARGE RATES AND DRY STORAGE NEEDS.

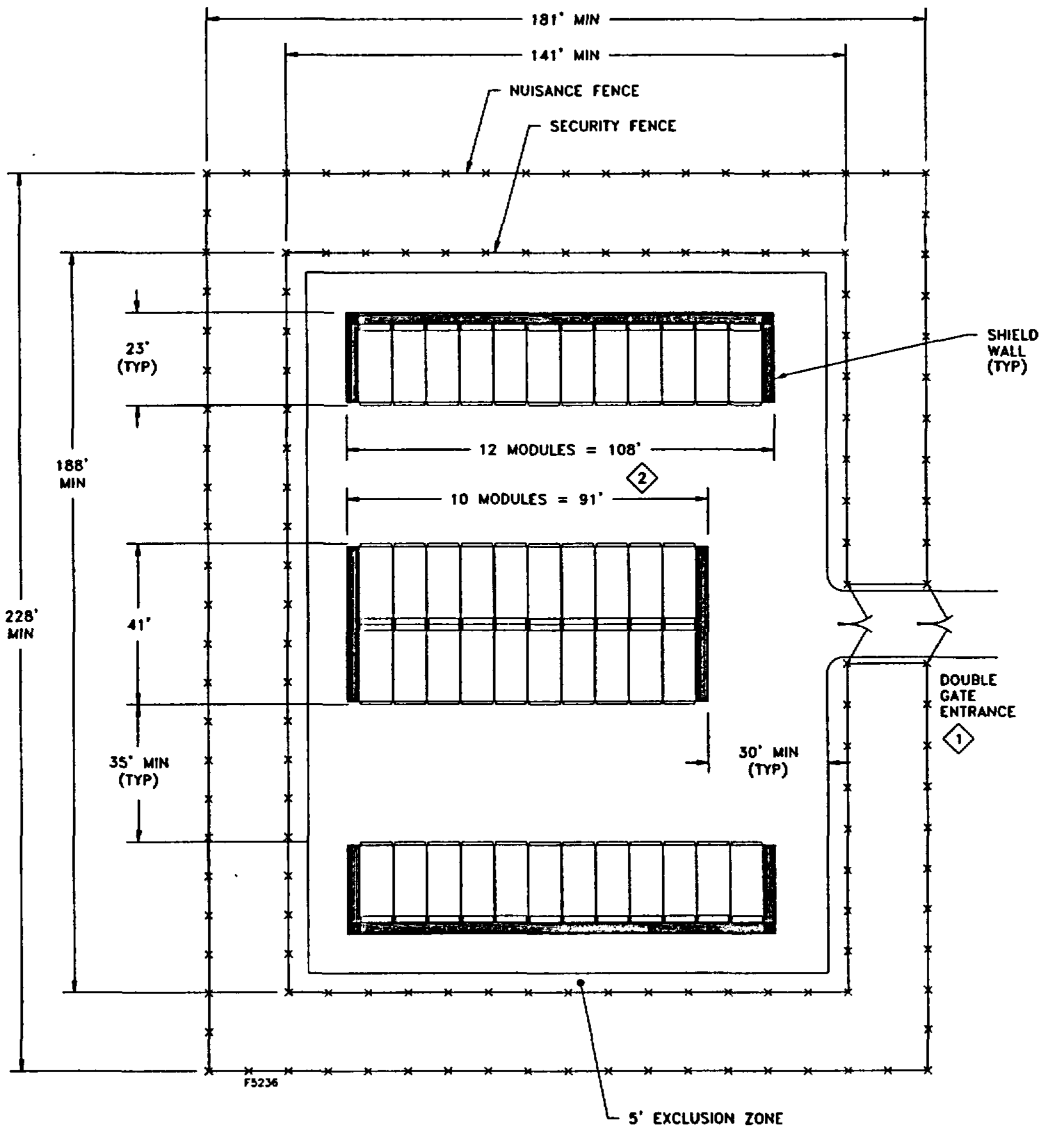
Figure 1-9
Typical Double Module Row HSM-H ISFSI Layout



NOTES:

1. LOCATION OF ENTRANCE TO ISFSI TO BE COMPATIBLE WITH PLANT SITE ROADS.
2. NUMBER OF MODULES DETERMINED BY USER BASED ON PLANT DISCHARGE RATES AND DRY STORAGE NEEDS.

Figure 1-10
Typical Single Module Row HSM-H ISFSI Layout



NOTES:

1. LOCATION OF ENTRANCE TO ISFSI TO BE COMPATIBLE WITH PLANT SITE ROADS.
2. NUMBER OF MODULES DETERMINED BY USER BASED ON PLANT DISCHARGE RATES AND DRY STORAGE NEEDS.

Figure 1-11
Typical Combined Single and Double Module Row HSM-H ISFSI Layout

Drawings

(PROPRIETARY INFORMATION)

CHAPTER 2
PRINCIPAL DESIGN CRITERIA

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2. PRINCIPAL DESIGN CRITERIA

2.1 Spent Fuel to be Stored

The NUHOMS® HD System components have currently been designed for the storage of 32 intact and or up to 16 damaged with remaining intact, Westinghouse 15x15 (WE 15x15), Westinghouse 17x17 (WE 17x17) and/or Framatome ANP Advanced 17x17 MK BW (FR 17x17) PWR fuel assemblies. Additional payloads may be defined in future amendments to this application.

The thermal and radiological characteristics for the PWR spent fuel were generated using the SCALE computer code package [1]. The physical characteristics for the PWR fuel assembly types are shown in Table 2-1. Free volume in the 32PTH DSC cavity is addressed in Chapter 4. Specific gamma and neutron source spectra are given in Chapter 5.

Although analyses in this SAR are performed only for the design basis fuel, any other intact or damaged PWR fuel which falls within the geometric, thermal, and nuclear limits established for the design basis fuel can be stored in the 32PTH DSC.

2.1.1 Detailed Payload Description

This payload consists of 32 PWR UO_2 fuel assemblies with or without Non-Fuel Assembly Hardware (NFAH) which includes Burnable Poison Rod Assemblies, (BPRAs), Vibration Suppression Inserts (VSI) or Thimble Plug Assemblies (TPAs). Each 32PTH DSC can accommodate a maximum of sixteen damaged fuel assemblies, with the remaining assemblies intact. The fuel to be stored in the 32PTH DSC is limited to fuel with a maximum assembly average initial enrichment of 5.00 weight % U-235. The maximum allowable burnup is given as a function of initial fuel enrichment but does not exceed 60,000 MWd/MTU. The minimum cooling time is five years.

The 32PTH DSC may store up to 32 PWR fuel assemblies arranged in accordance with a heat load zoning configuration as shown in Figure 2-1, with a maximum decay heat of 1.5 kW per assembly and a maximum heat load of 34.8 kW per DSC,

The 32PTH DSC can accommodate up to 16 damaged fuel assemblies which include assemblies with known or suspected cladding defects greater than hairline cracks or pinhole leaks, up to and including broken rods, portions of broken rods and rods with missing sections. Damaged fuel assemblies shall be placed into the sixteen inner most basket fuel compartments, as shown in Figure 2-2, which contain top and bottom end caps that confine any loose material and gross fuel particles to a known, subcritical volume during normal, off-normal and accident conditions and to facilitate handling and retrievability. A visual inspection of assemblies will be performed prior to placement of the fuel in the 32PTH DSC, which may then be placed in storage or transported anytime thereafter without further fuel inspection.

The NUHOMS®-32TH DSC basket is designed with three alternate poison materials: Borated Aluminum alloy, Boron Carbide/Aluminum Metal Matrix Composite (MMC) and Boral®. For

criticality analysis, 90% of B10 content present in the borated aluminum and MMC poison plates is credited, while only 75% is credited for Boral®.

The NUHOMS®-32PTH DSC basket is analyzed for seven alternate basket configurations, depending on the boron loadings and poison materials.

A summary of the alternate poison loadings considered for each poison material as a function of basket types is presented below:

NUHOMS®-32PTH DSC Basket Type	Minimum B10 Aerial Density, gm/cm^2	
	Natural or Enriched Boron Aluminum Alloy / Metal Matric Composite (MMC)	Boral®
IA or IIA	0.007	0.009
IB or IIB	0.015	0.019
IC or IIC	0.020	0.025
ID	0.032	N/A
IE	0.050	N/A

Table 2-2 shows a parametric equation that can be utilized to qualify spent fuel assemblies for the defined decay heat load zones. The decay heat load can be calculated based on a fuel assembly's burnup, cool time, and initial enrichment parameters. This table ensures that the fuel assembly decay heat load is within the appropriate zone. The development of this equation is provided in Appendix 4.16.2.

The maximum fuel cladding temperature limit of 400°C (752°F) is applicable to normal conditions of storage and all short term operations from spent fuel pool to ISFSI pad including vacuum drying and helium backfilling of the NUHOMS®-32PTH DSC per Interim Staff Guidance (ISG) No. 11, Revision 2 [15]. In addition, ISG-11 does not permit thermal cycling of the fuel cladding with temperature differences greater than 65°C (117°F) during DSC drying, backfilling and transfer operations.

The maximum fuel cladding temperature limit of 570°C (1058°F) is applicable to accidents or off-normal thermal transients [15].

Calculations were performed to determine the fuel assembly type which was most limiting for each of the analyses including shielding, criticality, thermal and confinement. These evaluations are performed in Chapters 5 and 6. The fuel assembly classes considered are listed in Table 2-1. It was determined that the Framatome 17x17 is the enveloping fuel design for the shielding, thermal and confinement source term calculation because of its total assembly weight and highest initial heavy metal loading. The bounding source term for shielding analysis is given in Table 2-3. Table 2-4 presents the thermal and radiological source terms for the Non-Fuel Assembly Hardware (NFAH).

These values are consistent with the cumulative exposures and cooling times of the fuel assemblies. The gamma spectra for the bounding fuel assembly and NFAH are presented in Chapter 5.

The shielding evaluation is performed assuming 32 fuel assemblies with the parameters (1.5kW) shown in Table 2-3. Any fuel assembly that is thermally qualified by Table 2-2 is also acceptable from a shielding perspective since only eight (8) Zone 3, (1.5 kW max), fuel assemblies are allowed in the 32PTH DSC.

For criticality safety, the WE 17x17 standard assembly is the most reactive assembly type for a given enrichment. This assembly is used to determine the most reactive configuration in the DSC. Using this most reactive configuration, criticality analysis for all other fuel assembly classes is performed to determine the maximum enrichment allowed as a function of the soluble boron concentration and fixed poison plate loading. The analyses results are presented in Chapter 6.

For calculating the maximum internal pressure in the NUHOMS®-32PTH DSC, it is assumed that 1% of the fuel rods are damaged for normal conditions, up to 10% of the fuel rods are damaged for off normal conditions, and 100% of the fuel rods will be damaged following a design basis accident event. A minimum of 100% of the fill gas and 30% of the fission gases within the ruptured fuel rods are assumed to be available for release into the DSC cavity, consistent with NUREG-1536 [17].

The maximum internal pressures used in the structural analysis for the NUHOMS®-32PTH DSC are 15, 20, and 120 psig for normal, off-normal and accident conditions, respectively, during storage and transfer operations and 70 psig during storage accident conditions.

2.2 Design Criteria for Environmental Conditions and Natural Phenomena

The 32PTH DSC and HSM-H form a self-contained, independent, passive system, which does not rely on any other systems or components for its operation. The criterion used in the design of the 32PTH DSC and HSM-H ensures that their exposure to credible site hazards does not impair their safety functions.

The design criteria satisfy the requirements of 10CFR Part 72 [2]. They include the effects of normal operation, natural phenomena and postulated man-made accidents. The criteria are defined in terms of loading conditions imposed on the 32PTH DSC. The loading conditions are evaluated to determine the type and magnitude of loads induced on the 32PTH DSC. The combinations of these loads are then established based on the conditions that can be superimposed. The load combinations are classified by Service Level consistent with Section III of the ASME Boiler and Pressure Vessel Code [3]. The stresses resulting from the application of these loads are then evaluated based on the rules for a Class I nuclear component prescribed by Subsection NB of the Code for the 32PTH DSC Shell Assembly important to safety components. Subsections NG and NF of the Code apply to the 32PTH DSC Basket Assembly. The HSM-H loads and load combinations are developed in accordance with the requirements of ANSI 57.9 [4] and ASCE 7-95 [5]. The HSM-H component stresses are evaluated based on the applicable ACI and AISC standards specified.

2.2.1 Tornado and Wind Loadings

The NUHOMS® HD System is designed to resist the most severe tornado and wind loads specified by NRC Regulatory Guide 1.76 [6] and NUREG-0800 [7]. The HSM-H is designed to safely withstand tornado missiles as defined by 10CFR 72.122(b) (2). Extreme wind effects are much less severe than the specified design basis tornado wind forces, which are used in load combinations specifying extreme wind for the design of the HSM-H.

There are no credible wind loads applied to the 32PTH DSC as the HSM-H and transfer cask provide the required environmental protection. The case of the canister inside the HSM-H is evaluated in Chapter 3 for the associated pressure drop condition.

Since the NUHOMS® HD System on-site transfer cask (TC) is used infrequently and for short durations, the possibility of a tornado funnel cloud enveloping the TC/32PTH DSC during transit to the HSM-H is a low probability event. Nevertheless, the TC is designed for the effects of tornadoes, in accordance with 10CFR 72.122 which includes design for the effects of worst case tornado winds and missiles [7]. Analyses are presented in Chapter 11.

2.2.1.1 Applicable Design Parameters

The design basis tornado (DBT) intensities used for the HSM-H are obtained from NRC Regulatory Guide 1.76 [6]. Region I intensities are utilized since they result in the most severe loading parameters. The maximum wind speed is 360 mph which is the sum of the rotational speed of 290 mph plus the maximum translational speed of 70 mph. The radius of the maximum rotational speed is 150 feet, the pressure drop across the tornado is 3.0 psi, and the rate of pressure drop is 2.0 psi per second.

2.2.1.2 Determination of Forces on Structures

The effects of a DBT are evaluated for the HSM-H. Tornado loads are generated for three separate loading phenomena:

1. Pressure or suction forces created by drag as air impinges and flows past the HSM-H with a maximum tornado wind speed of 360 mph,

2. Suction forces due to a tornado generated pressure drop or differential pressure load of 3 psi, and

3. Impact forces created by tornado-generated missiles impinging on the HSM-H.

The determination of the DBT velocity pressure is in accordance with the requirements of ASCE 7-95 [5]. The resistance to overturning and sliding of the HSM-H under these design pressures is evaluated in Chapter 3, Appendix 3.9.9.

2.2.1.3 Tornado Missiles

The determination of impact forces created by DBT generated missiles for the HSM-H is based on the criteria provided by NUREG-0800 [7], Section 3.5.1.4, III.4. Accordingly, eight types of missiles are postulated:

1. The utility wooden pole, 13.5" diameter, 35' long missile weighing 1500 lbs at a horizontal velocity of 294 fps.

2. The armor piercing artillery shell 8" diameter, weighing 276 lbs at a horizontal velocity of 185 fps.

3. The steel pipe missile 12" diameter, Schedule 40, 30' long weighing 1500 lbs at a horizontal velocity of 205 fps.

4. The massive automobile missile weighing 4000 lbs at a horizontal velocity of 195 fps traveling through the air not more than 25 ft above the ground and having contact area of 20 square ft.

5. Wood plank missiles traveling end on, 200 lbs, traveling at 440 fps.

6. Steel Pipe 3" diameter, Sch 40, weighing 115 lbs, traveling at 268 fps.

7. Steel Pipe 6" diameter, Sch 40, 285 lbs, traveling at 230 fps.

8. Steel rod, 1" diameter, 3' long weighing 8 lbs traveling at 317 fps.

In determining the overall effects of a DBT missile impact, overturning, and sliding of the HSM-H, the force due to the deformable massive missile impact is applied to the structure at the most adverse location. Conservation of momentum is used to demonstrate that sliding and/or tipping of a single

module will not result in an unacceptable condition for the module. The coefficient of restitution is conservatively assumed to be zero so that 100% of the missile energy is transferred to the module.

The missile energy is assumed to be dissipated as sliding friction, or an increase in potential energy due to raising the center of gravity with the force evenly distributed over the impact area. These overall effects are evaluated in Chapters 3, Appendix 3.9.9.

For the local damage analysis of the HSM-H for DBT missiles, the postulated missiles shall be used for the evaluation of concrete penetration, scabbing and perforation thickness. The modified NDRC empirical formula shall be used for this evaluation as recommended in NUREG-0800, Section 3.5.3.

Evaluation for the effects of small diameter solid spherical missiles on the 32PTH DSC is not required, as there are no openings in the HSM-H which lead directly to the canister.

2.2.2 Water Level (Flood) Design

The 32PTH DSC and HSM-H are designed for an enveloping design basis flood, postulated to result from natural phenomena such as tsunami, and seiches, as specified by 10CFR 72.122(b). For the purpose of this generic evaluation, a flood height of 50 feet with a water velocity of 15 fps is used. The 32PTH DSC is subjected to an external hydrostatic pressure equivalent to the 50 feet head of water or 21.7 psi. The HSM-H is evaluated for the effects of a water current of 15 fps impinging on the sides of a submerged HSM-H. For the flood case that submerges the HSM-H, the inside of the HSM-H will be rapidly filled with water through the HSM-H vents. Therefore, the HSM-H components are not evaluated for the resulting static head of water. The effects of flooding and submergence on the canister are addressed in Chapters 3, 4 and 11.

2.2.2.1 Flood Elevations

It is anticipated that the 32PTH DSC and HSM-H will be located on flood-dry sites. However, as stated above, the HSM-H and 32PTH DSC are designed for a flood elevation 50 ft. above the base of the HSM-H.

2.2.2.2 Phenomena Considered in Design Load Calculations

The HSM-H is designed to withstand loads from forces developed by the probable maximum flood including dynamic phenomena such as momentum and drag. The 32PTH DSC is designed for the hydrostatic head equal to 50 ft. water submergence.

2.2.2.3 Flood Force Application

All flood loadings and effects from floods on the NUHOMS® HD System are discussed in Chapters 3 and 11.

2.2.2.4 Flood Protection

Flood protection measures for the NUHOMS® HD System are discussed in Chapters 3 and 11.

2.2.3 Seismic Design

Seismic design criteria are dependent on the specific site location. These criteria are established based on the general requirements as stated in 10CFR Part 72.102.

The design earthquake (DE) for use in the design of the casks must be equivalent to the safe shutdown earthquake (SSE) for a co-located nuclear power plant, the site of which has been evaluated under the criteria of 10CFR 100, Appendix A[8].

2.2.3.1 Input Criteria

The seismic design criteria for the HSM-H is based on the NRC Regulatory Guide 1.60 (R.G.) [9]. The response spectra is anchored to a maximum ground acceleration of 0.30g for the horizontal components and 0.20g for the vertical component. The results of the frequency analysis of the HSM-H structure (which includes a simplified model of the DSC) yield a lowest frequency of 23.2 Hz in the transverse direction and 28.4 Hz in the longitudinal direction. The lowest vertical frequency exceeds 33 Hz. Thus, based on the R.G. 1.60 response spectra amplifications, the corresponding seismic accelerations used for the design of the HSM-H are 0.37g and 0.33g in the transverse and longitudinal directions respectively and 0.20g in the vertical direction. The corresponding accelerations applicable to the DSC are 0.41g and 0.36g in the transverse and longitudinal directions, respectively, and 0.20g in the vertical direction.

2.2.4 Snow and Ice Loadings

Snow and ice loads for the HSM-H are derived from ASCE 7-95 [5]. The maximum 100 year roof snow load, specified for most areas of the continental United States for an unheated structure, of 110 psf is assumed. There are no credible snow and ice loads applied to the 32PTH DSC as the HSM-H and TC provide the environmental protection. Snow and ice loads for the TC with a loaded 32PTH DSC are negligible due to the smooth curved surface of the cask, the heat rejection of the SFAs, and the infrequent short term use of the cask.

2.2.5 Tsunami

Specific analyses including analyses for tip-over are not done for tsunamis as they are typically bounded by the tornado, wind and flooding load conditions. The licensee should evaluate site specific impacts of a tsunami.

2.2.6 Lightning

A lightning strike will not cause a significant thermal effect on the HSM-H or stored 32PTH DSC. The effects on the HSM-H resulting from a lightning strike are discussed in Chapter 11.

2.2.7 Combined Load Criteria

2.2.7.1 Horizontal Storage Module

The reinforced concrete HSM-H is designed to meet the requirements of ACI 349-97 [10]. The alternate temperature criteria of NUREG-1536 will be utilized as discussed in Chapters 3 (Appendix 3.9.9) and 11. The ultimate strength method of analysis is utilized with the appropriate strength reduction factors as described in Chapter 3 (Appendix 3.9.9). The load combinations specified in ANSI 57.9-1984 [4] are used for combining normal operating, off-normal, and accident loads for the HSM-H. All seven load combinations specified are considered and the governing combinations and the appropriate load factors are presented in Chapter 3 (Appendix 3.9.9).

The resulting HSM-H load combinations and load factors are also presented in Chapter 3 (Appendix 3.9.9). The effects of duty cycle on the HSM-H are considered and found to have negligible effect on the design. The corresponding structural design evaluation for the 32PTH DSC support structure is presented in Chapter 3 (Appendix 3.9.9).

2.2.7.2 32PTH DSC

The 32PTH DSC is designed by analysis to meet the stress intensity allowables of the ASME Boiler and Pressure Vessel Code (1998 Edition with 2000 Addenda) Section III, Division I, Subsection NB as modified by Code Case N-595-3, NG and NF for Class 1 components and supports [3]. The 32PTH DSC is conservatively designed by utilizing linear elastic or non-linear elastic-plastic analysis methods. The load combinations considered for the 32PTH DSC normal, off-normal and postulated accident loadings are described in Chapter 3. ASME Code Service Level A allowables are used for normal and off-normal operating conditions. Service Level D allowables are used for accident conditions such as a postulated cask drop accident. Using these acceptance criteria ensures that in the event of a design basis drop accident, the 32PTH DSC confinement boundary is not breached. Normal operational stresses are combined with the appropriate off-normal and accident stresses. It is assumed that only one postulated accident condition occurs at any one time. The structural evaluation for the 32PTH DSC is documented in Chapter 3.

2.2.7.3 Transfer Cask

The on-site transfer cask is a pressure retaining component (maintain helium backfill) and is designed by analysis to meet the stress allowables of the ASME Code, Subsection NC for Class 2 components. The cask is designed by utilizing linear elastic analysis methods. The load combinations considered for the transfer cask, normal, off-normal, and postulated accident loadings are defined in Chapter 3. Service Level A allowables are used for all normal operating and off-normal conditions. Service Level D allowables are used for load combinations which include postulated accident loadings. Allowable stress limits for upper lifting trunnions are developed to meet the requirements of ANSI N14.6 [11] for non-redundant lifting. The appropriate dead load and thermal stresses are combined with the calculated drop accident scenario stresses to determine the worst case design stresses. The transfer cask structural analyses are presented in Chapter 3.

2.2.8 Burial under Debris

Debris resulting from natural phenomena or accidents that may affect system performance are to be determined by the licensee. Such debris can result from floods, wind storms and land slides. The principal effect is typically on thermal performance. See Chapters 4 and 11 for a generic evaluation of HSM-H blocked vent event.

2.2.9 Thermal Conditions

The NUHOMS® HD System component temperatures and thermal gradients are affected by the following thermal conditions:

- Fuel Loading
- Decay Heat
- Beginning of Storage Unloading
- Ambient Variations (including solar insulation)
- Lightning
- Fire

The thermal conditions which are of concern structurally are the temperature distributions in the system and the differential thermal expansion of interfacing components. See detailed analyses in Chapters 3, 4 and 11.

2.2.9.1 Fuel Loading

The 32PTH DSC/transfer cask is loaded in a spent fuel pool under water. The 32PTH DSC inner surfaces are cooled by pool water and the 32PTH DSC outer surface is cooled by water contained in the 32PTH DSC/transfer cask annulus; therefore, the thermal gradients established during fuel loading will be negligible. During DSC processing, draining and vacuum drying, DSC component temperatures increase. DSC component temperatures are evaluated in Chapter 4.

2.2.9.2 Decay Heat

After the 32PTH DSC/transfer cask is loaded and removed from the pool, the temperatures will gradually reach steady state conditions. The temperature gradients in the 32PTH DSC/TC have an insignificant effect on structural integrity.

The 32PTH DSC is designed for zoned loading as a function of decay heat. Four zones are designated: 1a, 1b, 2 and 3, with the maximum decay heat in zone 3. Details of the zone loading are discussed and evaluated in Chapter 4.

Thermal analysis calculations were performed for different ambient and decay heat load conditions. The methods used to obtain these results are discussed in Chapter 4. The effect on structural integrity is addressed in Chapters 3 and 11.

2.2.9.3 Beginning of Storage Unloading

Beginning of storage unloading would occur if it were necessary to place the 32PTH DSC back into the pool at the beginning of storage after it had been loaded and reached thermal equilibrium. Prior to unloading fuel, the 32PTH DSC and fuel would be cooled by circulating water through the 32PTH DSC. Therefore, cool water would contact the hotter 32PTH DSC inner surfaces. The thermal gradients in the 32PTH DSC body due to this condition are small and would have an insignificant effect on the cask body. The fuel cladding stresses during beginning of storage unloading is evaluated in Chapters 3 and 4.

2.2.9.4 Ambient Variations

Because the combined HSM-H and 32PTH DSC thermal inertia is large, the 32PTH DSC temperature response to changes in atmospheric conditions will be relatively slow. Ambient temperature variations due to changes in atmospheric conditions i.e., sun, ice, snow, rain and wind will not affect the performance of the 32PTH DSC. The cyclical variation of insolation during a day will also create insignificant thermal gradients.

The thermal effects due to ambient variations and conditions are discussed in further detail in Chapter 4.

2.2.9.5 Lightning

Thermal effects due to lightning are discussed in Chapter 11.

2.2.9.6 Fire

It is demonstrated in Chapter 11 that the 32PTH DSC will maintain confinement integrity during and after the postulated fire accident.

2.3 Safety Protection Systems

2.3.1 General

The NUHOMS® HD System is designed to provide long term storage of spent fuel. The canister materials are selected such that degradation is not expected during the storage period. The 32PTH DSC cylindrical shell, the top shield plug, and bottom form the pressure retaining confinement boundary for the spent fuel. The top shield plug and bottom provide shielding for the 32PTH DSC so that occupational doses are minimized during drying, sealing, and handling operations. The 32PTH DSC top closure has redundant welds which join the shell and the top cover plate assemblies to form the confinement boundary as defined by ASME Code Case N-595-3 [3]. The 32PTH DSC shell and bottom end assembly confinement boundary weld is made during fabrication of the 32PTH DSC. The top closure confinement welds are made after fuel loading. Both top plug penetrations (siphon and vent ports) are redundantly welded after drying operations are complete.

The NUHOMS® HD System is designed for safe and secure, long-term confinement and dry storage of SFAs. The key elements of the NUHOMS® HD System and their operation which require special design consideration are:

A. Minimizing the contamination of the 32PTH DSC exterior by fuel pool water.

B. The 32PTH DSC top end, double closure welds form dual pressure retaining confinement boundaries and maintains a helium atmosphere.

C. Minimizing personnel radiation exposure during 32PTH DSC loading, closure, and transfer operations.

D. Design of the HSM-H, OS187H TC, and 32PTH DSC for postulated accidents.

E. Design of the HSM-H passive ventilation system for effective decay heat removal to ensure the integrity of the fuel cladding. The HSM-H is designed with no active safety systems.

F. Design of the 32PTH DSC to ensure subcriticality.

G. Design of the OS187H TC for shielding, protection, and efficient operability.

2.3.2 Protection by Multiple Confinement Barriers and Systems

2.3.2.1 Confinement Barriers and Systems

The radioactive material which the NUHOMS® HD System confines is the spent fuel assemblies and the associated contaminated or activated materials.

During fuel loading operations, the radioactive material in the plant's fuel pool is prevented from contacting the 32PTH DSC exterior by filling the cask/32PTH DSC annulus with

uncontaminated, demineralized water prior to placing the cask and 32PTH DSC in the fuel pool. In addition, the cask/32PTH DSC annulus opening at the top of the cask is sealed using an inflatable seal to prevent pool water from entering the annulus. This procedure minimizes the likelihood of contaminating the 32PTH DSC exterior surface. The combination of the above operations assures that the 32PTH DSC surface loose contamination levels are within those required for shipping cask externals. Compliance with these contamination limits is assured by taking surface swipes of the upper end of the 32PTH DSC before transferring the cask from the fuel building.

Once inside the 32PTH DSC, the SFAs are confined by the 32PTH DSC shell and the top and bottom cover plates. The fuel cladding integrity is ensured by maintaining the storage cladding temperatures below levels which are known to cause degradation of the cladding. In addition, the SFAs are stored in an inert atmosphere to prevent degradation of the cladding, specifically cladding rupture due to oxidation and its resulting volumetric expansion of the fuel. Thus, a helium atmosphere for the 32PTH DSC is incorporated in the design to protect the fuel cladding integrity by inhibiting the ingress of oxygen into the cavity.

Helium is known to leak through valves, mechanical seals, and escape through very small passages because it has a small atomic diameter, is an inert element, and exists in a monatomic species. Helium will not, to any practical extent, diffuse through stainless steel. For this reason the 32PTH DSC has been designed as a welded confinement pressure vessel with no mechanical or electrical penetrations and meets the leaktight criteria in accordance with the guidance provided in ISG-18 [12]. See Chapter 7 for a detailed discussion of the confinement boundary design.

The 32PTH DSC itself has a series of barriers to ensure the confinement of radioactive materials. The cylindrical shell is fabricated from rolled ASME stainless steel plate which is joined with full penetration welds that are 100% inspected by non-destructive examination. All top and bottom end closure welds are multiple-layer welds. This effectively eliminates any pinhole leaks which might occur in a single pass weld, since the chance of pinholes being in alignment on successive weld passes is not credible. Furthermore, the cover plates are sealed by separate, redundant closure welds. Pressure boundary welds and welders are qualified in accordance with Section IX of the ASME B&PV Code and inspected according to the appropriate articles of Section III, Division 1, Subsection NB. These criteria insure that the as-deposited weld filler metal is as sound as the parent metal of the pressure vessel.

Pressure monitoring instrumentation is not used since penetration of the pressure boundary would be required. The penetration itself would then become a potential leakage path and by its presence compromise the integrity of the 32PTH DSC design. The shell and welded cover plates provide total confinement of radioactive materials. Once the 32PTH DSC is sealed, there are no credible events, as discussed in Chapter 11, which could fail the cylindrical shell or the closure plates which form the confinement boundary.

2.3.2.2 32PTH DSC Cooling

The HSM-H provides a means of removing spent fuel decay heat by a combination of radiation, conduction, and natural convection. The passive convective ventilation system is driven by the pressure difference due to the stack buoyancy effect (ΔP_s) provided by the temperature difference between the 32PTH DSC and the ambient air outlet. This pressure difference is larger than the flow pressure drop (ΔP_f) at the design air inlet and outlet temperatures.

There are no radioactive releases of effluents during normal and off-normal storage operations. Also, there are no credible accidents which cause releases of radioactive effluents from the 32PTH DSC. Therefore, an off-gas monitoring system is not required for the HSM-H. The only time an off-gas system is required is during 32PTH DSC drying operations. During this operation, the spent fuel pool or plant's radwaste system is used to process the air and helium evacuated from the 32PTH DSC.

During transfer of the DSC from the reactor building to the HSM, cooling of the DSC is maintained by utilizing a helium environment inside the transfer cask.

2.3.3 Protection by Equipment and Instrumentation Selection

2.3.3.1 Equipment

The HSM-H, 32PTH DSC, and transfer cask encompass equipment which is important to safety. Other equipment important to safety associated with the NUHOMS® 32PTH System includes the equipment required for handling operations within the plant's fuel/reactor building. This equipment is regulated by the plant's 10CFR 50 [16] operating license.

2.3.3.2 Instrumentation

The NUHOMS® HD System is a totally passive system. No safety-related instrumentation is necessary for monitoring the 32PTH DSC. The maximum temperatures and pressures are conservatively bounded by analyses. Therefore, there is no need for monitoring the internal cavity of the 32PTH DSC for pressure or temperature during normal operations. The 32PTH DSC is conservatively designed to perform its confinement function during all worst case normal, off-normal, and postulated accident conditions.

2.3.4 Nuclear Criticality Safety

2.3.4.1 Control Methods for Prevention of Criticality

The design criteria for criticality is that an upper sub-critical limit (USL) of 0.95 minus statistical uncertainties and bias, shall be limiting for all postulated arrangements of fuel within the canister. The 32PTH DSC incorporates borated aluminum material(s) as fixed neutron absorbing materials to provide criticality control. Criticality control is discussed in Chapter 6.

The 32PTH DSC is designed to assure an ample margin of safety against criticality under the conditions of fresh fuel (fuel without burnup credit) in a canister flooded with borated pool

water. The methods of criticality control are in accordance with the requirements of 10CFR 72.124 [2].

Criticality analysis is performed using the SCALE computer code package [1]which is widely used for criticality analysis of shipping casks, fuel storage pools and storage systems. Benchmark problems are run to verify the codes, methodology and cross section library and to determine calculational bias and uncertainties. Chapter 6 of the SAR presents the NUHOMS® HD System criticality analyses.

In the criticality calculation, the fuel assemblies and canister geometries are explicitly modeled. Each fuel pin and each guide tube is represented within each assembly.

Reactivity analyses were performed for WE 15x15 and WE 17x17 classes of fuel assemblies. These analyses do not credit the neutron absorption capability of the NFAH.

2.3.4.2 Error Contingency Criteria

Provision for error contingency is built into the criterion used in Section 2.3.4.1. The criterion is common practice for licensing submittals. Because conservative assumptions are made in modeling, it is not necessary to introduce additional contingency for error.

2.3.4.3 Verification Analysis-Benchmarking

Evaluation and verification against critical benchmarking experiments are described in Chapter 6, Section 6.5.

2.3.5 Radiological Protection

The NUHOMS® HD System ISFSI is designed to maintain on-site and off-site doses as low as reasonably achievable (ALARA) during transfer operations and long-term storage conditions. ISFSI operating procedures, shielding design, and access controls provide the necessary radiological protection to assure radiological exposures to station personnel and the public are ALARA. Further details concerning on-site and off-site dose rates resulting from NUHOMS® 32PTH HD System, ISFSI operations and the ISFSI ALARA evaluation are provided in Chapter 10.

2.3.5.1 Access Control

The NUHOMS® HD System ISFSI will typically be located within the owner controlled area of an operating plant. A separate protected area consisting of a double fenced, double gated, lighted area may be installed around the ISFSI. Access is then controlled by locked gates, and guards are stationed when the gates are open. The licensee's Security Plan must describe the devices employed to detect unauthorized access to the facility. The specific procedures for controlling access to the ISFSI site and the restricted area within the site per 10CFR 72, Subpart H shall be addressed by the licensee's physical security and safeguards contingency plans. The system will not require the continuous presence of operators or maintenance personnel.

In addition to the controlled access, a method of providing a security tamper seal on the HSM-H door may be included after insertion of a loaded 32PTH DSC. This may be, but is not limited to, one of the following:

Tack welding the HSM-H access door

Tack welding 2 or more closure bolts on the HSM-H access door

Tamper seals

2.3.5.2 Shielding

Shielding has the objective of assuring that radiation dose rates at key locations are at acceptable levels for those locations. Three locations are of particular interest:

(1) Immediate Vicinity of the HSM-H

(2) Restricted Area Boundary

(3) Controlled Area Boundary

Dose rates in the immediate vicinity of the HSM-H are important for consideration of occupational exposure. Because of the passive nature of storage with this system, occupational tasks related to the system are infrequent and short in duration. Expected personnel exposures due to operational and maintenance activities are discussed in Chapter 10, Section 10.3. The estimated occupational doses for personnel comply with applicable requirements (occupational dose limits).

Restricted area boundaries may be selected so that monitoring of radiation exposure to people outside the restricted area is not required. Dose rates at the controlled area boundary are in accordance with applicable regulatory guides.

2.3.5.3 Radiological Alarm System

There are no radiological alarms required for the NUHOMS® HD System. There are no credible events which result in releases of radioactive products or unacceptable increases in direct radiation.

2.3.6 Fire and Explosion Protection

The NUHOMS® HD System HSM-H and 32PTH DSC do not contain flammable materials. The concrete and steel used for their fabrication can withstand any credible fire hazard. There is no fixed fire suppression system within the boundaries of the ISFSI. An evaluation of the system engulfed in a minor fuel fire is provided in Chapter 11. Due to the large thermal mass of the HSM-H, any minor fires in the vicinity of the ISFSI would raise the HSM-H temperature by only a few degrees and will not affect the confinement capability of the 32PTH DSC.

ISFSI initiated explosions are not considered credible since explosive materials are not present in the fission product or cover gases within the 32PTH DSC cavity. Externally initiated explosions are considered to be bounded by the design basis tornado generated missile load analysis. As indicated in Chapter 11, overpressures of a few psi can be conservatively postulated to occur at the ISFSI as a result of accidents involving explosive materials which are stored or transported near the site. This impact is significantly less than that postulated to result from the tornado wind loading and missile impact analysis, as described in Section 2.2.1, and is well within the design basis of the HSM-H. The licensee will evaluate the site specific external hazards to demonstrate these are bounded by the tornado effects.

2.3.7 Acceptance Tests and Maintenance

2.3.7.1 Acceptance Test

The acceptance tests and criteria for visual inspections, leak testing of components, valves, gaskets, shielding integrity, thermal acceptance and neutron absorbers are discussed in Chapter 9.

2.3.7.2 Maintenance Program

Because of their passive nature, the storage modules will require little, if any, maintenance over the lifetime of the ISFSI. The maintenance program is discussed in Chapter 9.

2.4 Decommissioning Considerations

The 32PTH DSC is designed to interface with a 10CFR 71 [13] transportation system for the eventual off-site transport of canisters by the DOE to either a monitored retrievable storage facility (MRS) or a permanent geologic repository, as discussed in Chapter 14.

Decommissioning of the NUHOMS® HD System ISFSI will be performed in a manner consistent with the decommissioning of the plant itself since all NUHOMS® HD System components are constructed of materials similar to those found in existing plants. The 32PTH DSC is compatible with wet or dry unloading facilities.

If the fuel is removed from the 32PTH DSC at the plant prior to shipment, the 32PTH DSC will likely be internally contaminated by crud from the spent fuel and may be slightly activated by spontaneous neutron emissions from the spent fuel. The 32PTH DSC internals may be cleaned to remove surface contamination and disposed of as low-level waste. Alternatively, if the contamination and activation levels are small enough (to be determined on a case-by-case basis), it may be possible to decontaminate the 32PTH DSC and dispose of it as commercial scrap.

While the intent for the NUHOMS® HD System includes the eventual disposal of each 32PTH DSC should fuel removal be required, current closure weld designs do not preclude future development of a non-destructive closure removal technique that allows for reuse of the 32PTH DSC shell/basket assembly. Economic and technical conditions existing at the time of fuel removal would be assessed prior to making a decision to reuse the 32PTH DSC.

The exact decommissioning plan for the ISFSI will be dependent on the DOE's fuel transportation system capability and requirements for a specific plant. Because of the minimal contamination of the outer surface of the 32PTH DSC, no contamination is expected on the internal surfaces of an HSM-H. It is anticipated that the prefabricated HSM-Hs can be dismantled and disposed of using commercial demolition and disposal techniques. Alternatively, the HSM-Hs may be refurbished and reused at another site or at the MRS for storage of intact 32PTH DSCs transported from the plant.

2.5 Structures, Systems and Components Important to Safety

Table 2-5 provides a list of major NUHOMS® HD System ISFSI components and their classification. Table 2-4 identifies all structures, systems and components that are "Important To Safety" (ITS). Components are classified in accordance with the criteria of 10CFR 72. Structures, systems, and components classified as ITS are defined in 10CFR 72.3 as those features of the ISFSI whose function is:

A. To maintain the conditions required to store spent fuel safely.

B. To prevent damage to the spent fuel container during handling and storage.

C. To provide reasonable assurance that spent fuel can be received, handled, packaged, stored, and retrieved without undue risk to the health and safety of the public.

These criteria are applied to the NUHOMS® HD System components in determining their classification in the paragraphs which follow.

2.5.1 Dry Shielded Canister

The 32PTH DSC provides fuel assembly support required to maintain the fuel geometry for criticality control. Accidental criticality inside a 32PTH DSC could lead to off-site doses comparable with the limits in 10CFR 100 [8] which must be prevented. The 32PTH DSC also provides the confinement boundary for radioactive materials. Therefore, the 32PTH DSC is designed to remain intact under all accident conditions identified in Chapters 3 and 11 without losing its function to provide confinement of the spent fuel assemblies. The 32PTH DSC is designed, constructed, and tested in accordance with a QA program incorporating a graded quality approach for ITS requirements as defined by 10CFR 72, Subpart G, paragraph 72.140(b) and described in Chapter 13.

The welding materials required making the closure welds on the 32PTH DSC inner and outer top cover plates shall be fabricated to the same ASME Code criteria as the 32PTH DSC shell (Subsection NB, Class 1).

2.5.2 Horizontal Storage Module

The HSM-H is considered ITS since it provides physical protection and shielding for the spent fuel container (32PTH DSC) during storage. The reinforced concrete HSM-H is designed in accordance with ACI 349-97 [10] and built to ACI-318 [14]. The level of testing, inspection, and documentation provided during construction and maintenance is in accordance with the quality assurance requirements as defined in 10CFR 72, Subpart G and as described in Chapter 13. Thermal instrumentation for monitoring HSM-H concrete temperatures is considered "Not Important To Safety" (NITS).

2.5.3 ISFSI Basemat and Approach Slabs

The ISFSI basemat and approach slabs are considered NITS and are designed, constructed, maintained, and tested as commercial grade items.

Licensees are required to perform an assessment to confirm that the license seismic criteria described in Section 2.2.3 are met.

2.5.4 Transfer Equipment

2.5.4.1 Transfer Cask and Yoke

The on-site transfer cask OS-187H is ITS since it protects the spent fuel canister (32PTH DSC) during handling and is part of the primary load path used while handling the 32PTH DSC in the fuel/reactor building. An accidental drop of a loaded transfer cask (weighing approximately 115 tons) has the potential for creating conditions in the plant which must be evaluated. These possible drop conditions are evaluated with respect to the impact on the 32PTH DSC in Chapters 3 and 11. Therefore, the transfer cask is designed, constructed, and tested in accordance with a QA program incorporating a graded quality approach for ITS requirements as defined by 10CFR 72, Subpart G, paragraph 72.140(b) and described in Chapter 13.

A lifting yoke is used for handling the transfer cask within the fuel/reactor building and it is used by the licensee (utility) under their 10CFR 50 [16] program requirement.

Due to site unique requirements, rigid or sling lifting members may be used to augment the lifting yoke. These members shall be designed, fabricated and tested in accordance with the same requirements as the cask lifting yoke.

2.5.4.2 Other Transfer Equipment

The NUHOMS® HD System transfer equipment (i.e., ram, skid, transfer trailer) are necessary for the successful loading of the 32PTH DSC into the HSM-H. However, the analyses described in Chapter 11 demonstrate that the performance of these items are not required to provide reasonable assurance that spent fuel can be received, handled, packaged, stored, and retrieved without undue risk to the health and safety of the public. Therefore, these components are considered NITS and need not comply with the requirements of 10CFR 72. These components are designed, constructed, and tested in accordance with good industry practices.

2.5.5 Auxiliary Equipment

The vacuum drying system and the automated welding system are NITS. Performance of these items is not required to provide reasonable assurance that spent fuel can be received, handled, packaged, stored, and retrieved without undue risk to the health and safety of the public. Failure of any part of these systems may result in a delay of operations, but will not result in a hazard to the public or operating personnel. These components are designed, constructed, and tested in accordance with good industry practices.

2.6 References

1. Oak Ridge National Laboratory, RSIC Computer Code Collection, "SCALE: A Modular Code System for Performing Standardized Computer Analysis for Licensing Evaluations for Workstations and Personal Computers," NUREG/CR-0200, Revision 6, ORNL/NUREG/CSD-2/V2/R6.

2. U.S. Government, "Licensing Requirements for the Storage of Spent Fuel in an Independent Spent Fuel Storage Installation (ISFSI)," Title 10 Code of Federal Regulations, Part 72, Office of the Federal Register, Washington, D.C.

3. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section III, Division 1, 1998 Edition through 2000 Addenda, including exceptions allowed by Code Case N-595-3.

4. American National Standards Institute, American Nuclear Society, ANSI/ANS 57.9-1984, Design Criteria for an Independent Spent Fuel Storage Installation (Dry Storage Type).

5. American Society of Civil Engineers, ASCE 7-95, Minimum Design Loads for Buildings and Other Structures, (formerly ANSI A58.1).

6. U.S. Atomic Energy Commission, "Design Basis Tornado for Nuclear Power Plants," Regulatory Guide 1.76 (1974).

7. NUREG-0800, Standard Review Plan, Section 3.3.1, "Wind Loading" and Section 3.5.1.4 "Missiles Generated by Natural Phenomenon".

8. U.S. Government, "Reactor Site Criteria," Title 10 Code of Federal Regulations, Part 100, Office of the Federal Register, Washington, D.C.

9. U.S. Atomic Energy Commission, "Design Response Spectra for Seismic Design of Nuclear Power Plants," Regulatory Guide 1.60, Revision 1 (1973).

10. American Concrete Institute, Code Requirements for Nuclear Safety Related Concrete Structures, ACI 349-97.

11. American National Standards Institute, ANSI N14.6-1993, American National Standard for Special Lifting Device for Shipping Containers Weighing 10,000 lbs. or More for Nuclear Materials.

12. US NRC, Interim Staff Guidance -18, "The Design/Qualification Final Closure Welds on Austenitic Stainless Steel Canisters as Confinement Boundary for Spent Fuel Storage and Containment Boundary for Spent Fuel Transport", May 2003.

13. U.S. Government, "Packaging and Transportation of Radioactive Material" Title 10 Code of Federal Regulations, Part 71, Office of the Federal Register, Washington, D.C.

14. American Concrete Institute, "Building Code Requirements for Reinforced Concrete," ACI-318, 1989 (92).

15. US NRC, Interim Staff Guidance -11, Rev 2, "Cladding Considerations for the Transportation and storage of Spent Fuel", dated July 30, 2002.

16. U.S. Government, "Domestic Licensing of Production and Utilization Facilities," Title 10 Code of Federal Regulations, Part 50, Office of the Federal Register, Washington, D.C.

17. NUREG-1536, "Standard Review Plan for Dry Cask Storage Systems," 1997.

Table 2-1
Spent Fuel Assembly Physical Characteristics

Parameter	WE 15x15	WE 17 x 17 Std	FR 17x17 MK BW	WE 17x17 Vantage 5H	WE 17x17 OFA
Initial Enrichment, wt % U235 (max)	5.00	5.00	5.00	5.00	5.00
Clad material	Zircaloy, Zirlo	Zircaloy, Zirlo	M5	Zircaloy, Zirlo	Zircaloy, Zirlo
No of fuel rods	204	264	264	264	264
No of guide/instrument tubes	21	25	25	25	25
Assembly Length(4)	162.2	162.4	162.4	162.4	162.4
Max Uranium Loading (MTU)	467	467	476	467	467
Assembly Cross Section	8.424 x 8.424	8.426 x 8.426	8.425 x 8.425	8.426 x 8.426	8.426 x 8.426
Max Assembly Weight with NFAHs(5) (lbs)	1528	1575	1554	1533	1533

(1) Nominal values shown unless stated otherwise
(2) All dimensions in inches
(3) Where there are variations in a reported value for a given design, the values highlighted are chosen for the criticality analysis. Thermal analysis uses the minimum values
(4) Includes irradiation growth
(5) Weights of TPAs and VSIs are enveloped by BPRAs

Table 2-2

Fuel Qualification Table(s)

The Decay Heat (DH) in watts is expressed as:

$$F1 = A + B*X1 + C*X2 + D*X1^2 + E*X1*X2 + F*X2^2$$
$$DH = F1*Exp(\{[1-(5/X3)]*G\}*[(X3/X1)^H])*[(X2/X1)^I])$$

where,

F1	Intermediate Function, basically the Thermal source at 5 year cooling
X1	Assembly Burnup in GWD/MTU
X2	Initial Enrichment in wt. % U-235 (max 5% wt)
X3	Cooling Time in Years (min 5 yrs)

A=13.69479 B= 25.79539 C= -3.547739 D= 0.307917 E= -3.809025
F= 14.00256 G= -0.831522 H= 0.078607 I= -0.095900

Examples for Zone 1a -1050 watts (Burnup GWD/MTU)

Enrichment (wt. % U-235)	5 Years	6 Years	7 Years	8 Years	10 Years	15 Years
2.50	34.7	39.2	42.7	45.6	50.0	57.0
3.00	35.5	40.1	43.6	46.5	51.0	57.9
3.50	36.2	40.9	44.5	47.4	52.0	58.9
4.00	36.8	41.5	45.3	48.3	52.8	59.9
4.50	37.2	42.1	45.9	49.0	53.7	60.8

Examples for Zone 1b -800 watts (Burnup GWD/MTU)

Enrichment (wt. % U-235)	5 Years	6 Years	7 Years	8 Years	10 Years	15 Years
2.50	27.7	31.5	34.5	37.0	40.8	46.7
3.00	28.2	32.1	35.2	37.7	41.5	47.5
3.50	28.5	32.5	35.7	38.3	42.2	48.3
4.00	28.5	32.9	36.2	38.8	42.8	49.0
4.50	28.5	33.0	36.4	39.2	43.3	49.7

Examples for Zone 2 -1100 watts (Burnup GWD/MTU)

Enrichment (wt. % U-235)	5 Years	6 Years	7 Years	8 Years	10 Years	15 Years
2.50	36.0	40.6	44.2	47.2	51.7	58.9
3.00	36.9	41.5	45.2	48.2	52.8	59.9
3.50	37.6	42.4	46.1	49.1	53.7	60.9
4.00	38.3	43.1	46.9	50.0	54.7	61.9
4.50	38.7	43.8	47.7	50.8	55.6	62.9

Examples for Zone 3 -1500 watts (Burnup GWD/MTU)

Enrichment (wt. % U-235)	5 Years	6 Years	7 Years	8 Years
3.50	47.9	53.5	57.8	61.4
4.00	48.9	54.6	59.0	62.5
4.25	49.4	55.1	59.5	63.1
4.50	49.9	55.6	60.1	63.7

Table 2-3
Bounding Spent Fuel Assembly Thermal and Radiological Characteristics

Parameter	17x17 MkBW
Enrichment (%wt U-235)	4.0$^{(3)}$
Burnup (MWd/MTU)	60,000
Minimum Cooling Time (years)	7
Decay Heat (kW/assy)	1.5$^{(1)}$ or less
Gamma Source (γ/sec/assy)$^{(2)}$	6.33E+15
Neutron Source (n/sec/assy)$^{(2)}$	1.10E+09

(1) Decay heat for fuel assembly excluding control components. Decay heat for control components (0.009 kW per assembly maximum) is specified in Table 2.-3
(2) Gamma/neutron source spectrum by energy group is presented in Chapter 5.
(3) *For criticality max enrichment is 5.0%wt.*

Table 2-4
Non-Fuel Assembly Hardware Thermal and Radiological Characteristics

Parameter	BPRA (Bounding)
Gamma Source$^{(1)}$ (γ/sec/assy)	2.30E+14
Decay heat (Watts/assy)$^{(2)}$	9

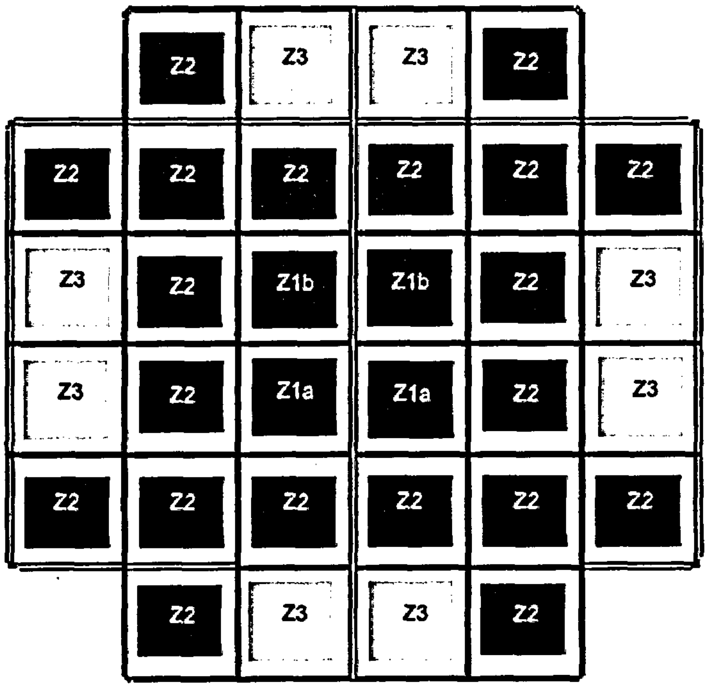
(1). Four days cooled, 30 GWD/MTU source.
(2). Five years cooled, 30 GWD/MTU source.
(3) Gamma source by energy group is presented in Chapter 5

Table 2-5
NUHOMS® HD System Major Components and Safety Classification

Component		10CFR 72 Classification(1)
Dry Shielded Canister (32PTH DSC)		
	Fuel compartment	Important to Safety
	Poison Plate	Important to Safety
	Basket Plate	Important to Safety
	Basket Rail	Important to Safety
	Weld Studs	Important to Safety
	Shell	Important to Safety
	Outer Top Cover Plate	Important to Safety
	Top Shield Plug/Inner Top Cover	Important to Safety
	Shell Bottom	Important to Safety
	Bottom Shield Plug (alternate design)	Not Important to Safety
	DSC Support Ring	Important to Safety
	Siphon and Vent Port Cover Plates	Important to Safety
	Grapple Ring and Grapple Support	Important to Safety
	Weld Filler Metal	Important to Safety
Horizontal Storage Module (HSM-H)		
	Reinforced Concrete	Important to Safety
	32PTH DSC Support Structure	Important to Safety
	Thermal Instrumentation (if used)	Not Important to Safety
ISFSI Basemat and Approach Slabs		Not Important to Safety
Transfer Equipment		
	On-site OS187H Transfer Cask	Important to Safety
	Cask Lifting Yoke	Safety Related(2)
	Transfer Trailer/Skid	Not Important to Safety
	Ram Assembly	Not Important to Safety
	Dry Film Lubricant	Not Important to Safety
Auxiliary Equipment		
	Vacuum Drying System	Not Important to Safety
	Automatic Welding System	Not Important to Safety
	Transfer Cask/DSC Annulus Seal	Not Important to Safety

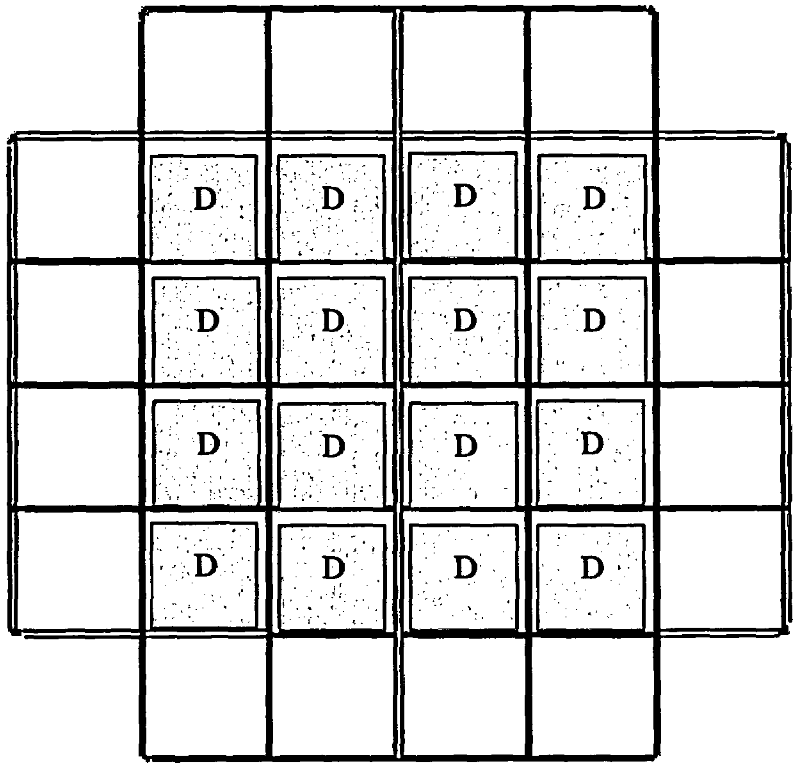
(1) Structures, systems and components "important to safety" are defined in 10CFR 72.3 as those features of the ISFSI whose function is (1) to maintain the conditions required to store spent fuel safely, (2) to prevent damage to the spent fuel container during handling and storage, or (3) to provide reasonable assurance that spent fuel can be received, handled, packaged, stored, and retrieved without undue risk to the health and safety of the public.

(2) Yoke and rigid or sling lifting members are classified as "Safety Related" in accordance with 10CFR 50.



- Q_{zi} is the maximum decay heat per assembly in zone i
- Total Decay Heat ≤ 34.8 kW
- 4 fuel assemblies in zone 1 with
  - total decay heat ≤ 3.2 kW
  - $Q_{z1a} \leq 1.05$ kW in the lower compartments
  - $Q_{z1b} \leq 0.8$ kW in the upper compartments
- 20 fuel assemblies in zone 2 with $Q_{z2} \leq 1.1$ kW
- 8 fuel assemblies in zone 3 with $Q_{z3} \leq 1.5$ kW
- $Q_{z1} \leq Q_{z2} \leq Q_{z3}$

Figure 2-1
Heat Load Zones



Up to 16 damaged assemblies with the remaining intact assemblies.

Figure 2-2
Location of Damaged Assemblies

CHAPTER 3
STRUCTURAL EVALUATION

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3. STRUCTURAL EVALUATION

3.1 Structural Design

This chapter, including its appendices, presents the structural evaluation of the NUHOMS® HD System.

The NUHOMS® HD system consists of the 32PTH DSC basket and shell assemblies, the HSM-H, and the OS187H Transfer Cask. The 32PTH DSC is a new dual purpose canister that is designed to accommodate up to 32 intact PWR fuel assemblies (or up to 16 damaged assemblies, with the remaining intact) with total heat load of up to 34.8 kw. The HSM-H is an enhanced version of the NUHOMS® Standardized HSM and incorporates design features to enable storage of the higher heat load 32PTH DSC. The OS187H is the modified version of OS197 transfer cask with a redesigned shielding panel to improve the thermal performance, shortened the cavity length and increased inside diameter to accommodate the larger diameter of 32PTH DSC.

The overall design bases for the NUHOMS® HD system are described in Chapters 1 and 2. This Chapter discusses the structural design criteria and associated design bases applicable to the 32PTH DSC, HSM-H, and OS187H transfer cask. This Chapter also describes the ability of these components to perform their design function during normal and off-normal operating conditions, as well as under postulated accident conditions and extreme natural phenomena events.

3.1.1 Discussion

The NUHOMS® HD system consists of the 32PTH DSC, a high-integrity stainless steel dry shielded canister that provides for the dry storage of spent fuel assemblies in an inert atmosphere; the HSM-H, a massive reinforced concrete storage module that houses and provides environmental protection and shielding to the 32PTH DSC; and the OS187H transfer cask, a stainless steel cask, with lead shielding, that handles and protects the 32PTH DSC during transfer to and from the HSM-H.

Multiple HSM-Hs are grouped together to form arrays in single or double rows to provide storage capacity consistent with available site space and reactor SFA discharge rates. The HSM-H is placed next to, and in contact with, adjacent module(s) to form a continuous single or double row arrays.

For purposes of the structural analyses and agreement with the criteria set forth in Regulatory Guide 3.61 [1] and NUREG 1536 [2], a single NUHOMS® HD System 32PTH DSC plus an HSM-H form the "cask" cited in [1] and [2].

The codes and standards used for the design, fabrication, and construction of the NUHOMS® HD system components, equipment, and structures are summarized in Table 3-1 and are identified throughout the SAR. Exceptions to the ASME Code [4] are provided in Section 3.10.

3.1.1.1 General Description of the 32PTH DSC

The principal characteristics of the 32PTH DSC are described in Chapter 1, Section 1.2.1. The drawings in Section 1.5 provide the principal dimensions and design parameters of the 32PTH DSC.

For purposes of the structural analysis, the 32PTH DSC is divided into the 32PTH DSC shell assembly and the internal basket assembly. The 32PTH DSC pressure boundary (shown in Figure 1-5) consists of the cylindrical shell, the shell bottom assembly, and top shield plug/inner cover plate, and the associated welds. The outer top cover plate provides a redundant closure which satisfies 10CFR 72.24 [3]. Both the inner and outer top cover plate welds are in accordance with ASME Section III Code Case N-595-3 [4].

A. DSC Canister (Shell) Assembly Description

The canister shell assembly and details are shown on drawings 10494-72-2 through 10494-72-7 in Chapter 1, Section 1.5. The shell assembly is a high integrity stainless steel (SA-240 Type 304) welded pressure vessel that provides confinement of radioactive materials, encapsulates the fuel in an inert atmosphere (the canister is backfilled with Helium before being seal welded closed), and provides biological shielding (in axial direction).

The remaining 32PTH DSC shell assembly components include the solid stainless steel top shield plug, the grapple ring assembly, support ring, and the lifting blocks. The outer top cover, top shield plug and shell bottom provide biological shielding during fuel loading operations and storage of a loaded 32PTH DSC. The grapple ring assembly is welded to the shell bottom or outer bottom cover plate for the purpose of inserting/extracting the 32PTH DSC to and from the Horizontal Storage Module (HSM-H). The support ring, welded to the cylindrical shell, supports the top shield plug. Four lifting blocks are welded to the inside of the shell bottom and are used in conjunction with a lifting fixture to lift the unloaded 32PTH DSC into the transfer cask prior to fuel loading operations.

All primary components of the 32PTH DSC are constructed from Type 304 stainless steel. The 32PTH DSC cylindrical shell and shell bottom assembly (which includes the shell bottom and the grapple ring assembly), and the internal basket assembly, are shop-fabricated (and assembled) components. The top shield plug and outer top cover plate is installed at the plant after the spent fuel assemblies have been loaded into the 32PTH DSC internal basket.

The 32PTH DSC shell assembly is designed, fabricated, examined and tested in accordance with the requirements of Subsection NB of the ASME Code including Code Case N-595-3 [4]. The circumferential and longitudinal shell plate weld seams are multi-layer full penetration butt welds. The butt weld joints are fully radiographed and inspected according to the requirements of NB-5000 of the ASME Boiler and Pressure Vessel Code. The full penetration inner bottom cover plate to shell weld is inspected to the same Code standards.

The 32PTH DSC top closure is compliant with Code Case N-595-3 and NRC's ISG-4 [5]. The outer top cover plate and top shield plug are sealed by separate, redundant closure welds. The top shield plug is welded to the 32PTH DSC shell to form the inner pressure boundary at the top end of the 32PTH DSC, as shown in Chapter 1, Figure 1-5. The redundant confinement boundary is provided by the outer top cover plate. All closure welds are multiple-layer welds. This effectively eliminates any pinhole leaks which may occur in a single-pass weld, since the chance of pinholes being in alignment on successive weld passes is negligibly small. Also, both welds are examined by multi-level liquid penetrant to effectively eliminate through wall leaks.

The top shield plug of the 32PTH DSC incorporates a vent and a siphon port, with two small-diameter tubing penetrations into the 32PTH DSC cavity for draining and filling operations. One penetration, the vent port, is terminated at the bottom of the shield plug assembly. The other port is attached to a siphon tube, which continues to the bottom of the 32PTH DSC cavity. The vent and siphon ports terminate in normally closed quick-connect fittings. Both ports are used to remove water from the 32PTH DSC during the drying and sealing operations.

During fabrication, leak tests of the 32PTH DSC shell assembly are performed in accordance with ANSI N14.5-1997[6] to demonstrate that the shell is leaktight (1×10^{-7} std. cm^3/sec).

The stringent design and fabrication requirements described above ensure that the pressure retaining confinement function is maintained for the design life of the 32PTH DSC. Pressure monitoring instrumentation is not used since penetration of the pressure boundary would be required. The penetration itself would then become a potential leakage path and, by its presence, compromise the leaktightness of the 32PTH DSC design.

During draining, backfilling, and leak testing, a "Strongback Device" may be installed to minimize deformation of the inner top cover plate during blowdown. The strongback is bolted to the top flange of the transfer cask and provides support to the inner cover plate during those operations that may involve significant pressurization of the 32PTH DSC cavity.

Transfer of the 32PTH DSC from the transfer cask into the HSM-H is performed using a hydraulic ram that applies a load to the bottom cover plate assembly, at the center of the DSC.

Frictional loads during 32PTH DSC transfer are reduced by application of a dry film lubricant to the hardened nitronic surface on the support rails of the HSM-H and the transfer cask. The lubricant chosen for this application is a tightly adhering inorganic lubricant with an inorganic binder. The dry film lubricant provides a thin, clean, dry, layer of lubricating solids that is intended to reduce wear, and prevent galling in metals. It is applied as a thin sprayed coating, similar to paint, using a carefully controlled process. The lubricant is not affected by water and is designed to be highly resistant to aggressive chemicals. This product is designed for radiation service and has a low coefficient of sliding friction for stainless steel.

B. <u>Fuel Basket Assembly Description</u>

The details of the 32PTH DSC basket are shown in drawings 10494-72-8 through 10494-72-12 on Chapter 1, Section 1.5. The 32PTH DSC basket is a welded assembly of stainless steel fuel

compartment boxes, and designed to accommodate 32 PWR fuel assemblies. The sections of the stainless steel fuel compartments are fusion welded to Type 304 stainless steel structural plates, sandwiched between the box sections. The fusion welds are spaced intermittently along the box sections. Neutron poison plates, composed of a boron-aluminum alloy (or a boron carbide aluminum metal matrix composite), are sandwiched between the sections of the stainless steel walls of the adjacent box and the adjacent stainless steel plates. The Type 304 stainless steel members are the primary structural components. The neutron poison plates provide criticality control and a heat conduction path from the fuel assemblies to the canister shell. The bottom rows of plates which are 304 SST (no poison) are also sandwiched between the fuel compartment box sections, and provide structural support to the basket.

Stainless steel rails are oriented parallel to the axis of the canister and attached to the periphery of the basket to establish and maintain basket orientation and to support the basket.

The nominal open dimension of each fuel compartment cell is 8.70 inches × 8.70 inches, which provides clearance around the fuel assemblies. The overall length of the fuel basket is 162.00 inches, which is less than the canister cavity length of the canister (164.50 inches minimum) to allow for thermal expansion, tolerances, and access to the top of the fuel assemblies.

The basket structure is open at each end. Therefore, longitudinal fuel assembly loads are applied directly on the DSC body and not on the fuel basket structure. The fuel assemblies are laterally supported by the stainless steel fuel compartments and structural plates, and the fuel basket is laterally supported by the rails and the canister shell.

The circumferential orientation of the basket, relative to the canister shell, is maintained by the four lifting blocks attached to the bottom closure assembly of the canister. The four canister lifting blocks mate with the hollow portions of the basket outer support rails, without interfering with the spent fuel assemblies. During normal transfer conditions, the DSC rests on four transfer support rails, attached to the inside surface of the NUHOMS®-OS187H Transfer Cask.

3.1.1.2 General Description of the HSM-H

The details of the HSM-H module are shown in drawings 10494-72-100 through 10494-72-109 on Chapter 1, Section 1.5. The HSM-H is a free standing reinforced concrete structure designed to provide environmental protection and radiological shielding for the 32PTH DSC. Each HSM-H provides a self contained modular structure for the storage of a 32PTH DSC containing up to 32 PWR fuel assemblies. The HSM-H provides heat rejection from the spent fuel decay heat by a combination of radiation, conduction and convection.

The HSM-H is a reinforced concrete structure consisting of two separate units: a base storage unit, where the 32PTH canister is stored, and a top shield block that serves to provide environmental protection and radiation shielding. The top shield block is attached to the base unit by vertical ties. Three-foot thick shield walls are installed behind each HSM-H (single row array only) and at the ends of each row to provide additional shielding.

The HSM-H module design for 32PTH canister is identical to the HSM-H design for 24PTH canister except the following modifications:

1. The module for the 32PTH canister is designed such that the center line of the loaded 32PTH canister is approximately four inches higher compared to that of the 24PTH canister.

2. The diameter of the door openings in the front and rear of the front wall are approximately four inch and two inch larger for the 32PTH canister compared to those of the 24PTH canister.

3. The transfer cask docking surface in the module for the 32PTH canister transfer cask is approximately half inch wider compared to the cask docking surface for the 24PTH canister transfer cask.

4. The diameters of the front inner circular steel plate and rear circular concrete block of the shielded door for the 32PTH canister are approximately four inch and two inch larger compared to those of the 24PTH canisters.

5. For the 32PTH design the spacers at the canister stop plate of the module will be provided similar to the 24PTH short cavity design.

The drawings in Chapter 1, Section 1.5 provide the principal dimensions and design parameters of the HSM-H. The dimension differences between the HSM-H to be used for storing the 32PTH canister and 24PTH canister are listed in the following tables.

TN drawing No. 10494-72-104

HSM-H		
Dimension	System Type	
	For 32PTH Canister	For 24PTH Canister
A	8' – 10"	8' – 6"
B	ø 5' – 11 5/8"	ø 5' – 9"
C	ø 7' – 5"	ø 7' – 1 1/2"

TN drawing No. 10494-72-107

HSM-H		
Dimension	System Type	
	For 32PTH Canister	For 24PTH Canister
A	34.88"	33.60"

TN drawing No. 10494-72-108

HSM-H		
Dimension	System Type	
	For 32PTH Canister	For 24PTH Canister
A	8' – 1 1/2"	7' – 10"
B	ø 7' – 3"	ø 6' – 11 1/2"
C	ø 5' – 8 5/8"	ø 5' – 6"
D	ø 7' – 7 1/4"	ø 7' – 3 3/4"
E	ø 1' – 10 1/2"	ø 1' – 10 1/2"

The design of the HSM-H for 32PTH DSC is the same as the HSM-H which is under NRC review as Amendment 8 to CoC 1004 for 24PTH DSC. Analyses performed for HSM-H with 24PTH DSC used bounding values to envelop both 24PTH DSC and 32PTH DSC.

3.1.1.3 General Description of the OS187H On-Site Transfer Cask

The NUHOMS®-OS187H On-Site Transfer Cask consists of a structural shell, gamma shielding material, and solid and liquid (water) neutron shield. The OS187H is the modified version of OS197 transfer cask with a redesigned shielding panel to improve the thermal performance, shortened the cavity length and increased inside diameter to accommodate the larger diameter of 32PTH DSC. The cavity between the DSC and the transfer cask contains an inert gas during transfer operations. Sets of upper and lower trunnions, welded to the structural shell of the cask that provide support, lifting, and rotation capability for the OS187H transfer cask.

The overall dimensions of the OS187H transfer cask are 197.07 inches long and 92.20 inches in diameter. The transfer cask structural shell is 82.70 inches in diameter. The transfer cask cavity is 186.60 inches long and 70.50 inches in diameter. Detailed design drawings for the OS187H Transfer Cask are provided in drawings 10494-72-15 through 10494-72-21 on Chapter 1, Section 1.5. The materials used to fabricate the transfer cask are shown in the Parts List on Drawing 10494-72-15. Where more than one material has been specified for a component, the most limiting properties are used in the analyses in the subsequent chapters of this Safety Analysis Report.

The gross weight of the loaded transfer cask is 114.3 tons including a maximum payload of 54.4 tons. Section 3.2.2 summarizes the weights of the NUHOMS®-OS187H packaging components. Trunnions, welded to the structural shell of the transfer cask, are provided for lifting and handling operations, including rotation of the packaging between the horizontal and vertical orientations. The OS187H cask transfers the DSC in the horizontal orientation, on a specially designed transfer skid, with the lid end facing the direction of travel.

The transfer cask is fabricated primarily of stainless steel. Non-stainless steel members include the cast lead shielding between the inner radial shell and the structural shell, the o-ring seals, the resin and water neutron shield material and the carbon steel closure bolts. The lead is poured into the annulus in a molten state using a carefully controlled procedure. The top cover is bolted to the top flange by 24-1 1/2 in. diameter high strength bolts and sealed with O-ring. A cover plate

is provided to seal the bottom hydraulic ram access penetration of the cask (by 12-1/2 in. high strength bolts with O-ring) during fuel loading and transferring the canister to the ISFSI. Drawing 10494-72-15 provides the part list for the NUHOMS®-OS187H transfer cask. Drawing 10494-72-16 shows the overall configuration of the NUHOMS®-OS187H transfer cask. Drawing 10494-72-17 shows the details of the transfer cask top cover. The remaining drawings (10494-72-18 through 10494-72-21) show the details of the remaining individual components that make up the transfer cask.

The following sections provide physical and functional descriptions of each major component of the transfer cask. Detail drawings showing dimensions of significance to the safety analyses, welding and NDE information, as well as a complete materials list are provided in Chapter 1, Section 1.5. Reference to these drawings is made in the following physical description sections, and in general, throughout this SAR.

A. Transfer Cask Body and Structural Components

The shell or cask body cylinder assembly is an open ended (at the top) cylindrical unit with an integral closed bottom end. This assembly consists of concentric inner shell and outer shell (both SA-240 Type 304), welded to massive closure flanges (SA-240 Gr. Type 304) at the top and bottom ends. The inner shell is 0.50 inches thick and has a 70.50 inch inside diameter. The outer shell is the primary structural shell and is 1.5 inches to 2.0 inches thick, and has an 82.70 inch outside diameter. The annulus between the shells is filled with lead shielding. The lead gamma shield is 3.60 inch thick and is poured into the annulus in a molten state using a carefully controlled procedure.

The transfer cask bottom end assembly consists of a 2.00 inch bottom end plate and a 0.75 inch bottom neutron shield plate, that sandwich a 2.25 inch think resin neutron shield. The RAM access penetration at the center of the bottom end assembly is used during insertion/removal operations to and from the HSM-H. The RAM access penetration is four inches thick in the radial direction and 4.25 inches thick in the axial direction. A cover plate is provided to seal the bottom hydraulic ram access penetration of the cask (by 12-1/2 in. high strength bolts with O-ring) during fuel loading and transferring the canister to the ISFSI.

The transfer cask top cover consists of a 3 inch thick structural plate constructed from SA-240, Type XM-19, and a top radial neutron shield constructed from *Vyal B* resin encased in a 0.25 inch thick SA-240 Type 304 stainless steel shell. The top cover is fastened to the top flange of the transfer cask body with 24-1.5 inch diameter SA-540 Grade B24 Class 1 high strength steel bolts. The top closure is designed to maintain confinement of the 32PTH DSC inside the transfer cask during all normal, off normal and hypothetical accident conditions.

The transfer cask body provides additional radiation shielding and structural support for the 32PTH DSC. It also maintains an inert atmosphere (helium) in the cask cavity. Helium assists in heat removal during transfer operations and provides a non-reactive environment. To preclude air in-leakage, the cask cavity is pressurized with helium to above atmospheric pressure.

The NUHOMS®-OS187H transfer cask is designed, fabricated, examined and tested in accordance with the requirements of Subsection NC [7] of the ASME Code to the maximum practical extent. Exceptions to the ASME Code are discussed in Section 3.10.

B. Gamma and Radial Neutron Shielding

The lead and steel shells of the transfer cask provide shielding between the DSC and the exterior surface of the package for the attenuation of gamma radiation.

Axial neutron shielding is primarily provided by a borated polyester resin compound. The resin compound is cast into stainless steel cavities on the outside surface of the top closure and bottom assembly.

The *Vyal B* resin material is an unsaturated polyester cross-linked with styrene, with about 50% weight mineral and fiberglass reinforcement. The components are polyester resin, styrene monomer, alpha methyl styrene, aluminum oxide, zinc borate, and chopped fiberglass which produce the elemental resin composition shown below [11].

Element	% Weight
H	4.77
B	0.895
C	24.09
O	47.00
Al	21.445
Zn	1.80

Radial neutron shielding is primarily provided by liquid water enclosed in a radial outer stainless steel shield shell. The shield shell around the neutron shield consists of a cylindrical shell section, with closure plates at each end. The closure plates are welded to the outer surface of the structural shell of the cask body. The outer shield shell has no structural function other than to provide an enclosure for the neutron shield water. The shell is made of SA-240 Type 304 stainless steel.

C. Tiedown and Lifting Devices

There are four trunnions welded to the exterior of the structural shell of the transfer cask. There are two front trunnions located on opposite sides of the cask near the top closure, and two rear trunnions located similarly, near the bottom of the cask. The two top trunnions are used to first lift the cask, containing a canister and an empty basket, into a fuel pool for loading of the spent fuel. After the spent fuel has been loaded into the basket, the cask is lifted to a decontamination area. After draining and drying of the pool water, welding of the canister cover, and bolting of the cask cover, the cask is placed in a trailer for transfer to ISFSI. The cask is vertically lifted onto the trailer and is initially supported by the bottom trunnions which are mated to transfer trailer. Then the cask is allowed to pivot about the bottom trunnions, into a horizontal position until the top trunnions rest on their supports in the trailer. The trunnions are secured to the skid's trunnion tower.

The top trunnions are constructed from SA-182 Type FXM-19 and the bottom trunnions are constructed from SA-182 Type 304. Both materials are stainless steel forgings. The top trunnions are designed fabricated and tested in accordance with ANSI N14.6 [8] as single failure proof lifting devices. Consequently they are designed with a factor of safety of six against the material yield strength and a factor of ten against the material ultimate strength.

D. Operational Features

The NUHOMS®-OS187H transfer cask is not considered to be operationally complex and is designed to be compatible with spent fuel pool loading/unloading methods. All operational features are readily apparent from inspection of the General Arrangement Drawings provided in Chapter 1, Section 1.5. The sequential steps to be followed for cask loading, testing, and unloading operations are provided in Chapter 8.

3.1.2 Design Criteria

This section specifies the design requirements of the NUHOMS® HD system. The system consists of the Transportable Dry Shielded Canister (DSC), the Horizontal Storage Module, HSM-H and the OS-187H onsite transfer cask. The system is designed for high burnup fuel, up to 60 GWD/MTU, with a maximum 5% initial enrichment. The design will be based on the NUHOMS® design concept of horizontal storage, and is intended for use with a compatible transport cask.

General design requirements include structural, thermal, nuclear criticality safety, confinement/containment, and radiological protection criteria.

The overall storage system consists of three major components:

- 32PTH Dry Storage Container
- 32PTH Horizontal Storage Module
- OS187H Transfer Cask

The reinforced concrete 32PTH HSM-H, including the 32PTH-DSC support structure, the 32PTH-DSC, and the structural components of the OS187H transfer cask are important to safety of NUHOMS® HD System components. Consequently, they are designed and analyzed to perform their intended functions under the extreme environmental and natural phenomena specified in 10CFR 72.122 [3] and ANSI 57.9 [9]. These include tornado and wind, seismic, and flood design criteria.

This section addresses component specific design criteria, loads, and load combinations for the structural analyses of the 32PTH DSC, 32PTH HSM-H and the OS187H Transfer Cask.

3.1.2.1 32PTH DSC Structural Design Criteria

3.1.2.1.1 32PTH DSC Stress Criteria

A. 32PTH DSC Shell Stress Limits

The stress limits for the DSC shell are taken from the ASME Boiler and Pressure Vessel Code, Section III, Subsection NB, Article NB-3200 for Level A through D Service Limits [12]. In accordance with NB-3225, Appendix F [13] is used for accident condition loads (Level D).

The stress due to each load shall be identified as to the type of stress induced, e.g. membrane, bending, etc., and the classification of stress, e.g. primary, secondary, etc.

Stress limits for Level A through D service loading conditions are summarized in Table 3-2. Local yielding is permitted at the point of contact where Level D load is applied. If elastic stress limits cannot be met, the plastic system analysis approach and acceptance criteria of Appendix F of Section III shall be used.

The allowable stress intensity value, S_m, as defined by the Code, is to be based on the calculated (or a bounding) temperature for each service load condition.

B. 32PTH DSC Basket Stress Limits

The basket fuel compartment tube wall thickness is established to meet heat transfer, nuclear criticality, and structural requirements. The basket structure must provide sufficient rigidity to maintain a sub-critical configuration under the applied loads.

The stress analyses of the basket do not take credit for the poison plates except for through thickness compression. However, the weight of the poison plate is considered when determining stresses in the stainless steel.

The basis for the stainless steel fuel compartment section stress allowables is the ASME Code, Section III, Subsection NG [14]. Stress limits for Level A through D service loading conditions are summarized in Table 3-3.

Alternatively, and in accordance with NG-3222 and Note 9 to Figure NG-3221-1, the Limit Analysis provisions of NG-3228 may be used for Level A Service Limits.

The basket shall be evaluated under Level D Service loadings in accordance with the Level D Service limits for components in Appendix F [13] of Section III of the Code. The hypothetical impact accidents are evaluated as short duration Level D conditions.

For fusion welds between the stainless steel plates and the stainless steel fuel compartments shall be qualified by testing. The required minimum tested capacity of the weld connection shall be based on a margin of safety (test to design) of 2.0, corrected for temperature difference between testing and basket operating conditions and the maximum weld load at any weld location in the

basket. This margin of safety, 2, is larger than the ASME Code-implied margin of safety for level D loads. The minimum capacity shall be determined by shear test (pull test) of individual specimen made from production material. In addition to the ASME Code requirements for weld qualification, as part of the weld qualification procedure, in order to verify proper machine setting and operation, a shear test (pull test) of test coupon from each welding machine will be performed prior to the start of each working shift.

3.1.2.1.2 32PTH DSC Stability Criteria

The fuel compartment tube walls and canister shell, when subjected to compressive loadings, are also evaluated against ASME Code rules for component supports to ensure stability. The acceptance criteria (allowable buckling loads) are taken from ASME Code, Section III, Appendix F, paragraph F-1341.3, Collapse Load. The allowable buckling load is 90% of the calculated limit analysis collapse load using a yield stress which is the lesser of 2.3 S_m and 0.7 S_u or 100% of the calculated plastic analysis collapse load or 100% of the test collapse load. Alternate acceptable criteria as provided in Appendix F of the ASME Code or other justifiable criteria are acceptable.

3.1.2.2 HSM-H Structural Design Criteria

The HSM-H concrete and steel components shall be designed to the requirements of ACI 349 [15] and the AISC Manual of Steel Construction [16], respectively, meeting the load combinations in accordance with the requirements of ANSI 57.9 [9]. A detail design criteria is described in Appendix 3.9.9.

3.1.2.3 OS187H Transfer Cask Structural Design Criteria

The OS187H TC is designed to meet the criteria of ASME Code Subsection NC [7] for Class 2 components. Service Level A allowables are used to for all normal operating and off-normal loadings. Service Level D allowables are used for load combinations that include postulated accidents loadings. The OS187H Transfer Cask allowable stresses for normal and accident conditions are summarized in Table 3-4.

3.2 Weights

The nominal DSC, HSM-H and OS187H Transfer Cask geometry is used to compute the weights of the NUHOMS® HD system components.

The following densities are used to compute the component weights.

NUHOMS® HD Component Material Densities

Material	Density (lb./in^3.)	Reference
Stainless Steel	0.29	10
Aluminum	0.098	10
Water	0.0361	10
Lead	0.41	10
Vyal B Resin (neutron shield)	0.065	11

3.2.1 32PTH DSC Weight

The total weight of the loaded 32PTH DSC is 108.76 kips (54.38 tons). The weights of the major individual subassemblies are listed in following table.

32PTH DSC Summary of Nominal Component Weights

Component	Nominal Weight (lbs. x 1000)
Canister Shell	5.86
Outer Top Cover Plate	2.14
Top Shield Plug and Support Ring	10.71
Bottom Shield Plug	9.42
Grapple Ring	0.06
Total Canister Assembly	**28.19**
Fuel Compartments (32)	10.02
Aluminum Plates	3.73
Poison Plates	0.55
Stainless Steel Plates	1.94
Small Support Rails (4)	3.24
½ Large Support Rails (8)	10.38
Total Fuel Basket	**29.85**
Total Empty DSC (Basket & Canister)	**58.04**
Fuel Assembly Weight (32) @ 1585 lbs/assembly	50.72
Total Loaded DSC Weight	**108.76**

3.2.2 OS187H Transfer Cask Weight

The total weight of the loaded NUHOMS®-OS187H Transfer Cask is 228.69 kips (114.3 tons). The weights of the major individual subassemblies are listed in following table.

OS187H Transfer Cask Summary of Nominal Component Weights

Component	Nominal Weight (lbs. x 1000)
Structural Shell	20.85
Inner Shell	5.86
Lead Gamma Shield	62.37
Top Flange	3.18
Bottom Flange	3.37
Top Cover Assembly	5.20
Bottom Assembly	3.46
Neutron Shield Panel	4.29
Radial Neutron Shield (water)	8.46
Upper Trunnion Pair	1.61
Lower Trunnion Pair	1.27
Total Empty Transfer Cask Weight	**119.92**
Total Transfer Cask with Empty DSC Weight	**177.97**
Total Transfer Cask with Loaded DSC Weight	**228.69**

Note: 250.0 kips is conservatively used for trunnion analysis.

3.2.3 HSM-H Weight

The following table summaries the weights of the loaded HSM-H single module.

Summary Weights of 32PTH HSM-H

Component	Nominal Weight (lbs. x 1000)
HSM-H Single Module Weight (empty)	306.1
Total Empty DSC (Basket & Canister)	58.04
Fuel Assembly Weight (32)	50.72
Total Loaded HSM-H Weight	**414.86**

3.3 Mechanical Properties of Materials

3.3.1 32PTH-DSC Material Properties

The principal material of construction for the 32PTH DSC is Type 304 stainless steel. The 32PTH DSC cylindrical shell, cover plates and shield plugs are constructed from SA-240 Type 304 stainless steel for plate material and SA-182 F304 for forging material. The 32PTH DSC basket assembly fuel compartments and structural plate assemblies are also constructed from SA-240 Type 304 stainless steel. Table 3-5 contains the ASME Code material properties for SA-240 Type 304 and SA-182 F304 stainless steel materials.

The neutron absorber plates are constructed from boron carbide/aluminum metal matrix composite material and the aluminum thermal conduction plates are constructed from SB-209 (Type 1100 Aluminum). No structural credit is taken for either the neutron absorber plates or the aluminum thermal conducting plates, except for through the thickness load transmission.

3.3.1.1 Radiation Effects on 32PTH DSC Materials

Gamma radiation has no significant effect on metals. The effect of fast neutron irradiation of metals is a function of the integrated fast neutron fluence, which is on the order of 1×10^{15} neutrons/cm^2 inside the 32PTH DSC after 50 years. Studies on fast neutron damage in stainless steel, and low alloy steels rarely evaluate damage below 10^{17} n/cm^2 because it is not significant [17]. Extrapolation of the data available down to the 10^{15} range confirms that there will be no measurable neutron damage to any of the 32PTH DSC metallic components.

3.3.1.2 DSC Weld Material

Welding processes, welders and welding materials used for the welding of the 32PTH DSC meet the requirements of the appropriate ASME Section III subsections and Section IX. Non-Code welds meet the provisions of Section IX of the ASME Code or AWS D1.1 [18] or D1.6 [19]. Weld metal material properties meet the requirements of Section II of the ASME Code or associated AWS requirements.

3.3.1.3 DSC Material Brittle Fracture

Brittle fracture is not a concern for the stainless steel components, which comprises all structural components of the DSC.

3.3.2 HSM-H Material Properties

The temperature dependent material properties for concrete and reinforcing steel are provided in Tables 3-6 and 3-7. The material properties of the ASTM A992 steel used for fabrication of the rail support structures are listed in 3-7A. The material properties for type 304 used for the heat shield support plate are provided in Table 3-5. The A-36 steel used for rail assembly extension plate is provided Table 3-7B.

3.3.2.1 Radiation Effects on HSM-H Concrete

The accumulated neutron flux over a 40 year service life of the HSM-H is estimated to be 1.5×10^{14} neutrons/cm^2. From the study by Hilsdorf, Kropp, and Koch [20], the compressive strength and modulus of elasticity of concrete is not affected by a neutron flux of this magnitude. The gamma energy flux deposited in the HSM-H concrete is 1.7×10^9 MeV/cm^2-sec. or 3.0×10^{-4} watt/cm^2. According to ANSI/ANS-6.4-1977 [21], the temperature rise in concrete due to this level of radiation is negligible. Thus, radiation effects on concrete strength are not evaluated further for the HSM-H design.

3.3.2.2 HSM-H Materials Durability

As shown in Table 3-5 through Table 3-7B, all materials meet the appropriate requirements of the ASME Code, ACI Code, and ASTM Standards. The durability of the steel components is well beyond the design life of the applicable components. The specifications controlling the mix of concrete, specified minimum concrete strength requirements, and fabrication control ensure durability of the materials for this application.

3.3.3 OS187H Transfer Cask Material Properties

The principal material of construction for the OS187H transfer cask is Type 304 stainless steel. The transfer cask structural, inner and outer neutron shield shells and the bottom closure assembly are constructed from SA-240 Type 304 stainless steel. The primary structural member of the top cover plate is constructed from SA-240 Type XM-19 stainless steel. Table 3-5 contains the ASME Code material properties for SA-240 Type 304 stainless steel material. ASME Code material properties for the top cover material (SA-240 Type XM-19) are given in Table 3-8.

The transfer cask top cover and ram access cover bolts are constructed from SA-540 Grade B24 Cass 1. ASME Code material properties for SA-540 Grade B24 Class 1 are given in Table 3-9.

Material properties for ASTM B-29 (Chemical Lead), which is used for the transfer cask radial gamma shield, are given in Table 3-10.

The outer radial neutron shield consists of a SA-240 Type 304 stainless steel shell that contains the neutron absorbing material (water). The top and bottom axial neutron absorber material is *Vyal B* Resin, described in Section 3.1.1.3 B. No structural credit is taken for the neutron absorber material, except for through the thickness load transmission. Material Properties for the *Vyal B* Resin are given in Table 3-11.

3.3.3.1 Radiation Effects on the Transfer Cask Materials

Gamma radiation has no significant effect on metals. The effect of fast neutron irradiation of metals is a function of the integrated fast neutron fluence, which is on the order of 1×10^{15} neutrons/cm^2 inside the cask after 50 years. Studies on fast neutron damage in stainless steel, and low alloy steels rarely evaluate damage below 10^{17} n/cm^2 because it is not significant [17]. Extrapolation of the data available down to the 10^{15} range confirms that there will be no measurable neutron damage to any of the cask metallic components.

3.3.3.2 Transfer Cask Weld Material

Welding processes, welders and welding materials used for the welding of the 32PTH DSC meet the requirements of the appropriate ASME Section III subsections and Section IX. Non-Code welds meet the provisions of Section IX of the ASME Code or AWS D1.1 [18] or D1.6 [19]. Weld metal material properties meet the requirements of Section II of the ASME Code or associated AWS requirements.

3.3.3.3 Transfer Cask Brittle Fracture

Brittle fracture is not a concern for the stainless steel components, which comprises all structural components of the cask.

3.4 General Standards for 32PTH DSC, HSM-H, and OS187H TC

3.4.1 Chemical and Galvanic Reactions

The materials of the DSC shell and basket assemblies, the HSM-H, and the transfer cask components have been reviewed to determine whether chemical, galvanic or other reactions among the materials, contents and environment might occur during any phase of loading, unloading, handling or storage.

The canister, transfer cask, and HSM-H are exposed to the following environments:

- During loading and unloading, the canister is inside of the transfer cask. Thus, the exterior of the canister will not be exposed to pool water. The annulus between the transfer cask and canister is filled with clean water and sealed.
- The interior surfaces of the canister, top shield plug, and the basket will be exposed to (borated) pool water. The transfer cask and canister are kept in the spent fuel pool for only about 6 hours to load or unload fuel, and 2 hours to lift the loaded transfer cask/canister out of the spent fuel pool. An additional 12 to 24 hours is typically needed to decontaminate the cask, weld the DSC cover, and drain the water.
- The canister is vacuum dried before storage. It is then backfilled with helium, thus providing a non-corrosive environment. During storage, the interior of the canister is exposed only to the helium environment. The dry helium environment does not support chemical or galvanic reactions.
- During storage, the exterior of the canister is protected by the concrete HSM-H. The HSM-H is vented, so the exterior of the canister is exposed to the atmosphere. The exterior is exposed to the weather.

Materials used for the DSC, transfer cask, and HSM-H are shown in the parts lists of the drawings provided in Chapter 1, Section 1.5.

Within the canister, there is a basket with support rails made from SA-240 Type 304 stainless steel with aluminum inserts. The basket structure consists of an assembly of stainless steel tubes (fuel compartments) separated by aluminum and neutron absorber plates. The basket fuel compartments are constructed from Type 304 stainless steel. The neutron absorber is borated aluminum alloy or composite sandwiched. The neutron poison plates are not welded or bolted to the fuel compartments, but are held in place by the fuel compartments and the stainless steel structural plates. The aluminum thermal conduction plates are constructed from Type 1100 aluminum.

The only potential galvanic couples are the low alloy steel transfer cask bolts and hoist rings with stainless steel, and stainless steel with aluminum in the DSC. The lid, test, drain cover, and ram cover bolts will be exposed to the weather or pool water for only a short period during DSC transfer. Galvanic corrosion during transfer will be negligible and will have no adverse affect on

design functions. Furthermore, the bolts are subject to periodic inspection per Section 9.2.1. The couple of stainless steel and aluminum is discussed in Section 3.4.1.2.

Typical water chemistry in a PWR spent fuel pool is as follows:

pH (77 °F)	4.5 – 9.0
Chloride, max	0.15 ppm
Fluoride, max	0.1 ppm
Dissolved Air, max	Saturated
Lithium, max	2.5 ppm
Boric Acid	2,100 – 2,600 ppm
Pool Temperature Range	40 – 140 °F

3.4.1.1 Behavior of Austenitic Stainless Steel (DSC and Transfer Cask)

With the exception of the low alloy and aluminum parts noted above, all exposed surfaces of the transfer cask and the canister are stainless steel Type 304, Nitronic 60, or XM-19. Stainless steel does not exhibit general corrosion when immersed in borated water.

The chloride ion concentration and exposure duration are too short to cause stress corrosion cracking in the stainless steel welds, which may have residual fabrication stresses. Although stress corrosion cracking can take place at very low chloride concentrations and temperatures such as those in spent fuel pools (less than 10 ppb and 160 °F, respectively), the effect of low chloride concentration and low temperature is to greatly increase the initiation time, that is, the period during which the corrodent is breaking down the passive oxide film on the stainless steel surface. Below 60 °C (140 °F), stress corrosion cracking of austenitic stainless steel does not occur at all. At 100 °C (212 °F), chloride concentration on the order of 15% is required to initiate stress corrosion cracking [24]. At 288 °C (550 °F), with tensile stress at 100% of yield in PWR water containing 100 ppm O_2, time to crack is about 40 days in sensitized 304 stainless steel [24]. Thus, the combination of low chlorides, low temperature and short time of exposure to the corrosive environment eliminates the possibility of stress corrosion cracking in the fuel compartment welds.

3.4.1.2 Behavior of Aluminum (DSC Basket)

Aluminum is used for many applications in spent fuel pools. In order to understand the corrosion resistance of aluminum within the normal operating conditions of spent fuel storage pools, a discussion of each of the types of corrosion is addressed separately. None of these corrosion mechanisms are expected to occur in the short time period that the cask is submerged in the spent fuel pool.

General Corrosion

General corrosion is a uniform attack of the metal over the entire surfaces exposed to the corrosive media. The severity of general corrosion of aluminum depends upon the chemical nature and temperature of the electrolyte and can range from superficial etching and staining to

dissolution of the metal. Figure 3-1 shows a potential -pH diagram for aluminum in high purity water at 77 °F[25]. The potential for aluminum coupled with stainless steel and the limits of pH for BWR pools are shown in the diagram to be well within the passivation domain. The passivated surface of aluminum (hydrated oxide of aluminum) affords protection against corrosion in the domain shown because the coating is insoluble, non-porous and adherent to the surface of the aluminum. The protective surface formed on the aluminum is known to be stable up to 275 °F and in a pH range of 4.5 to 8.5 [25].

Galvanic Corrosion

Galvanic corrosion is a type of corrosion which could cause degradation of dissimilar metals exposed to a corrosive environment for a long period of time.

Galvanic corrosion is associated with the current of a galvanic cell consisting of two dissimilar conductors in an electrolyte. The two dissimilar conductors of interest in this discussion are aluminum and stainless steel in deionized water. There is less galvanic current flow between the aluminum-stainless steel couple than the potential difference on stainless steel which is known as polarization. It is because of this polarization characteristic that stainless steel is compatible with aluminum in all but severe marine, or high chloride, environmental conditions [26].

Pitting Corrosion

Pitting corrosion is the forming of small sharp cavities in a metal surface. The first step in the development of corrosion pits is a local destruction of the protective oxide film. Pitting will not occur on commercially pure aluminum when the water is kept sufficiently pure, even when the aluminum is in electrical contact with stainless steel. Pitting and other forms of localized corrosion occur under conditions like those that cause stress corrosion, and are subject to an induction time which is similarly affected by temperature and the concentration of oxygen and chlorides. As with stress corrosion, at the low temperatures and low chloride concentrations of a spent fuel pool, the induction time for initiation of localized corrosion will be greater than the time that the cask internal components are exposed to the aqueous environment.

Crevice Corrosion

Crevice corrosion is the corrosion of a metal that is caused by the concentration of dissolved salts, metal ions, oxygen or other gases in crevices or pockets remote from the principal fluid stream, with a resultant build-up of differential galvanic cells that ultimately cause pitting. Crevice corrosion could occur at the contact surfaces between the aluminum plates, neutron absorber, and fuel compartment tubes.
Due to the short time in the spent fuel pool, this type of corrosion is not expected to be significant.

Intergranular Corrosion

Intergranular corrosion is corrosion occurring preferentially at grain boundaries or closely adjacent regions without appreciable attack of the grains or crystals of the metal itself.

Intergranular corrosion does not occur with commercially pure aluminum and other common work hardened aluminum alloys.

Stress Corrosion

Stress corrosion is failure of the metal by cracking under the combined action of corrosion and high stresses approaching the yield stress of the metal. During normal operations there are negligible loads, mostly compressive, imposed on the aluminum parts.

3.4.1.3 Behavior of Aluminum Based Neutron Poison (DSC Basket)

The aluminum component of the borated aluminum is a ductile metal having a high resistance to corrosion. Its corrosion resistance is provided by the buildup of a protective oxide film on the metal surface when exposed to a corrosive environment. As stated above for aluminum, once a stable film develops, the corrosion process is arrested at the surface of the metal. The film remains stable over a pH range of 4.5 to 8.5.

Tests were performed by Eagle Picher [29] which concluded that borated aluminum exhibits a strong corrosion resistance at room temperature in deionized water. Satisfactory long-term usage in these environments is expected. At high temperature, the borated aluminum still exhibits high corrosion resistance in the pure water environment.

From tests on pure aluminum, it was found that borated aluminum was more resistant to uniform corrosion attack than pure aluminum.

An alternate neutron poison material is a boron carbide / aluminum composite, which is a matrix of full-density aluminum with a fine dispersion of boron carbide particles throughout. The corrosion behavior is similar to that of the base aluminum alloy.

The third neutron poison material is Boral™. The faces of the Boral sheet are 1100 aluminum, while the aluminum/boron carbide core is exposed at the edges of the sheet. There are no chemical, galvanic or other reactions that could reduce the areal density of boron in any of the poison plate materials for the 32PTH DSC. Boral™ is a proven neutron poison used extensively in spent fuel storage racks. The short term exposure of the material to borated water in the spent fuel storage canisters will have significantly less effect on the Boral™ than that experienced in spent fuel pools.

3.4.1.4 Behavior of HSM Materials

The exterior of the HSM-H is exposed to the weather. The interior is dry, and is subject to the thermal and radiological environment created by the decay of the spent fuel stored in the DSC. Chapter 4 demonstrates that the concrete remains below its operational temperature limit under all normal and off-normal conditions.

The sliding surface of the support rails for the DSC consists of Nitronic® 60 or equivalent stainless steel. Carbon steel embedments in the HSM-H concrete are coated to protect them from

corrosion or they may be stainless steel. Other carbon steel components such as bolts, nuts, tie plates, etc., are also coated. Bird screens are stainless steel. The side of the heat shield facing the DSC is made of anodized aluminum. The side facing the HSM-H concrete is plan aluminum surface.

Because of the coatings and the dry environment, degradation of concrete or steel parts inside the HSM-H is unlikely. Exterior parts or surfaces are also visible and accessible, and if any degradation occurs from exposure to weather, it can be corrected.

3.4.1.5 Lubricants, Sealants, and Cleaning Agents

Lubricants may be used to coat the slide rails, the threads and shoulders of bolts, o-rings, and the contact areas of the trunnions. Lubricants are generally selected from the list of materials approved for contact with the pool water at the facility where wet loading occurs.

Sealants may be used at pipe threads, e.g., at quick connect fittings.

The transfer cask and DSC are cleaned during fabrication using procedures approved by Transnuclear. After loading, exterior surfaces of the cask will be decontaminated using procedures and decontamination agents approved at the loading facility.

The cleaning agents, sealants, and lubricants have no significant effect on the cask and canister materials.

3.4.1.6 Hydrogen Generation

There is no mechanism for galvanic corrosion in the space between the DSC and the transfer cask, because both the inner shell of the TC and the outer shell of the DSC are stainless steel, and because the canister is sealed before the lid is placed on the transfer cask. Therefore, any concern for hydrogen generation applies solely inside the canister during wet loading.

Monitoring of the hydrogen concentration before and during welding operations will be performed to ensure that the hydrogen concentration does not exceed 2.4%. If the concentration exceeds 2.4%, welding operations will be suspended and the DSC will be purged with an inert gas.

Numerous NUHOMS® canisters fabricated using aluminum, neutron absorber, and stainless steel have been loaded in both borated and deionized water. Hydrogen monitoring has measured hydrogen in the range 0-2%, well below the 4% lower limit of flammability, provided that sufficient plenum space is provided between the water in the DSC and the inner top cover/shield.

3.4.1.7 Polymers (Transfer Cask and DSC)

The transfer cask lid and port cover o-rings may be fluorocarbon, silicone, EPDM, or other material with a service temperature range from -15 °F to 300 °F. Accident conditions assume the loss of the transfer cask seal, whose function is to retain helium, not radionuclides. All sealing

surfaces are stainless steel 304 or XM-19. Quick connect fittings and the neutron shield pressure relief valve also contain elastomer seals. The o-rings and quick connect fittings are subject to periodic inspection per Section 9.2.1.

The axial neutron shield is a proprietary reinforced polymer. The fire retardant mineral fill makes it self-extinguishing. Furthermore, the material is contained inside a steel shell, so that it is retained in place and isolated from sources of ignition. The trunnion plugs include polypropylene neutron shielding in Type 304 or Type XM-19 stainless steel. Polypropylene is slow burning to non-burning according to Table 24, Section 1 of the Handbook of Plastics and Elastomers [32].

Chapter 4 demonstrates that the transfer cask o-rings and sold neutron shielding remain below the upper limit of their service temperature. Polymers such as these used in the transfer cask have been demonstrated to be adequate for use in continuous thermal and radiation environments of spent fuel storage for 40 years duration. The intermittent usage of the transfer cask is less challenging for these materials.

The DSC uses o-rings at the connection of the drain tube to the inner top cover/shield, and in the quick connect fittings. The long term exposure to the thermal and radiation environment inside the DSC is likely to cause hardening of these seals. These seals do not provide any confinement function for either helium or radionuclides, and their function is not essential to reflooding and fuel removal operations.

3.4.1.8 Coatings (Transfer Cask)

Corrosion-resistant coatings are optional on transfer cask alloy steel bolts.

3.4.1.9 Effect of Degradation Mechanisms on the Performance of the System

For the environment and materials of the NUHOMS® HD system, there is no chemical, galvanic, thermal, or radiological reaction or degradation mechanism that would have a measurable adverse effect on design functions. There are no significant reactions that could reduce the overall integrity of the HSM, transfer cask, canister, or the spent fuel during storage. There are no reactions that would cause binding of the mechanical surfaces or of the fuel to fuel compartments.

3.4.2 Positive Closure

Positive closure is provided by the redundant closure welds for the inner top cover / shield and outer top cover plate and by the leak-tight DSC shell assembly.

3.4.3 Lifting Devices

There are no permanent lifting devices used for lifting a loaded DSC. The loaded DSC is always inside a transfer/transportation cask during handling.

The evaluation of lifting devices is performed in the transfer system (see Appendix 3.9.5).

3.4.4 Heat

3.4.4.1 Summary of Pressures and Temperatures

Temperatures and pressures for the 32PTH DSC, HSM-H and OS187H Transfer Cask are described in Chapter 4. Section 4.3 and Section 4.4 describe the thermal evaluations performed for normal, off-normal, and accident conditions. Section 4.5 describes the thermal evaluations during fuel loading/unloading operations. Maximum and minimum temperatures for the various components of the NUHOMS® HD System for normal, off-normal, and accident conditions are summarized in Table 4-1 to 4-6. These temperatures are used for the structural evaluations documented in Sections 3.6 and 3.7. Stress allowables for the cask components are a function of component temperature. The temperatures used to perform the structural analysis are based on actual calculated temperatures or conservatively selected higher temperatures.

Table 4-10 provides a summary of the maximum 32PTH DSC pressures for normal, off-normal and accident conditions. The pressures used in the 32PTH DSC stress analyses in Appendix 3.9.1 bound those summarized in Table 4-10.

3.4.4.2 Differential Thermal Expansion

Potential interference due to differential thermal expansion between the 32PTH DSC shell assembly, the basket assembly, and transfer cask components is evaluated in Appendix 3.9.1, Section 3.9.1.4.

3.4.4.3 Stress Calculations

The stress analyses have been performed using the acceptance criteria presented in Section 3.1.2. The structural analyses for the 32PTH DSC, the HSM-H and OS187H Transfer Cask are summarized in Sections 3.6 and 3.7, for normal, off-normal, and hypothetical accident conditions, respectively.

3.4.5 Cold

The 32PTH DSC and OS187H TC lifting, structural, and confinement materials are Type 304 and Type XM-19 stainless steels that are not subject to brittle fracture, so they do not impose a limit on low temperature operations. Cask lid and ram access port seal o-ring operating temperatures are -15 to -40 F depending on the material. The seal function is not confinement of radioactive material, but retention of helium for heat rejection. At very low ambient temperatures, this is not a concern, so the seal operating temperature lower limit does not impose an operations limit on the NUHOMS® HD system.

For operations below 32 °F, operations shall make provision to prevent freezing of water in the neutron shield.

3.5 Fuel Rods General Standards for 32PTH DSC

This section provides the temperature criteria used in the 32PTH DSC thermal evaluation for the safe storage and handling of SFA's in accordance with the requirements of 10CFR 72. This section also contains the analysis of the thermal and irradiation growth of the fuel assemblies to ensure adequate space exists within the 32PTH DSC cavity for the fuel assemblies to grow thermally under all conditions.

In addition, this section provides an evaluation of the fuel rod stresses and critical buckling loads due to accident drop loads.

3.5.1 Fuel Rod Temperature Limits

The fuel rod temperature limits during transfer operation and storage are defined by Interim Staff Guidance ISG11, revision 3. The temperature limits are summarized in the following table.

Transfer		Storage	
Normal/Off Normal	Accident	Normal	Off Normal/Accident
752°F	1058°F	752°F	1058°F

3.5.2 Fuel Assembly Thermal and Irradiation Growth

The thermal and irradiation growth of the fuel assemblies were calculated to ensure there is adequate space for the fuel assemblies to grow within the 32PTH DSC canister cavity. Detail thermal expansion evaluations of canister cavity versus lengths of basket and fuel assembly, canister ID vs. basket OD, canister OD vs. transfer cask ID, and overall length of canister vs. transfer cask cavity length are included in Appendix 3.9.1, Section 3.9.1.4 (page 3.9.1-78)

The extreme metal temperatures for the fuel cladding and canister under different cases are obtained from Chapter 4 for computation of the differential length growth. These temperatures are conservatively rounded and used in this calculation as listed in the following table.

Thermal Expansion Evaluation Cases

Cases \ Component Temperature	Length Growth Between Fuel Cladding and Canister	
	Fuel Cladding Temp. (°F)	**Canister (DSC Shell) Temp. (°F)**
Vacuum Drying	750	210
Transfer	730	390
Storage – Off Normal	700	310
Storage – Blocked Vent	810	500

The following table summaries the minimum gap between the canister cavity and the fuel assembly in the above thermal cases.

	Thermal Load Cases			
	Vacuum Drying	Transfer	Storage – Off Normal	Storage – Blocked Vent
Fuel assembly length	162.4 in.	162.4 in.	162.4 in.	162.4 in.
Total thermal growth	0.4 in.	0.38 in.	0.36 in.	0.43 in.
Irradiation growth	1.25 in.	1.25 in.	1.25 in.	1.25 in.
Total fuel assembly length after thermal growth	**164.05 in.**	**164.03 in.**	**164.01 in.**	**164.08 in.**
Min. canister cavity length	164.5 in.	164.5 in.	164.5 in.	164.5 in.
Canister thermal growth	0.2 in.	0.5 in.	0.3 in.	0.69 in.
Canister cavity length after thermal growth	**164.7 in.**	**165.0 in.**	**164.8 in.**	**165.19 in.**
Min. calculated gap	**0.65 in.**	**0.97 in.**	**0.79 in.**	**1.11 in.**

Based on the evaluations, there is adequate space within the 32PTH DSC cavity for thermal and irradiation growth of the fuel assemblies and spacers.

3.5.3 Fuel Rod Integrity During Drop Scenario

The purpose of this section is to calculate Zircaloy clad fuel rod stresses and critical buckling loads due to transfer cask side and end drop incidents.

3.5.3.1 Side Drop

The fuel rod side impact stresses are computed by treating the fuel rod as a continuous beam supported at locations of spacer grids. Continuous beam theory is used to determine the maximum bending moment in the entire beam. The maximum bending stress corresponding to the maximum bending moment in the cladding tubes is then calculated. The fuel gas internal pressure is also considered in the calculation. The cladding axial tensile stress due to the gas pressure is added to the bending stress due to the 75g drop load. The combined stresses in each cladding for different fuel assemblies are computed and tabulated in Table 3-12. It shows that among all fuel assemblies the highest axial stress is calculated to be 58,710 psi in the cladding of WE17×17OFA fuel assembly. This highest stress is lower than the yield strength of zircaloy (80,500 psi at 750 °F).

3.5.3.2 End Drop

Model

The failure mode of a fuel cladding during an end drop event is buckling of its sections between two spacer grids. The lowest section, among all sections between spacer grids, of each cladding is most likely the first one to buckle for it carries the most weight of the cladding. This lowest section of a cladding is therefore focused in the buckling analysis. An ANSYS model is created to simulate the lowest section of a cladding and its fuel content between two support spacer grids. The fuel pellets inside the cladding is assumed to have fused with each other and with cladding, ANSYS elements of plastic PIPE20 and elastic PIPE16 are used to simulate the cladding and the fuel pellets, respectively. Only a length of one section of a vertical cladding between two spacer grids is modeled. The weight of one entire cladding and its fuel content is conservatively applied at the top of the model. This applied force is calculated by dividing the weight of one assembly by the number of claddings in the assembly. The collinear cladding and fuel elements are coupled in lateral directions. Fuel assemblies WE15x15 and WE17x17OFA are selected as bounding cases for buckling analysis of cladding under the end drop event.

Large deformation option is used in the ANSYS analysis. A ramped vertical 120g inertial load is applied to the top of the ANSYS model. A small perturbation lateral force of 0.001 lb is applied at the mid span of the model to initiate a lateral deflection and begin buckling response of the cladding. Should the buckling of the cladding take place, the modeled structure of the cladding becomes unstable and the ANSYS solution will not be able to converge. The g load at which a last converged sub-step solution is produced in the ANSYS analysis is considered to be the critical axial buckling load of the cladding under the end drop event.

The geometry and the applied loads in each bounding fuel cladding are as follows. Reference to Chapter 2, Table 2-1, the fuel cladding thickness is reduced by 120 micron or 0.0027 in. (60 microns thickness oxidation at inner and outer surfaces).

a) WE 15x15 fuel cladding.

Cladding OD = 0.4193"

Cladding Thickness = 0.0216"

Fuel Pellet OD = 0.3659"

Fuel assembly weight = 1555 lb

No. of fuel claddings = 204

Length of cladding section between two spacer grids = 27"

Applied load at top of modeled cladding
= (Entire assembly weight / no. of claddings) × 120g
= (1555 lb / 204) × 120g
= 7.6225 lb × 120g
= 914.71 lb

b) WE 17x17OFA fuel cladding

Cladding OD = 0.3573"

Cladding Thickness = 0.0198"

Fuel Pellet OD = 0.3088"

Fuel assembly weight = 1575 lb

No. of fuel claddings = 264

Length of cladding section between two spacer grids = 25"

Applied load at top of modeled cladding
= (Entire assembly weight / no. of claddings) × 120g
= (1575 lb / 264) × 120 g
= 5.966 lb × 120g = 715.91 lb

Material properties (at 750 °F)

a) Fuel Cladding (Zircaloy)

Young's Modulus, $E = 10.4 \times 10^6$

Yield Strength, $S_y = 80{,}500$ psi

Ultimate Strength, $S_u = 91{,}760$ psi

Elongation, $e = 1.58\%$

Tangent Modulus,
$E_T = (91{,}760 - 80{,}500)$ psi / $[0.0158 - (80{,}500 / (10.4 \times 10^6))]$
$= 1.4 \times 10^6$ psi

Poisson Ratio, $\upsilon = 0.3$

b) Fuel Pellets (UO_2)

$E = 27.7 \times 10^6$ psi
$\upsilon = 0.316$

The fuel gas pressure inside the fuel cladding, which helps resist buckling, is conservatively ignored in the analysis of the end drop event.

3.5.3.3 Results

Side Drop

Table 3-12 summarizes the maximum bending stresses in various specified fuel cladding during the 75g side drop of their transfer cask. The maximum bending stress was calculated to be 58,710 psi in the cladding of fuel WE17x17OFA. It is less than the cladding yield strength of 80,500 psi at 750 °F. It is, therefore, concluded that the fuel cladding will not fail under the 75g side drop load.

End Drop

ANSYS produces all converged sub-step solutions throughout the ramped 120g load for both fuel assemblies WE15x15 and WE17x17OFA. Both claddings are stable without initiation of buckling beyond the design 75g drop load. The maximum stress intensities in claddings are also calculated in the analysis at the last converged solutions for fuel assemblies WE15x15 and WE17x17OFA. These maximum stress intensities in the claddings are tabulated in Table 3-13 and are compared against the yield strength of the cladding material.

3.5.4 Fuel Unloading

For unloading operations, the DSC will be filled with the spend fuel pool water through the siphon port. During this filling, the DSC vent port is maintained open with effluents routed to the plant's off-gas monitoring system.

When the pool water is added to a DSC cavity containing hot fuel and basket components, some of the water will flash to steam causing internal cavity pressure to rise. The steam pressure is released through the vent port. The initial flow rate of the reflood water must be controlled such that the internal pressure in the DSC cavity does not exceed 20 psig. This is assured by monitoring the maximum internal pressure in the DSC cavity during reflood event. The reflood of the DSC is considered as a "Service Level D" event and the design pressure of the DSC is 120 psig. Therefore, there is sufficient margin in the DSC internal pressure during the reflooding event to assure that the DSC will not be over pressurized.

The maximum fuel cladding temperature during reflooding process is significantly less than the vacuum drying condition owing to the presence of water/steam in the DSC cavity. Hence, the peak cladding temperature during the reflooding operation will be less than 734°F calculated for vacuum drying procedure A in Chapter 4, Section 4.5.1 when water circulates in the annulus between the DSC and transfer cask.

To evaluate the effects of the thermal loads on the fuel cladding during reflooding operations, a conservative high fuel rod temperature of 750°F and a conservative low quench water temperature of 50°F are used. These evaluations are performed in Chapter 4, Section 4.5.2. The calculated maximum fuel cladding stress is 25,910 psi. This calculated maximum stress is much less than the claddings yield stress of 80,500 psi. Therefore, cladding integrity is maintained during reflooding operation.

3.6 Normal Conditions of Storage and Transfer

This section presents the structural analyses of the 32PTH DSC, the HSM-H and the OS187H Transfer Cask subjected to normal conditions of storage and transfer. The analyses performed evaluate these three major NUHOMS® HD System components for the design criteria described in Section 3.1.2 of this chapter.

The 32PTH DSC is subjected to both storage and transfer loading conditions, while the HSM-H is only subjected to storage loading conditions and the OS187H Transfer Cask is only subjected to transfer loading conditions.

Numerical analyses have been performed for the normal and accident conditions, as well as for the lifting loads. In general, numerical analyses have been performed for the regulatory events. These analyses are summarized in the main body of this section, and described in detail in Appendices 3.9.1 through 3.9.9.

The detailed structural analysis of the NUHOMS® HD System is included in the following appendices:

Appendix 3.9.1	32PTH DSC (Canister and Basket) Structural Analysis
Appendix 3.9.2	OS187H Transfer Cask Body Structural Analysis
Appendix 3.9.3	OS187H Transfer Cask Top Cover and RAM Access Cover Bolts Analyses
Appendix 3.9.4	OS187H Transfer Cask Lead Slump and Inner Shell Buckling Analyses
Appendix 3.9.5	OS187H Transfer Cask Trunnion Analysis
Appendix 3.9.6	OS187H Transfer Cask Shield Panel Structural Analysis
Appendix 3.9.7	OS187H Transfer Cask Impact Analysis
Appendix 3.9.8	Damaged Fuel Cladding Structural Evaluation
Appendix 3.9.9	HSM-H Structural Analysis

3.6.1 32PTH DSC Normal Conditions Structural Analysis

Details of the structural analysis of the 32PTH DSC are provided on Appendix 3.9.1. The Fuel Basket and Canister are analyzed independently. The Fuel Basket is analyzed in Appendix 3.9.1, Section 3.9.1.2, while the Canister is analyzed in Appendix 3.9.1, Section 3.9.1.3. Three separate finite element models are constructed for the structural evaluation of the fuel basket while four finite element models are used for the structural evaluation of the canister shell.

3.6.1.1 32PTH DSC Fuel Basket Normal Condition Structural Evaluation

The fuel basket stress analysis is performed for normal condition loads during fuel transfer and storage. The detailed stress analysis is presented in Appendix 3.9.1, Section 3.9.1.2.3. A summary of the fuel basket load cases is provided in Appendix 3.9.1, Section 3.9.1.2.2.

The basket stress analysis is performed using a finite element method for the transfer handling, storage dead weight, and both transfer and storage thermal load cases. A 3-dimensional cross-section finite element model is utilized to evaluate the effect of transverse inertial loads on the fuel basket. The finite element model is described in detail in Appendix 3.9.1, Section 3.9.1.2.3.A (page 3.9.1-7). Analytical calculations are used for the vertical dead weight load case.

The mechanical properties of structural materials used in the basket, rail and canister are shown in the Appendix 3.9.1, Tables 3.9.1-1 and 3.9.1-2 as a function of temperature. All structural components of the fuel basket and support rails are constructed from SA-240, Type 304 stainless steel, with properties taken from AMSE B&PV Code [10].

ANSYS nonlinear elastic stress analyses are conducted for computing the elastic stresses in the fuel basket model. The nonlinearity of analysis results from the gaps in the model. In general, for each load case, the maximum total load is applied in small steps. The automatic time stepping program option "Autots" is activated. This option lets the program decide the actual size of the load-substep for a converged solution. Where shell elements are used, the shell middle surface nodal stress intensity is the membrane stress intensity and top or bottom surface stress intensity is the membrane plus bending stress intensity.

The calculated stresses in the 32PTH DSC fuel basket under normal conditions are summarized and compared with the corresponding ASME code allowable stresses for transfer load cases in Appendix 3.9.1, Table 3.9.1-3 and storage load cases in Appendix 3.9.1, Table 3.9.1-5.

The fusion weld is qualified by a pull test (shear). The required minimum test load is 16.5 kips. This load corresponds to the maximum fusion weld loads generated during a 75*g* hypothetical accident impact with a safety factor of 2 and a correction for material strength for room temperature testing. The maximum force generated in the fusion welds due to transfer load is 246 lb (Appendix 3.9.1, page 3.9.1-11) and thermal load in fusion weld during transfer is 1,253lb (Appendix 3.9.1, page 3.9.1-14). The combined load is 1,499 lb (1.5kip). This combined load is much smaller than the required test load of 16.5 kips.

Based on the results of these analyses, the design of the 32PTH DSC basket is structurally adequate with respect to normal condition transfer and storage loads.

3.6.1.2 32PTH DSC Canister Shell Normal Condition Structural Evaluation

This section summarizes the evaluation of the structural adequacy of the 32PTH DSC canister under all applied normal condition loads. Detail evaluation of the stresses generated in the

canister is presented in Appendix 3.9.1, Section 3.9.1.3.2 (page 3.9.1-36). The DSC canister shell buckling evaluation is presented in Appendix 3.9.1, Section 3.9.1.3.3 (page 3.9.1-61).

An enveloping technique of combining various individual loads in a single analysis is used in this evaluation for several load combinations. This approach greatly reduces the number of computer runs while remains conservative. However, for some load combinations, the stress intensities under individual loads are added to obtain resultant stress intensities for the specified combined loads. This stress addition at the stress intensity level for the combined loads, instead of at component stress level, is also a conservative way to reduce numbers of analysis runs.

The ANSYS calculated stresses are the total stresses of the combined membrane, bending, and peak stresses. These total stresses are conservatively taken to be membrane stresses (P_m) as well as membrane plus bending stresses ($P_L + P_b$) and are evaluated against their corresponding ASME code stress limits. In the case where the total stresses, evaluated in this manner, exceed the ASME allowable stresses, a detailed stress linearization is performed to separate the membrane, bending, and peak stresses. The linearized stresses are then compared to their proper Code allowable stresses. ASME B&PV Code Subsection NB [12] is used for evaluation of loads under normal conditions. The thermal stress intensities are classified as secondary stress intensities, Q, for code evaluations.

Material properties obtained from Reference 10 for the 32PTH DSC canister materials, taken at the highest metal temperature of 500° F (from thermal evaluation presented in Chapter 4). The ANSYS Multilinear Kinematic Harding material option of inelastic analysis is employed in the analyses of all canister accident side drops. A multi-linear stress-strain curve for type 304 stainless steel at 500° F is constructed using the yield and tensile stress values taken from Reference 10.

Elastic and elastic-plastic analyses are performed to calculate the stresses in the 32PTH DSC canister under the transfer and storage loads. These detail load cases are summarized in Appendix 3.9.1, Tables 3.9.1-9, 3.9.1-10 and 3.9.1-19.

The calculated stresses in the canister shell due to normal transfer loading conditions are summarized in Appendix 3.9.1, Tables 3.9.1-11, 12, 15, and 16. The stresses due to normal storage loading conditions are summarized in Appendix 3.9.1, Tables 3.9.1-20, and 21.

An alternate 32PTH DSC canister design with a composite top and/or bottom is also evaluated for their structural adequacy.

Details of the structural evaluation of the alternate canister composite bottom design under loads of normal conditions are provided in Appendix 3.9.1, Section 3.9.1.3.4 (page 3.9.1-64). For the alternate canister composite bottom design, the stresses in the canister under the normal transfer loading conditions are summarized in Appendix 3.9.1, Tables 3.9.1-24, 25, 26, and 27. The loads under the normal storage conditions are bounded by the loads under the normal transfer conditions.

Under the loads of both the normal transfer and storage conditions, the stresses generated in the canister will not be significantly different between the canister designs with an one-piece top and with a composite top. SAR Drawing 10494–72–4, Rev. 0 shows the alternate composite top.

As described in Chapter 8, Section 8.1.1.3, operation steps 7 and 13, a maximum of 60 psig air pressure may be applied at the canister vent port to assist draining of the water. The canister is structurally evaluated for this 60 psi internal pressure using the 2-D ANSYS finite element model described in Appendix 3.9.1, Section 3.9.1.3.2. The outer cover plate of the canister is removed from the 2-D model, since it is not yet installed during the application of this 60 psig air pressure. The maximum primary stress intensity and the maximum primary plus secondary stress intensity in the canister during the application of 60 psig air pressure are calculated to be 8,247 psi and 26,070 psi, respectively. Their corresponding stress limits as per ASME B&PV Code Subsection NB [12] are 16,400 psig and 49,200 psi, respectively. The application of 60 psig air pressure to the canister is therefore acceptable.

Based on the results of these analyses, the design of the 32PTH DSC canister is structurally adequate with respect to both transfer and storage loads under the normal conditions.

3.6.2 HSM-H Normal Conditions Structural Analysis

The design of the HSM-H for 32PTH DSC is the same as the HSM-H which is under NRC review as Amendment 8 to CoC 1004 for 24PTH DSC. Analyses performed for HSM-H with 24PTH DSC used bounding values to envelop both 24PTH DSC and 32PTH DSC. Following table shows how the bounding loads are used for structural evaluation of the HSM-H.

	Weight	Thermal
24PTH DSC (loaded weight)	93.7 kips	40.8 kw
32PTH DSC (loaded weight)	108.76 kips	34.8 kw
Weight used for HSM-H structural evaluation to envelop both 24PTH & 32PTH	110.0 kips (max.) (1) 72.0 kips (min.) (2)	
Thermal load used for HSM-H structural evaluation to envelop both 24PTH & 32PTH		40.8 kw

Notes:

1. Maximum weight is used for structural evaluation of the HSM-H.
2. Minimum weight is used for stability evaluation of the HSM-H.

Detail geometry descriptions, material properties, loadings, and structural evaluation of the HSM-H as presented in Amendment 8 to CoC 1004 for 24PTH DSC is reproduced and included in Appendix 3.9.9 of this chapter.

3.6.3 OS187H Transfer Cask Normal Conditions Structural Analysis

Details of the structural analysis of the OS187H Transfer Cask are provided in Appendices 3.9.2 through 3.9.7. The contents of each of these appendices are as follows.

- 3.9.2 OS187H Transfer Cask Body Structural Analysis
- 3.9.3 OS187H Transfer Cask Lid and RAM Access Cover Bolt Analyses
- 3.9.4 OS187H Transfer Cask Lead Slump and Inner Shell Buckling Analyses
- 3.9.5 OS187H Transfer Cask Trunnion Analysis
- 3.9.6 OS187H Transfer Cask Shield Panel Structural Analysis
- 3.9.7 OS187H Transfer Cask Impact Analysis

3.6.3.1 Structural Analysis of the Transfer Cask Body under Normal Conditions

The details of the structural analyses of the NUHOMS®-OS187H Transfer Cask body including the cylindrical shell assembly and bottom assembly, the top cover, and the local stresses at the trunnion/cask body interface are presented in Appendix 3.9.2. The specific methods, models and assumptions used to analyze the cask body for the various individual loading conditions specified in 10CFR72 [3] are described in that appendix.

The OS187H transfer cask body structural analyses generally use static or quasistatic linear elastic methods. The stresses and deformations due to the applied loads are generally determined using the ANSYS [33] computer program.

Table 3.9.2-1 of Appendix 3.9.2 Summarizes the maximum stresses in the Transfer Cask Body computed for normal conditions of transfer. The maximum stresses in each component are listed along with the normal loading condition that generates the stress. The results are evaluated against the ASME Code [7] design criteria described in Section 3.1.2 of this chapter.

Based on the results of these analyses, the design of the OS187H transfer cask is structurally adequate with respect to normal condition (Level A) transfer loads.

3.6.3.2 Transfer Cask Top Cover and RAM Access Cover Bolt Normal Condition Analysis

The detailed calculations for the top cover and RAM access cover bolts are presented in Appendix 3.9.3. The analysis is based on NUREG/CR-6007 [34]. The bolts are analyzed for the following normal loading conditions: operating pre-load, gasket seating load, internal pressure, and temperature changes.

The bolt preload is calculated to withstand the worst case load combination and to maintain a clamping (compressive) force on the closure joint, under normal conditions. Based upon the load combination results (see Appendix 3.9.3, Sections 3.9.3.3 and 3.9.3.8), it is shown that a positive (compressive) load is maintained on the clamped joint for all normal condition load combinations.

A summary of the calculated Top Cover bolt stresses is listed in Table 3-14 of this chapter. The calculations result in a maximum average tensile stress of 37.4 ksi, which is below the allowable tensile stress of 92.4 ksi for normal conditions. The maximum average shear stress in the bolts is due to torsion during pre-loading. This stress is 6.8 ksi, which is well below the allowable shear stress of 55.4 ksi. The maximum combined stress intensity due to tension plus shear plus bending is 74.0 ksi., which is also less than the maximum allowable stress intensity of 124.7 ksi.

A summary of the calculated RAM access bolt stresses is listed in Table 3-15 of this chapter. The analysis results in a maximum average tensile stress of 45.2 ksi, which is below the allowable tensile stress of 92.4 ksi for normal conditions. The maximum normal condition shear stress is 8.0 ksi, which is well below the allowable shear stress of 55.4 ksi. The maximum combined stress intensity due to tension plus shear plus bending is 97.0 ksi., which is also less than the maximum allowable stress intensity of 124.7 ksi.

3.6.3.3 Transfer Cask Normal Condition Trunnion Analysis

Appendix 3.9.5 presents the evaluation of the trunnion stresses in the NUHOMS®-OS187H Transfer Cask due to all applied loads during fuel loading and transfer operations.

NUHOMS® -OS187H transfer cask has two top trunnions constructed from SA-182 Gr. FXM19 and two bottom trunnions constructed from SA-182 Gr. F304. Both sets of trunnions are welded to the structural shell of the transfer cask, which is constructed from Type 304 stainless steel. The two top trunnions are used to first lift the cask, containing a canister and an empty basket, into a fuel pool for loading of the spent fuel. After the spent fuel has been loaded into the basket, the cask is lifted to a decontamination area. After draining and drying of the pool water, welding of the canister cover, and bolting of the cask lid, the cask is placed in a trailer for transfer to onsite HSM. The cask is vertically lifted onto the trailer and is initially supported by the bottom trunnions which are mated to transfer trailer. Then the cask is allowed to pivot about the bottom trunnions, into a horizontal position until the top trunnions rest on their supports in the trailer. Throughout the operation the maximum total load is applied to the cask top trunnions. After the cask has been placed on the trailer, it is supported by all four trunnions and is subject to a set of specified design handling loads.

Based on the loading and transfer scenario described above, the top trunnions are analyzed for 6*g* vertical lifting loads, and both sets of trunnions are evaluated for the prescribed set of transfer handling loads.

The transfer cask shell and trunnions are assumed to be at 300° F during transfer. This assumption is conservative based on the thermal evaluation performed in Chapter 4.

The calculated maximum trunnion stresses are summarized in Appendix 3.9.5, Table 3.9.5-1 and compared with their corresponding allowable stresses. Table 3.9.5-1 shows that all calculated trunnion stresses are less than their corresponding allowable stresses. Therefore, the NUHOMS®-OS187H Transfer Cask top and bottom trunnions are structurally adequate to withstand loads during lifting and transfer operations.

3.6.3.4 Transfer Cask Shield Panel Structural Analysis for Normal Conditions

Appendix 3.9.6 presents the evaluation of the stresses in the NUHOMS®-OS187H Transfer Cask neutron shield shell due to all applied loads during fuel loading and transfer operations.

A finite element model was built for the structural analysis of the outer neutron shield shell, end closure, central plates and structural shell. These structural components were modeled with two dimensional axisymmetric elements. The same finite element model is used for all loading conditions.

Table 3.9.6-1 of Appendix 3.9.6 summarizes the calculated stresses for the transfer cask lifting and transfer loads. Based on the results of the analysis, it is concluded that the outer shell structure is structurally adequate for the specified transfer loads.

3.7 Off Normal and Hypothetical Accident Conditions

This section presents the structural analyses of the 32PTH DSC, the HSM-H and the OS187H Transfer Cask subjected to off normal and hypothetical accident conditions of storage and transfer. The analyses are summarized in Sections 3.7.1, 3.7.2 and 3.7.3 of this chapter and are evaluated against the design criteria described in Section 3.1.2 of this chapter.

The 32PTH DSC is subjected to both storage and transfer loading conditions, while the HSM-H is only subjected to storage loading conditions and the OS187H Transfer Cask is only subjected to transfer loading conditions.

3.7.1 32PTH DSC Off Normal and Accident Conditions Structural Analysis

Details of the structural analysis of the 32PTH DSC are provided in Appendix 3.9.1. The Fuel Basket and Canister are analyzed independently. The Fuel Basket is analyzed in Appendix 3.9.1, Section 3.9.1.2, while the Canister is analyzed in Section 3.9.1.3. Three separate finite element models are constructed for the structural evaluation of the fuel basket, while four finite element models are used for the structural evaluation of the canister shell.

3.7.1.1 32PTH DSC Fuel Basket Off Normal and Accident Condition Structural Analysis

3.7.1.1.1 32PTH Fuel Basket Off Normal and Accident Condition Stress Analysis

The fuel basket stress analyses are performed for off normal and accident condition loads during fuel transfer and storage. The detailed stress analysis is presented in Appendix 3.9.1, Section 3.9.1.2.3 (page 3.9.1-7). A summary of the fuel basket load cases is provided in Section 3.9.1.2.2 (page 3.9.1-5).

The basket stress analyses are performed using a finite element method for the transfer side drop impact loads, as well as, storage seismic loads, and both transfer and storage thermal load cases. A 3-dimensional cross-section finite element model is utilized to evaluate the effect of transverse inertial loads on the fuel basket. The finite element model is described in detail in Appendix 3.9.1, Section 3.9.1.2.3.A (page 3.9.1-7). Analytical calculations are used for the axisymmetric transfer end drop load case.

The mechanical properties of structural materials used in the basket, rail and canister are shown in the Appendix 3.9.1, Tables 3.9.1-1 and 3.9.1-2 as a function of temperature. All structural components of the fuel basket and support rails are constructed from SA-240, Type 304 stainless steel, with properties taken from AMSE B&PV Code [10].

Nonlinear elastic stress analyses are conducted for computing the elastic stresses in the fuel basket model. The nonlinearity of analysis results from the gaps in the model. In general, for each load case, the maximum total load is applied in small steps. The ANSYS automatic time

stepping program option "Autots" is activated. This option lets the program decide the actual size of the load-substep for a converged solution. Where shell elements are used, the shell middle surface nodal stress intensity is the membrane stress intensity and the top or bottom surface stress intensity is the membrane plus bending stress intensity.

The calculated stresses in the 32PTH DSC fuel basket is summarized and compared with their corresponding ASME code allowable stresses. Tables 3.9.1-4a and 3.9.1-4b of Appendix 3.9.1 show these summaries for the transfer accident loads and Table 3.9.1-5 for the storage accident loads.

The maximum shear load in the fusion welds during the accident loading condition is calculated in Appendix 3.9.1 (page 3.9.1-16). The calculated maximum shear force during side drop is 6,897 lb.

The fusion weld is qualified by a pull test (shear). The minimum test load is 16.5 kips. This test load includes a safety factor of 2 and a correction for material strength for room temperature testing.

Based on the results of these analyses, the design of the 32PTH DSC basket is structurally adequate with respect to off-normal and accident conditions of transfer and storage loads.

3.7.1.1.2 32PTH DSC Fuel Basket Accident Condition Buckling Analysis

Buckling analysis of the fuel basket plates and support rails are only performed for the bounding hypothetical accident condition impact loads. The accident condition buckling evaluation is presented in detail in Appendix 3.9.1, Section 3.9.1.2.4 (page 3.9.1-26).

Only the most critical fuel basket section is analyzed in detail. The critical basket section is depicted in Appendix 3.9.1, Figure 3.9.1-11. This approach is then validated by performing a buckling evaluation for the entire fuel basket cross section for the worst case loading condition.

All structural members of the 32PTH fuel basket are constructed from SA-240 Type 304 stainless steel. A bilinear stress-strain curve for SA-240 Type 304 stainless steel is used for the elastic-plastic buckling analysis.

The material properties used for the basket plates are taken from ASME Code, Section II, Part D [10], at 611° F. This temperature represents an average temperature for the fuel basket section analyzed and depicted in Appendix 3.9.1, Figure 3.9.1-11.

Nonlinear stress analyses are conducted in order to evaluate the plastic buckling loads for the 32PTH DSC basket plates. The three critical azimuth drop orientations analyzed are:

i) 0° (load applied in the direction parallel to the basket plates)
ii) 30° (load applied at 30° relative to the basket plate direction)
iii) 45° (load applied at 45° relative to the basket plate direction)

In order to calculate the buckling load, a small three-dimensional ANSYS finite element model is constructed using SHELL43 plastic large strain shell elements. This model is shown in Appendix 3.9.1, Figure 3.9.1-12. The small model is constructed by selecting the appropriate elements and nodes from full basket cross section model as described in Appendix 3.9.1, Section 3.9.1.2.3 (page 3.9.1-7). As described in Section 3.9.1.2.3.A., the stiffness from aluminum plates is conservatively neglected but their weight is accounted for in the applied loads.

The loading on the small model are appropriately transferred from the full size basket loading. A maximum load of 200*g* is applied in each analysis. The ANSYS automatic time stepping option "AUTOTS" is activated. This option lets the program decide the actual size of the load sub-step for a converged solution. The program stops at the load sub-step that fails to result in a converged solution. The last load step with a converged solution is used to compute the allowable collapse load for the fuel basket grid.

The following table summarizes the input load and last converged load for all three load cases:

Basket Orientation	200*g* loads (psi)			Last converged Load (g)	Max. Deflection u_x (in)
	Vertical Load (lb)	Lateral Pressure (psi)	Applied Acceleration		
Vertical	239,400	0	(0, 200, 0)	107	0.005
30°	207,326	129.0	(-100, 173, 0)	84	0.1090
45°	169,281	182.4	(-141, 141, 0)	84	0.1277

As per paragraph F-1340 [13], the acceptability of a component may be demonstrated by collapse load analysis. The allowable collapse load shall not exceed 100% of plastic analysis collapse load ([13] F-1341.3). The plastic analysis collapse load is defined as that determined by plastic analysis according to the criteria given in ASME, Section III, Appendix II, Paragraph II-1430 ([13] F-1321.6(c)).

It is seen from the above table that the 45° drop orientation is critical with respect to buckling. Using the methodology described in II-1430 ([13] F-1321.6(c)) the allowable collapse load is determined for the critical 45° azimuth drop orientation, and is 81*g* (see Figure 3.9.1-13 of Appendix 3.9.1). For the 0° drop orientations, collapse load will be higher than 81g, and for the 30° drop orientation, collapse load will be of the same order of magnitude.

The small finite element model technique used for the buckling analyses of the fuel basket is verified by a full basket cross section finite element analysis as well as analytical methods. Details of these verification methods are provided in Appendix 3.9.1, Section 3.9.1.2.4 (page 3.9.1-29).

Since the critical collapse load for the 32PTH DSC basket (81*g* for the 45° Orientation) is greater than the maximum design acceleration of 75*g*, the basket will not fail in buckling during the accident condition events.

3.7.1.1.3 32PTH DSC Fuel Basket Support Rail Accident Condition Buckling Analysis

Nonlinear stress analyses are conducted to evaluate the buckling loads for the 32PTH fuel basket support rails. A nonlinear finite element analysis, with large deflection option, is conducted to evaluate the buckling loads for the support rails.

In order to calculate the buckling load for both support rail types, small three-dimensional ANSYS finite element models are constructed using SHELL43 plastic large strain shell elements. The small finite element models are extracted from large basket cross section model described in Appendix 3.9.1, Section 3.9.1.2.4 (page 3.9.1-32). A fuel tube compartment is included in both models to correctly simulate the stiffness of the top face of the rails.

The vertical loading on the rail models are appropriately transferred from the full size basket loads. A maximum load of 200*g* is applied in each analysis. The automatic time stepping option AUTOTS is also activated. This option lets the program decide the actual size of the load sub-step for a converged solution. The program stops at the load sub-step that fails to result in a converged solution. The last load step with a converged solution is used to compute the allowable collapse load for each model.

For both the large and small support rails, the finite element analysis converged up to the maximum 200*g* applied load. Therefore, both support rails are stable up to 200*g*. The results of the analysis show that the buckling loads of support rails are much higher than the maximum design acceleration of 75*g*. Consequently, there is no potential for the basket support rails to buckle during side drop event under the hypothetical accident condition.

3.7.1.2 32PTH DSC Canister Shell Off Normal and Accident Condition Structural Evaluation

3.7.1.2.1 32PTH Canister Shell Off Normal and Accident Condition Stress Analysis

An enveloping technique of combining various individual loads in a single analysis is used in this evaluation for several load combinations. This approach greatly reduces the number of computer runs while remains conservative. However, for some load combinations, the stress intensities under individual loads are added to obtain resultant stress intensities for the specified combined loads. This stress addition at the stress intensity level for the combined loads, instead of at component stress level, is also a conservative way to reduce numbers of analysis runs.

The ANSYS calculated stresses are the total stresses of the combined membrane, bending, and peak stresses. These total stresses are conservatively taken to be membrane stresses (P_m) as well as membrane plus bending stresses ($P_L + P_b$) and are evaluated against their corresponding ASME code stress limits. In the case where the total stresses, evaluated in this manner, exceed the ASME allowable stresses, a detailed stress linearization is performed to separate the membrane, bending, and peak stresses. The linearized stresses are then compared to their proper Code allowable stresses. ASME B&PV Code Subsection NB [12] is used for evaluation of loads under off normal conditions and Appendix F [13] for evaluation of loads under hypothetical accident conditions. The thermal stress intensities are classified as secondary stress intensities, Q, for code evaluations.

Material properties obtained from Reference 10 for the 32PTH DSC canister materials, taken at the highest metal temperature of 500° F (from thermal evaluation presented in Chapter 4). The ANSYS Multilinear Kinematic Harding material option of inelastic analysis is employed in the analyses of all canister accident side drops. A multi-linear stress-strain curve for type 304 stainless steel at 500° F is constructed using the yield and tensile stress values taken from [10].

Elastic and elastic-plastic analyses are performed to calculate the stresses in the 32PTH DSC canister under the transfer and storage loads. These load cases are summarized in Appendix 3.9.1, Tables 3.9.1-9, 3.9.1-10 and 3.9.1-19. All side drop loads are analyzed by elastic-plastic analyses and the rest by elastic analyses.

The calculated stresses in the canister shell due to off-normal and accident transfer loading conditions are summarized in Appendix 3.9.1, Tables 3.9.1-12, 13, 14, 16, 17, and 18. The stresses due to accident storage loading conditions are summarized in Appendix 3.9.1, Tables 3.9.1-20, and 21.

The alternate 32PTH DSC canister design with the composite bottom is also evaluated for the worst case accident condition loads, which bounds all possible applied loads to the canister. Details of the Alternate canister design structural evaluation are provided in Appendix 3.9.1, Section 3.9.1.3.4 (page 3.9.1-64).

For the alternate canister composite bottom design, the stresses in the off-normal and accident transfer loading conditions are summarized in Appendix 3.9.1, Tables 3.9.1-28 to 35. The loads

under the off normal and accident storage conditions are bounded by the loads under the off normal and accident transfer conditions, except the bottom end drop load.

For the alternate canister composite top design, as shown in SAR Drawing 10494-72-4, Rev. 0, only the bottom end drop load, out of all specified off-normal and accident loads, will generate significantly different stresses in the composite top from that in the one-piece top. Therefore a 2-D finite element model is created for the canister with the alternate composite top and is analyzed for a 75g bottom end drop load. The maximum primary membrane plus bending stress intensity in the entire canister is calculated to be 22,003 psi. The limit for a general primary membrane stress intensity is given 40,600 psi by ASME B&PV Code Appendix F [13]. The calculated maximum primary membrane plus bending stress intensity of 22,003 psi is less than the limit for a general primary membrane stress intensity of 40,600 psi. Therefore, the alternate canister composite top design is structurally adequate.

Based on the results of these analyses, the design of the 32PTH DSC canister is structurally adequate with respect to off-normal and accident condition transfer and storage loads.

3.7.1.2.2 32PTH DSC Canister Shell Accident Condition Buckling Analysis

This section summarizes the evaluation of 32PTH DSC canister against buckling under a vertical end drop during transfer operations. The detail of the DSC canister shell buckling analysis is provided in Appendix 3.9.1, Section 3.9.1.3.3 (page 3.9.1-61). A finite element plastic analysis with large displacement option is performed to monitor occurrence of canister shell buckling under the specified loads.

The thermal evaluation presented in Chapter 4 shows that the metal temperatures of the entire canister are below 500° F during the transfer operations. The material properties of the canister at 500° F are therefore conservatively used for the canister buckling analysis.

The following two hypothetical accident load cases for the canister are considered in this buckling analysis.

Buckling Load Case 1: 15 psig external pressure and 75*g* axial acceleration due to 30 foot hypothetical accident condition drop

Buckling Load Case 2: 30 psig internal pressure and 75*g* axial acceleration due to 30 foot hypothetical accident condition drop

The two-dimensional axisymmetric finite element model of the canister described in Appendix 3.9.1, Section 3.9.1.3.2.D.2 (page 3.9.1-39) for the DSC canister stress analysis is used for this analysis. Since the top end of the canister is heavier than the bottom end, it is a more severe case when the canister drops on its bottom end. A bottom end drop is therefore chosen for analysis in this calculation.

For each load case, a quasi-static plastic analysis consisting of two load steps is performed to monitor the buckling of canister. The first load step applies external pressure or internal pressure

alone. A subsequent inertial load of 150*g* is added in the second load step. The outer surface of the canister bottom is held in order to simulate the case that the canister drops on a rigid cask bottom face.

In the load step 1, the stepped external or internal pressure is applied as a static load.

In the load step 2, the weight of the canister internals (basket and fuel assemblies) is accounted for by applying an equivalent internal pressure on the canister bottom. This inertial load is uniformly distributed over the bottom surface of the canister cavity.

A multilinear stress-strain relationship (with kinematic hardening) is used to obtain stresses and deflections beyond the elastic limit of the material. The large deflection option in ANSYS is activated to monitor the buckling response.

The ANSYS program stops at the first load sub-step that fails to result in a converged solution, corresponding to buckling of the structure. When the structure buckles and the ANSYS solution fails to converge, the loads applied in the last converged load sub-step are considered to be the critical buckling load for the structure. The 150*g* drop loads applied in load step 2 is ramped in small sub-steps (1*g* load increment in each sub-step).

In both load cases, converged solutions are obtained up to 113.24*g* load. This load is much higher than the required 75*g* load in either Load Case 1 or 2. The analysis shows that the canister does not buckle up to an end drop load of 113.24*g*, which is well beyond the design 75*g* load. It is, therefore, concluded that buckling of the canister will not occur during a hypothetical accident end drop.

3.7.2 HSM-H Off Normal and Accident Conditions Structural Analysis

The design of the HSM-H for 32PTH DSC is the same as the HSM-H which is under NRC review as Amendment 8 to CoC 1004 for 24PTH DSC. Analyses performed for HSM-H with 24PTH DSC used bounding values to envelop both 24PTH DSC and 32PTH DSC. Following table shows how the bounding loads are used for structural evaluation of the HSM-H.

	Weight	Thermal
24PTH DSC (loaded weight)	93.7 kips	40.8 kw
32PTH DSC (loaded weight)	108.76 kips	34.8 kw
Weight used for HSM-H structural evaluation to envelop both 24PTH & 32PTH	110.0 kips (max.) (1) 72.0 kips (min.) (2)	
Thermal load used for HSM-H structural evaluation to envelop both 24PTH & 32PTH		40.8 kw

Notes:

1. Maximum weight is used for structural evaluation of the HSM-H.
2. Minimum weight is used for stability evaluation of the HSM-H.

Detail geometry descriptions, material properties, loadings, and structural evaluation of the HSM-H as presents in Amendment 8 to CoC 1004 for 24PTH DSC is reproduced and included in Appendix 3.9.9 of this Chapter.

3.7.3 OS187H Transfer Cask Off Normal and Accident Conditions Structural Analysis

3.7.3.1 Structural Analysis of the Transfer Cask Body for Off Normal and Accident Conditions

The details of the structural analyses of the NUHOMS®-OS187H Transfer Cask body including the cylindrical shell assembly and bottom assembly, the top cover, and the local stresses at the trunnion/cask body interface are presented in Appendix 3.9.2. The specific methods, models and assumptions used to analyze the cask body for the various individual loading conditions specified in 10CFR72 [3] are described in that appendix.

The OS187H transfer cask body structural analyses generally use static or quasistatic linear elastic methods. The stresses and deformations due to the applied loads are generally determined using the ANSYS [33] computer program.

The maximum stresses in each of the major components of the transfer cask are reported for each load case and load combination in Appendix 3.9.2, Table 3.9.2-1. The results are evaluated against the ASME Code [7] design criteria described in Section 3.1.2 of this chapter.

Based on the results of these analyses, the design of the OS187H transfer cask is structurally adequate with respect to off normal and hypothetical accident transfer loads.

3.7.3.2 Transfer Cask Top Cover and RAM Access Cover Bolt Accident Condition Analysis

The detailed calculations for the top cover and RAM access cover bolts are presented in Appendix 3.9.3. The analysis is based on NUREG/CR-6007 [34]. The bolts are analyzed for the hypothetical accident condition impact loads and load combinations.

A summary of the calculated top cover bolt stresses is listed in Appendix 3.9.3, Section 3.9.3.5 (page 3.9.3-13). The calculations result in a maximum average tensile stress of 106.6 ksi, which is below the allowable tensile stress of 115.5 ksi for accident conditions. The maximum average shear stress in the bolts is due to torsion during pre-loading. This stress is 6.8 ksi, which is well below the allowable shear stress of 69.3 ksi.

A summary of the calculated RAM access bolt stresses is listed in Appendix 3.9.3, Section 3.9.3.10 (page 3.9.3-26). The analysis results in a maximum average tensile stress of 45.2 ksi, which is below the allowable tensile stress of 115.5 ksi for accident conditions. The maximum accident shear stress is 9.5 ksi, which is well below the allowable shear stress of 69.3 ksi.

3.7.3.3 Transfer Cask Lead Slump Analysis

Appendix 3.9.4 presents the details of the OS187H Transfer Cask lead slump evaluation. The load considered is a 75*g* top and bottom end drop load in both hot (115° F) and cold (-20° F) ambient environments.

During a hypothetical accident condition end drop, permanent deformation of the lead gamma shield may occur. The lead gamma shield is supported by friction between the lead and transfer cask shells, in addition to bearing at the end of the lead column.

A nonlinear finite element analysis is performed in order to quantify the amount of lead slump generated during an end drop event. A 2-dimensional axisymmetric ANSYS finite element model is constructed for this purpose. The displacement results are used to determine the maximum size of the axial gap that develops between the lead gamma shield column and the structural shell of the transfer cask.

Figures 3.9.4-9 and 3.9.4-10 of Appendix 3.9.4 show the deformed shape of the transfer cask for 75*g* bottom end drop, and Figures 3.4.9-11 and 3.9.4-12 of Appendix 3.9.4 show the deformed shape of the transfer cask for the 75*g* top end drop. The maximum calculated lead slump is 0.787 inches and occurs during the 75g bottom end drop in the 115°F hot ambient environment. The effect of this cavity size on the shielding ability of the transport package is evaluated in Chapter 8.

3.7.3.4 Transfer Cask Inner Containment Buckling Analysis

Appendix 3.9.4 also presents the details of the evaluation of the structural adequacy of the OS187H Transfer Cask inner shell with respect to bucking. The load considered includes an internal pressure of 30 psig and a 75*g* top and bottom end drop load in both hot (115° F) and cold (-20° F) ambient environments.

An ANSYS elastic-plastic buckling analysis is performed for the transfer cask end drop cases. A 100*g* drop load, which is greater than the design load of 75*g*, is applied to the ANSYS model. This100*g* drop load was ramped in small increments by many load sub-steps. The ANSYS solution was set to stop and exit at any load sub-step that fails to result in a converged solution. The failure of convergence represents the onset of buckling of the structure. Once the ANSYS solution fails to converge, the loads applied in the last converged load sub-step will be considered the critical buckling load for the structure.

The ANSYS solutions have converged at all load sub-steps in each of the 100*g* drop load cases. This indicates that the transfer cask will not buckle during 75*g* end drops in both 115° F and -20° F ambient environments.

3.7.3.5 Transfer Cask Trunnion Analysis

Appendix 3.9.5 presents the evaluation of the trunnion stresses in the NUHOMS®-OS187H Transfer Cask due to all applied loads during fuel loading and transfer operations.

The loads applied to the transfer cask trunnions only occur during normal condition loading and fuel transfer. There are no hypothetical accident condition events that cause loads to be applied to the trunnions.

The calculated maximum normal condition trunnion stresses are summarized in Table 3.9.5-1 and compared with their corresponding allowable stresses.

3.7.3.6 Transfer Cask Shield Panel Structural Analysis for Accident Conditions

Appendix 3.9.6 presents the evaluation of the stresses in the NUHOMS®-OS187H Transfer Cask neutron shield shell due to all applied loads during fuel loading and transfer operations.

The neutron shield shell is only designed to withstand normal condition transfer cask lifting and fuel transfer loads. Consequently, the neutron shield shell is only analyzed for these loads, and is not analyzed to hypothetical accident condition events.

The calculated maximum normal condition neutron shield shell stresses are summarized in Table 3.9.6-1 of Appendix 3.9.6.

3.7.3.7 Transfer Cask Impact Analysis

In spite of the incredible nature of any scenario that could lead to a drop accident for the Transfer Cask, a conservative range of drop scenarios are developed and evaluated. These bounding scenarios assure that the integrity of the DSC and spent fuel cladding is not compromised. Analyses of these scenarios demonstrate that the Transfer Cask will maintain the structural integrity of the DSC pressure containment boundary. Therefore, there is no potential for a release of radioactive materials to the environment due to a cask drop.

Appendix 3.9.7 presents the computation of the peak decelerations of NUHOMS® OS187H Transfer Cask during impact, subsequent to the hypothetical accident drop onto the concrete pad/soil system during transfer operations. The analytical methodology described in Reference 35 is used to perform this evaluation.

The hypothetical accident condition drop consists of an 80 inch end drop, side drop, and center of gravity (C.G.) over corner drop. The transfer cask is assumed rigid as compared to the flexibility of the concrete slab/soil system, which consists of a 36 inch thick concrete pad, with #11 rebar on 12 spacing, the at top and bottom and 2" coverage.

The following table summarizes the results of the analysis described in detail in Appendix 3.9.7.

Drop Orientation	Peak Deceleration (gs)	Target Penetration Depth (in.)
End Drop	49	3.10
Side Drop	44	2.5
Corner Drop	16	6.5

The ranges of drop scenarios conservatively selected for design are:

1. A horizontal side drop from a height of 80 inches (75g horizontal drop).

2. Vertical end drops for the NUHOMS® HD system are non-mechanistic and thus, no end drops are postulated for the 32PTH DSC. However, 75g vertical end drop analyses are performed as a means of enveloping the 16g corner drop (in conjunction with the 75g horizontal side drop).

3. An oblique corner drop from a height of 80 inches at an angle of 30° to the horizontal, onto the top or bottom corner of the Transfer Cask. This case is not specifically evaluated. The side drop and end drop cases envelop the corner drop.

3.8 References

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6. American National Standards Institute, ANSI N14.5-1997, Leakage Tests on Packages for Shipment of Radioactive Materials.

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3.9 Appendices

The detailed structural analyses of the NUHOMS® HD system are included in the following appendices:

Appendix 3.9.1	32PTH DSC (Canister and Basket) Structural Analysis
Appendix 3.9.2	OS187H Transfer Cask Body Structural Analysis
Appendix 3.9.3	OS187H Transfer Cask Top Cover and RAM Access Cover Bolt Analyses
Appendix 3.9.4	OS187H Transfer Cask Lead Slump and Inner Shell Buckling Analysis
Appendix 3.9.5	OS187H Transfer Cask Trunnion Analysis
Appendix 3.9.6	OS187H Transfer Cask Shield Panel Structural Analysis
Appendix 3.9.7	OS187H Transfer Cask Impact Analysis
Appendix 3.9.8	Damaged Fuel Cladding Structural Evaluation
Appendix 3.9.9	HSM-H Structural Analysis

3.10 ASME Code Exceptions

The confinement boundary of the 32PTH DSC canister shell, the inner top cover/shield plug, the inner bottom cover, the siphon vent block, and the siphon/vent port cover plate are designed, fabricated and inspected in accordance with the ASME Code Subsections NB to the maximum practical extent. The basket is designed, fabricated and inspected in accordance with ASME Code Subsection NG to the maximum practical extent. Other canister components (such as outer bottom cover and shield plugs) are not governed by the ASME Code.

ASME Code Exceptions for the 32PTH DSC

Reference ASME Code Section/Article	Code Requirement	Exception, Justification & Compensatory Measures
NCA	All	Not compliant with NCA
NB-1100	Requirements for Code Stamping of Components	The canister shell, the inner top cover/shield plug, The shell bottom, the inner bottom cover (alternate bottom design), and the siphon/vent port cover are designed & fabricated in accordance with the ASME Code, Section III, Subsection NB to the maximum extent practical. However, Code Stamping is not required. As Code Stamping is not required, the fabricator is not required to hold an ASME "N" or "NPT" stamp, or to be ASME Certified.
NB-2130 NB-4121	Material must be supplied by ASME approved material suppliers Material Certification by Certificate Holder	Material is certified to meet all ASME Code criteria but is not eligible for certification or Code Stamping if a non-ASME fabricator is used. As the fabricator is not required to be ASME certified, material certification to NB-2130 is not possible. Material traceability & certification are maintained in accordance with TN's NRC approved QA program.
NB-4243 and NB-5230	Category C weld joints in vessels and similar weld joints in other components shall be full penetration joints. This welds shall be examined by UT or RT and either PT or MT	The joint between the outer top cover and inner top cover/shield plug and shell are design and fabricated per ASME Code Case N-595-3. The welds are partial penetration welds and the root and final layer are PT examined.
NB-2531	Vent & siphon Port Cover; straight beam UT per SA-578 for all plates for vessel	SA-578 applies to 3/8" and thicker plate only; allow alternate UT techniques to achieve meaningful UT results.
NB-6100 and 6200	All completed pressure retaining systems shall be pressure tested	The inner top cover/shield plug assembly is not pressure tested due to the manufacturing sequence. The inner top cover/shield plug assembly weld is helium leak tested when fuel is loaded and then covered with the outer top closure plate.

Reference ASME Code Section/Article	Code Requirement	Exception, Justification & Compensatory Measures
NB-7000	Overpressure Protection	No overpressure protection is provided for the 32PTH DSC. The function of the 32PTH DSC is to contain radioactive materials under normal, off-normal, and hypothetical accident conditions postulated to occur during transportation. The 32PTH DSC is designed to withstand the maximum internal pressure considering 100% fuel rod failure at maximum accident temperature. The 32PTH DSC is pressure tested in accordance with the requirements of 10CFR71 and TN's approved QA program.
NB-8000	Requirements for nameplates, stamping & reports per NCA-8000	The 32PTH DSC nameplates provide the information required by 10CFR71, 49CFR173, and 10CFR72 as appropriate. Code stamping is not required for the 32PTH DSC. QA Data packages are prepared in accordance with the requirements of 10CFR71, 10CFR72, and TN's approved QA program.
NB-1132	Attachments with a pressure retaining function, including stiffeners, shall be considered part of the component.	Outer bottom cover (item 52), bottom plate (item 55), bottom casing plate (item 61), side casing plate (item 62), top shield plug shield plate (item 69), grapple ring and grapple ring support are outside code jurisdiction; these components together are much larger than required to provide stiffening for the confinement boundary cover. These component welds are subject to root and final PT examinations.

ASME Code Exceptions for the 32PTH DSC Fuel Basket

Reference ASME Code Section/Article	Code Requirement	Exception, Justification & Compensatory Measures
NG-1100	Requirement for Code Stamping of Components	The 32PTH DSC baskets are designed & fabricated in accordance with the ASME Code, Section III, Subsection NG to the maximum extent practical as described in the SAR, but Code Stamping is not required. As Code Stamping is not required, the fabricator is not required to hold an ASME N or NPT stamp or be ASME Certified.
NG-2000	Use of ASME Material	Material is certified to meet all ASME Code criteria but is not eligible for certification or Code Stamping if a non-ASME fabricator is used. As the fabricator is not required to be ASME certified, material certification to NG-2130 is not possible. Material traceability & certification are maintained in accordance with TN's NRC approved QA program. The poison material and aluminum plates are not used for structural analysis, but to provide criticality control and heat transfer. They are not ASME Code Class I materials. See note 1.
NCA	All	Not compliant with NCA as no code stamp is used.

Note:

1. Because Subsection NCA does not apply, the NCA-3820 requirements for accreditation or qualification of material organizations do not apply. CMTR's shall be provided using NCA-3862 for guidance.

Table 3-1
Codes and Standards for the Fabrication and Construction of Principal Components

Component, Equipment, Structure	Code of Construction
32PTH DSC Shell Assembly	ASME Code, Section III, Subsection NB (1998 Edition through 2000 Addenda) Code Case N-595-3 and ISG-4
32PTH DSC Basket	ASME Code, Section III, Subsection NG (1998 Edition through 2000 Addenda)
Transfer Cask	ASME Code, Section III, Subsection NC (1998 Edition through 2000 Addenda)
HSM-H	- ACI 349-97 (Concrete) - AISC Ninth Edition (Structural Steel) - AWS D1.1-98 (Structural Welds) - ASCE 7-95 (Loads) - ANSI 57.9-84 (Loads & Load Combinations)

Table 3-2
Summary of Stress Criteria for Subsection NB Pressure Boundary Components

Loadings	Stress Category	Notes
Design [NB-3221]	$P_m \leq 1.0S_m$ $P_L \leq 1.5S_m$ P_m (or P_L) + $P_b \leq 1.5S_m$ $Fp \leq 1.5S_y$	
Service Level A [NB-3222]	P_m (or P_L) + P_b + $Q \leq 3.0S_m$	Notes 1 & 2
Service Level B [NB-3223]	$P_m \leq 1.0S_m$ $P_L \leq 1.5S_m$ P_m (or P_L) + $P_b \leq 1.5S_m$ P_m (or P_L) + P_b + $Q \leq 3.0S_m$ (Note 1)	Note 3
Service Level C [NB-3224]	$P_m \leq \max(1.2S_m, 1.0S_y)$ $P_L \leq \max(1.8S_m, 1.5S_y)$ P_m (or P_L) + $P_b \leq \max(1.8S_m, 1.5S_y)$	

continued...

Table 3-2
Summary of Stress Criteria for Subsection NB Pressure Boundary Components (continued)

Loadings	Stress Category	Notes
Carbon Steel Components (e.g., shield plugs)		
Level D Elastic Analysis [NB-3225, App. F]	$P_m \leq 0.7S_u$ P_m (or P_L) + P_b $\leq 1.0S_u$	Note 4
Level D Plastic Analysis [NB-3225, App. F]	$P_m \leq 0.7S_u$ P_m (or P_L) + P_b $\leq 0.9S_u$	Note 4
Austenitic Steel Components (e.g., Shell)		
Level D Elastic Analysis [NB-3225, App. F]	$P_m \leq$ min $(2.4S_m, 0.7S_u)$ P_m (or P_L) + P_b $\leq$ min $(3.6S_m, 1.0 S_u)$	Note 5
Level D Plastic Analysis [NB-3225, App. F]	$P_m \leq$ max $(0.7S_u, S_y + (S_u\text{-}S_y)/3)$ P_m (or P_L) + P_b $\leq 0.9S_u$	Note 5

Notes:
1. This limit may be exceeded provided the criteria of NB-3228.5 are satisfied.
2. There are no specific limits on primary stresses for Level A events. However, the stresses due to primary loads during normal service must be computed and combined with the effects of other loadings in satisfying other limits. See NB-3222.1.
3. The 10% increase in allowables from NB-3223(a) may be applicable for load combinations for which the pressure exceeds the design pressure.
4. Criteria listed are for carbon steel components (e.g., shield plugs).
5. Criteria listed are for austenitic parts including shells, cover plates, and the grapple assembly.

Table 3-3
Summary of Stress Criteria for Subsection NG Components

Loadings	Stress Category (5)	Notes
Design [NG-3221]	$P_m \leq 1.0S_m$ $P_m + P_b \leq 1.5S_m$	
Level A [NG-3222]	$P_m \leq 1.0S_m$ (Note 6) $P_m + P_b \leq 1.5S_m$ (Note 6) $P_m + P_b + Q \leq 3.0S_m$ (Note 4)	
Level B [NG-3223](1)	$P_m \leq 1.0S_m$ (Note 6) $P_m + P_b \leq 1.5S_m$ (Note 6) $P_m + P_b + Q \leq 3.0S_m$ (Note 4)	Note 1
Level C Elastic Analysis [NG-3224]	$P_m \leq 1.5S_m$ $P_m + P_b \leq 2.25S_m$	Notes 2 & 3
Level D Elastic Analysis [NG-3225, App. F]	$P_m \leq \min(2.4S_m, 0.7S_u)$ $P_m + P_b \leq \min(3.6S_m, 1.0S_u)$	
Level D Plastic Analysis [NG-3225, App. F]	$P_m \leq \max(0.7S_u, S_y + (S_u - S_y)/3)$ $P_m + P_b \leq 0.9S_u$	

Notes:
1. There are no pressure loads on the basket, therefore the 10% increase permitted by NG-3223(a) for pressures exceeding the design pressure are not included.
2. Evaluation of secondary stresses not required for Level C and D events.
3. Criteria listed are for elastic analyses, other analysis methods permitted by NG-3224.1 are acceptable if performed in accordance with the appropriate paragraph of NG-3224.1.
4. This limit may be exceeded provided the requirements of NG-3228.3 are satisfied, see NG-3222.2 and NG-3228.3.
5. As appropriate, the special stress limits of NG-3227 should be applied.
6. In accordance with NG-3222 and Note 9 of Figure NG-3221-1, the Limit Analysis provisions of NG-3228 may be used.
7. The weld strength of each fusion weld nugget shall have a minimum capacity of 16.0 kips (70°F). The minimum capacity shall be determined by shear tests using test specimens made from production materials.

Table 3-4
Summary of Stress Criteria for Subsection NC Components
(OS187H Transfer Cask)

Service Level	Stress Category	Stress Criteria
A (Normal Conditions)	Primary Membrane Stress, P_m	S_m
	Primary Membrane + Bending Stress, $P_m + P_b$	1.5 S_m
	Primary + Secondary Stress, $P_m + P_b + Q$	3 S_m
D (Accident Conditions)	Primary Membrane Stress, P_m	Lesser of 2.4 S_m or 0.7 S_u
	Primary Local Membrane Stress, P_L	150% of P_m Stress Limit
	Primary Membrane + Bending Stress, $P_m + P_b$	Lesser of 3.6 S_m or S_u

Table 3-5
SA-240 Type 304/SA-182 F304 Temperature Dependent Material Properties

Temp. °F	Ultimate S_u (ksi)	Yield S_y (ksi)	Allow. S_m (ksi)	E (10^6 psi)	α (10^{-6} °F^{-1})	Density, ρ (lb./in.^{3})*	Poisson, ν (10^{-6})*
70	75.0	30.0	20.0	28.3	8.5	0.29	0.3
200	71.0	25.0	20.0	27.6	8.9	0.29	0.3
300	66.2	22.4	20.0	27.0	9.2	0.29	0.3
400	64.0	20.7	18.7	26.5	9.5	0.29	0.3
500	63.4	19.4	17.5	25.8	9.7	0.29	0.3
600	63.4	18.4	16.4	25.3	9.8	0.29	0.3
700	63.4	17.6	16.0	24.8	10.0	0.29	0.3

*Material Properties taken from Reference 23

Table 3-6
HSM-H Concrete Temperature Dependent Material Properties

Material	Temperature (°F)	28 Day Compressive Strength, f_c (ksi)	Modulus of Elasticity, E (1.0E3 ksi)	Coefficient of Thermal Expansion, α ($x10^{-6}$ °F^{-1})
Concrete Normal Weight 5000 psi Strength	100	5.0	4.0	5.5
	200	5.0	3.6	5.5
	300	4.8	3.3	5.5
	400	4.5	3.0	5.5
	500	4.5	2.9	5.5

Table 3-7
HSM-H Reinforcing Steel Properties at Temperature

Material	Temperature °F	Yield Strength (ksi)	Modulus of Elasticity E (1.0 E3 ksi)
Reinforcing Steel, ASTM A615 Grade 60 [(1)]	100	60.0	29.0
	200	57.0	28.4
	300	54.0	27.8
	400	51.0	27.3
	500	51.0	27.0

Note
(1) Reinforcing steel data obtained from Handbook of Concrete Engineering [36].

Table 3-7A
Material Data for ASTM A-992 Steel

Material	Temperature (°F)	Yield Strength, f_y (ksi)	Modulus of Elasticity, $E^{(1)}$ (ksi)	$\alpha_{AVG}^{(1)}$ ($1\times10^{-6}\ °F^{-1}$)
ASTM A-992 Steel	70	50.0	29,500	--
	100	50.0	--	5.53
	200	45.6	28,800	5.89
	400	42.8	27,700	6.61
	500	40.4	27,300	6.91
	600	36.9	26,700	7.17
	700	36.0	25,500	7.41

(1) E and α are assumed to be same as that of ASTM A36 steel. Yield strength f_y of ASTM A-992 material is assumed to vary with temperature in same proportion as A-36 steel.

Table 3-7B

Material Data for ASTM A-36 Steel

Material	Temperature (°F)	Stress Properties (ksi)			Elastic Modulus (x1.0E3 ksi) (E)	Average Coefficient of Thermal Expansion(1) (x10^{-6} in./in.-°F)
		Stress Intensity (S_m)	Yield Strength (S_y)	Ultimate Strength (S_u)		
Carbon Steel ASTM A-36	70	-	36.0	58.0	29.5	--
	100	19.3	36.0	58.0	-	5.53
	200	19.3	32.8	58.0	28.8	5.89
	400	19.3	30.8	58.0	27.7	6.61
	500	19.3	29.1	58.0	27.3	6.91
	600	17.7	26.6	58.0	26.7	7.17
	700	17.3	25.9	58.0	25.5	7.41

Table 3-8
SA-240 Type XM-19 Temperature Dependent Material Properties

Temp. °F	Ultimate S_u (ksi)	Yield S_y (ksi)	Allow. S_m (ksi)	E (10^6 psi)	α (10^{-6} °F^{-1})	Density, ρ (lb./in.^{3})*	Poisson, ν (10^{-6})*
70	100.0	55.0	33.3	28.3	8.2	0.29	0.3
200	99.4	47.1	33.2	27.6	8.5	0.29	0.3
300	94.2	43.3	31.4	27.0	8.8	0.29	0.3
400	91.1	40.7	30.2	26.5	8.9	0.29	0.3
500	89.1	38.8	29.7	25.8	9.1	0.29	0.3
600	87.7	37.4	29.2	25.3	9.2	0.29	0.3

*Material Properties taken from Reference 23

Table 3-9
SA-540 Grade B24 Class 1 Temperature Dependent Material Properties

Temp. °F	Ultimate S_u (ksi)	Yield S_y (ksi)	Allow. S_m (ksi)	E (10^6 psi)	α (10^{-6} °F^{-1})	Density, ρ (lb./in.^{3})*	Poisson, ν (10^{-6})*
70	165.0	150.0	50.0	27.8	6.4	0.29	0.3
200	165.0	143.4	47.8	27.1	6.7	0.29	0.3
300	165.0	138.6	46.2	26.7	6.9	0.29	0.3
400	165.0	134.4	44.8	26.1	7.1	0.29	0.3
500	165.0	130.2	43.4	25.7	7.3	0.29	0.3
600	165.0	124.2	41.4	25.2	7.4	0.29	0.3

*Material Properties taken from Reference 23

Table 3-10
ASTM B-29, Chemical Lead Temperature Dependent Material Properties

Temp. °F	E (10^6 psi)	α (10^{-6} °F^{-1})	Density, ρ (lb./in.^{3})*	Poisson, ν (10^{-6})*
70	2.35	16.21	0.41	0.45
200	2.28	16.70	0.41	0.45
300	2.06	17.34	0.41	0.45
400	1.92**	18.12	0.41	0.45
500	1.78**	18.90	0.41	0.45
600	1.64**	19.68	0.41	0.45

*Material Properties taken from Reference 23

**Extrapolated from available Reference 22 Data.

Table 3-11
***Vyal B* Resin Material Properties**

Temperature	Modulus of Elasticity, *E* (psi)*	Coefficient of Thermal Expansion, α (in./in.°F)	Density, ρ (lb./in.^{3})*	Poisson, ν (10^{-6})*
Room Temperature	0.16×10^6	**	0.065	0.20

* Material Properties taken from Reference 11

** The coefficient of thermal expansion for the resin material is not used in the transfer cask structural analysis. The resin material is not a structural component, and since the resin has a very low Modulus of Elasticity (relative to stainless steel) it's thermal expansion is not expected to affect the stresses in the structural components significantly.

Table 3-12
Maximum axial stresses in the cladding during 75g side drop

Fuel Assembly Type	WE15x15	WE 17x17std	17x17 MkBW	WE 17x17 Vantage5H	WE 17x17OFA
Fuel Rod OD, D (in) (1) (13)	0.4193	0.3713	0.3713	0.3713	0.3573
Clad Thickness, t (in) (1) (13)	0.0216	0.0198	0.0213	0.0198	0.0198
Average Radius, R (in) (2)	0.1989	0.1758	0.1750	0.1758	0.1688
Fuel Pallet OD, D_P (in) (1)	0.3659	0.3225	0.3195	0.3225	0.3088
Fuel Tube M.I., I_F (in^4) (3)	5.35E-04	3.39E-04	3.60E-04	3.39E-04	3.00E-04
Fuel Pellet M.I., I_P (in^4) (4)	8.80E-04	5.31E-04	5.12E-04	5.31E-04	4.46E-04
I_{Total} (in^4) (5)	1.42E-03	8.70E-04	8.71E-04	8.70E-04	7.46E-04
Span Length, S (in) (12)	27.0	25.0	25.0	25.0	25.0
Fuel Assembly Weight, W_F (lb) (1)	1,555	1,575	1,575	1,575	1,575
No.of Rods, N (1)	204	264	264	264	264
Active Fuel Length, L (in) (1)	144.0	144.0	144.0	144.0	144.0
30-Foot Side Drop Equivalent g load	75	75	75	75	75
w_s (lb/in) (11)	3.97	3.11	3.11	3.11	3.11
Moment, M (kip.in) (6)	0.31	0.21	0.21	0.21	0.21
Bending Stress, S_b (psi) (7)	45,368	43,858	43,771	43,858	49,185
Internal Pressure, P (psi) (8)	2,235	2,235	2,235	2,235	2,235
Pressure Axial Stress, $S_{press.}$(psi) (9)	10,289	9,921	9,183	9,921	9,525
$S=S_b+S_{press.}$ (psi)	55,657	53,778	52,954	53,778	58,710
Allowable Stress, $S_{all}=S_y$ (psi) (10)	80,500	80,500	80,500	80,500	80,500

Note:

1. Input data from Chapter 2, Table 2 -1
2. $R = (D-t)/2$
3. $I_F = \pi / 64 \times (D^4 - (D - 2t)^4)$
4. $I_P = \pi / 64 \times D_P^4$
5. $I_{Total} = I_F + I_P$
6. $M = 0.1058\, w_s\, S^2$
7. $S_b = M \times (D / 2) / I_{Total}$
8. Input data from Reference 6
9. $S_{Press.} = (P \times R) / (2 \times t)$
10. Yield strength for Zircaloy cladding tube at 750 °F.
11. $w_s = (W_F \times 75g) / N / L$
12. Input data from Reference 5
13. Reduction of wall thickness by 0.0027 inch

Table 3-13
Fuel Cladding Buckling Load and Compressive Stresses during 75g End Drop

Fuel Type	Applied g load at the ANSYS last converged solution	Maximum stress intensity in cladding under 75g load	Yield Strength Cladding (Zircaloy)
WE15x15	120g	1,865 psi	80,500 psi
WE17x17OFA	120g	2,099 psi	80,500 psi

Table 3-14
Summary of OS187H Transfer Cask Top Cover Bolt Stress Analysis

Stress Type	Normal Condition		Accident Condition	
	Stress	Allowable	Stress	Allowable
Average Tensile (ksi.)	37.4	92.4	106.6	115.5
Shear (ksi)	6.8	55.4	6.8	69.3
Combined (ksi)	74.0	124.7	Not Required [1]	
Interaction Equation $R_t^2 + R_s^2 < 1$	0.179	1	0.862	1
Bearing (ksi) Allowable (ksi) (S_y of lid material)	11.5	43.3	Not Required [1]	

Table 3-15
Summary of OS187H Transfer Cask RAM Access Cover Bolt Stress Analysis

Stress Type	Normal Condition		Accident Condition	
	Stress	Allowable	Stress	Allowable
Average Tensile (ksi.)	45.2	92.4	45.2	115.5
Shear (ksi)	8.0	55.4	9.5	69.3
Combined (ksi)	97.0	124.7	Not Required [1]	
Interaction E.Q. $R_t^2 + R_s^2 < 1$	0.548	1	0.282	1
Bearing (ksi) Allowable (ksi) (S_y of lid material)	10.1	22.4	Not Required [1]	

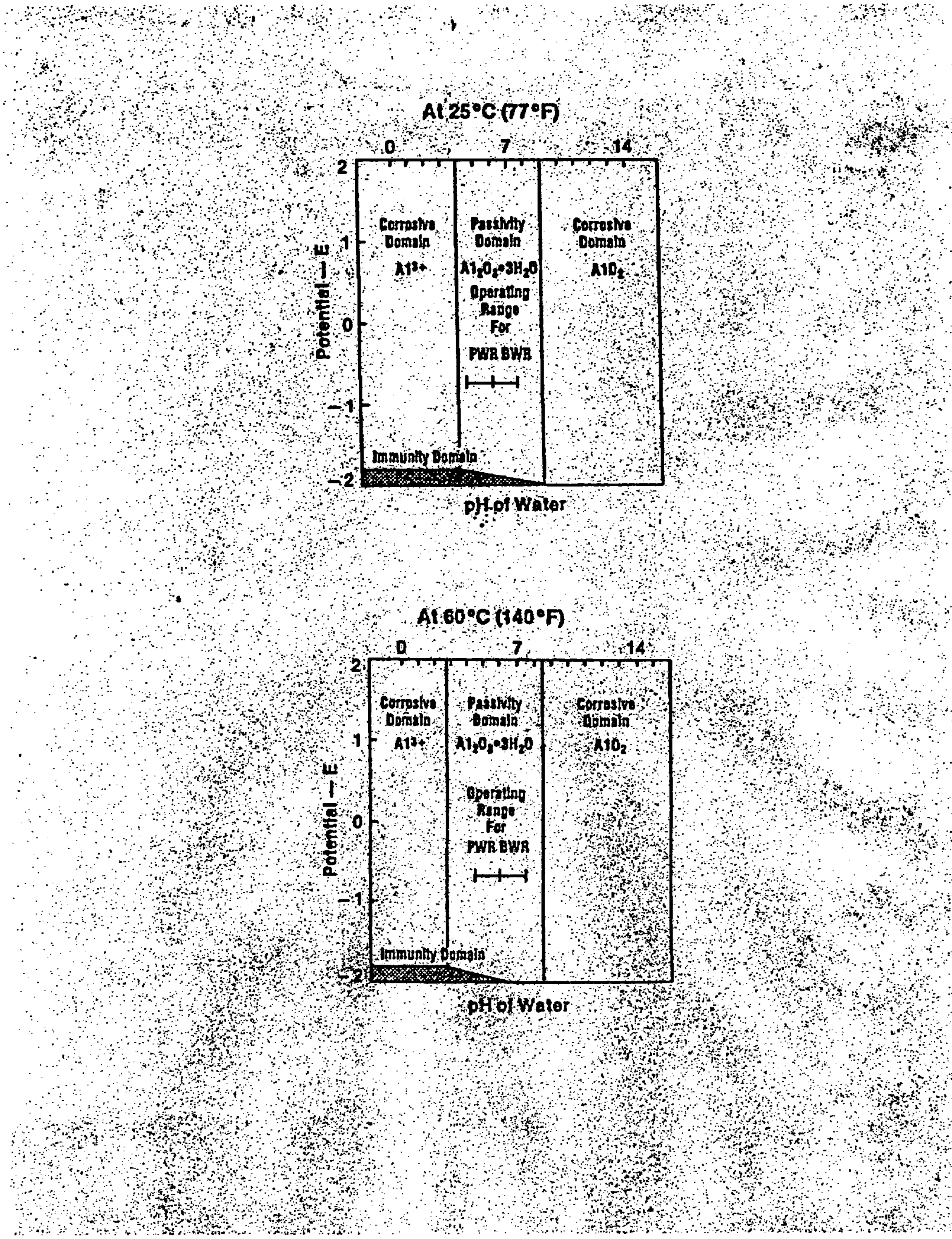

Figure 3-1
Potential Versus pH Diagram for Aluminum-Water System

APPENDIX 3.9.1
32PTH DSC (CANISTER AND BASKET) STRUCTURAL ANALYSIS

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3.9.1 32PTH DSC (CANISTER AND BASKET) STRUCTURAL ANALYSIS

3.9.1.1 Introduction

Each NUHOMS®-32PTH DSC consists of a fuel basket and a canister body (cylindrical shell, outer top cover plate, top shield plug/inner cover, and shell bottom or shell bottom assembly). The 32PTH DSC canister body is provided with four alternate shell bottom assembly configurations. The primary confinement boundary for the 32PTH DSC consists of DSC shell, the top shield plug/inner cover plate, and shell bottom or inner bottom cover plate of the shell bottom assembly (see Figure 1-5).

The Canister shell thickness is 0.50 inches, and the top and bottom closure assemblies are 12.0 inches and 8.75 inches, respectively. The Canister is constructed entirely from SA-240 Type 304 stainless steel and SA-182 Type F304. There are no penetrations through the confinement vessel. The draining and venting systems are covered by the port plugs and the outer cover plate and the top shield plug are welded to the cylindrical shell with multi-layer welds. The canister cavity is pressurized above atmospheric pressure with helium. The 32PTH DSC shell assembly geometry and the materials used for its analysis and fabrication are shown on drawings 10492-72-1 to 12 included in Chapter 1.

The basket structure consists of assemblies of stainless steel fuel compartments and support rails. The borated aluminum or boron carbide/aluminum metal matrix composite plates (neutron poison plates) provide the necessary criticality control and also provide the heat conduction paths from the fuel assemblies to the cask cavity wall. This method of construction forms a very strong structure of compartment assemblies which provide for storage of 32 PWR fuel assemblies. The open dimension of each fuel compartment is 8.70 in. × 8.70 in., which provides clearance around the fuel assemblies.

The Fuel Basket and Canister are analyzed separately. The Fuel Basket is analyzed in Section 3.9.1.2, while the Canister is analyzed in Section 3.9.1.3. Three separate finite element models are constructed for the structural evaluation of the fuel basket, while four finite element models are used for the structural evaluation of the canister shell.

A 3-dimensional cross-section finite element model is utilized to evaluate the effect of transverse inertial loads on both the basket and canister radial shell. A 3-dimensional model of a fuel compartment section is used to perform a buckling evaluation for the basket during lateral impact loads. Three-dimensional models of support rails are used to perform both thermal stress and buckling analyses.

A 2-dimensional axisymmetric model of the DSC canister shell is used to evaluate axial inertial loads as well as internal, external and thermal loads. Two 3-dimensional finite element models of the DSC canister bottom cover assembly and top cover assembly are utilized to evaluate the effects of transverse inertial loads on them. Also, a finite element model of the alternate canister

composite bottom cover assembly is constructed to evaluate the structural adequacy of this alternate design.

3.9.1.2 32PTH DSC Fuel Basket Structural Evaluation

3.9.1.2.1 Approach

The Fuel Basket is evaluated for normal and accident condition impact and thermal loads. The basket stress analysis is performed using a finite element method for the side drop and thermal load cases and classical hand calculations for the end drop load cases. Buckling of the basket plates, when subjected to lateral impact loads, is evaluated by collapse load analysis using a finite element model to generate a relationship between displacement and applied load. A summary of the basket load cases is provided in Section 3.9.1.2.2. Stress and buckling analyses are provided in Sections 3.9.1.2.3 and 3.9.1.2.4, respectively.

A. Material Properties

The mechanical properties of structural materials used in the basket, rail and canister are shown in the Table 3.9.1-1 as a function of temperature. All structural components of the fuel basket and support rails are constructed from SA-240, Type 304 stainless steel, with properties taken from AMSE B&PV Code [1]. The yield and ultimate strengths of the structural steel, shown in Table 3.9.1-1, are the minimum values specified in the material specifications. In general, the temperatures chosen for the evaluation of material properties for each component of the DSC bound the maximum temperatures computed in Chapter 4.

B. Design Criteria

For normal conditions, the basis for the basket allowable stress is the ASME Code, Section III, Subsection NG [2]. The primary membrane stress intensity and membrane plus bending stress intensities are limited to S_m (S_m is the code allowable stress intensity) and 1.5 S_m, respectively, at any location in the basket for Level A (Normal Service) load combinations. The average shear stress is limited to 0.6 S_m.

The ASME Code provides a $3S_m$ limit on primary plus secondary stress intensity for Level A conditions. This limit is specified to prevent ratcheting and distortion of a structure under primary plus secondary loads.

For accident conditions, stresses are evaluated as short duration Level D conditions as per ASME B&PV Code, Section III, Appendix F [3]. When evaluating the stainless steel basket results from the elastic analysis, the general primary membrane stress intensity in, P_m, shall not exceed $2.4S_m$ or $0.7S_u$ and membrane plus bending stress intensity ($P_m + P_b$) is limited to the smaller of $3.6S_m$ or $1.0S_u$. The average primary shear stress is limited to the smaller of $0.42S_u$ or $2(0.6S_m)$.

When evaluating the results from the non-linear elastic-plastic analysis, the general primary membrane stress intensity, P_m, shall not exceed $0.7S_u$ and the maximum stress intensity at any location (P_l or $P_m + P_b$) shall not exceed $0.9S_u$. The average primary shear stress is limited to $0.42S_u$ or $2(0.6S_m)$.

The acceptability of a component, against buckling, shall be demonstrated by collapse load analysis per Sections F-1340 and II-1430 of Reference 3.

For fusion welds between the stainless steel plates and the stainless steel fuel compartment are qualified by testing. The required minimum tested capacity of the weld connection (each weld) shall be 16.5 kips (at room temperature). This value is based on a margin of safety (test to design) of 2.0, corrected for temperature difference and the maximum weld load of 6,897 lbs calculated from a 75g side drop (see page 3.9.1-16). This margin of safety, 2, is larger than the ASME Code-implied margin of safety for level D loads. The minimum capacity shall be determined by shear test (pull test) of individual specimen made from production material. In addition to the ASME Code requirements for weld qualification, as part of the weld qualification procedure, in order to verify proper machine setting and operation, a shear test (pull test) of test coupon from each welding machine will be performed prior to the start of each working shift.

The allowable stresses for both normal and accident conditions are summarized in the following table.

Loading Condition	Stress Category	Stress Criteria
Normal Conditions, Elastic Analysis	Membrane Stress, P_m	S_m
	Membrane + Bending Stress, $P_m + P_b$	$1.5\ S_m$
	Average Shear Stress	$0.6\ S_m$
	Primary + Secondary Stress, $P_m + P_b + Q$	$3\ S_m$
Accident Conditions, Elastic Analysis	Membrane Stress, P_m	min$\{2.4S_m$ or $0.7S_u\}$
	Membrane + Bending Stress, $P_m + P_b$	min$\{3.6S_m$ or $1.0S_u\}$
	Average Shear Stress	min$\{1.2S_m$ or $0.42S_u\}$
Accident Conditions, Elastic-Plastic Analysis	Membrane Stress, P_m	$0.7\ S_u$
	Membrane + Bending Stress, $P_m + P_b$	$0.9\ S_u$
	Average Shear Stress	min$\{1.2S_m$ or $0.42S_u\}$

3.9.1.2.2 Loading Conditions

The transfer and storage basket loads are summarized in the tables below.

Basket Transfer Loads in Transfer Cask

Loading	Basket Orientation	Service Level	Load	Enveloped Load for Analysis	Load Combination
Dead Weight (DW)	Vertical[1]	A	1g Down (Axial)	1g Down (Axial)	1g Down (Axial)
Dead Weight (DW)	Horizontal[2]	A	1g Down	2g Axial + 2g Trans. + 2 g Vertical	A = 2g Axial + 2g Trans. + 2 g Vertical A + Thermal (115°F amb.) A + Thermal (-20°F amb.)
Handling load in Transfer Cask	Horizontal[2]	A	DW +1g Axial DW + 1g Trans. DW +1g Vert. DW +0.5g Axial + 0.5g Trans. + 0.5 g Vert.		
Thermal	Vertical[1]	A	Vacuum Dry	Vacuum Dry	DW + Vacuum Dry
Thermal	Horizontal[2]	A	Thermal Stress (115°F amb.)	Thermal Stress (115°F amb.)	Thermal Stress (115°F amb.)
Thermal	Horizontal[2]	A	Thermal Stress (-20°F amb.)	Thermal Stress (-20°F amb.)	Thermal Stress (-20°F amb.)
Accident Side Drop	Horizontal	D	44g Transverse[3]	75g Transverse	75g Transverse
Accident End Drop	Vertical	D	49g Vertical[3]	75g Vertical	75g Vertical

(1) Canister supported at the bottom.
(2) Canister supported at transfer cask rails.
(3) Impact accelerations computed in Appendix 3.9.7.

Basket Storage Loads in HSM-H

Loading	Basket Orientation	Service Level	Load	Enveloped Load for analysis	Load Combinations
Dead Weight (DW)	Horizontal(1)	A	1g Down	1g Down	1g Down
Seismic Loads	Horizontal(1)	C(2)	0.43g Axial + 0.43g Trans. + 0.20g Vertical +DW	0.65g Axial + 0.65 Trans.+ 1.30g Vertical	0.65g Axial + 0.65 Trans.+ 1.30g Vertical 0.65g Axial + 0.65 Trans.+ 1.30g Vertical Thermal (-20° F amb.) 0.65g Axial + 0.65 Trans.+ 1.30g Vertical Thermal (115° F amb.)
Thermal	Horizontal	A	Thermal (-20° F ambient)	Thermal (-20° F ambient)	Thermal
Thermal	Horizontal	A	Thermal (115° F ambient)	Thermal (115° F ambient)	Thermal
Thermal	Horizontal	D(2)	Thermal (Blocked Vent)	Thermal (Blocked Vent)	1g Down + Thermal (Blocked Vent)

(1) Canister Supported at HSM-H Rails and axially restrained by seismic restraint devices.
(2) Level C and D loads are conservatively treated as Level A loads.

3.9.1.2.3 Fuel Basket Stress Analysis

A. 3D Cross Section Finite Element Model Description

A three-dimensional finite element model of the basket fuel compartments, peripheral rails and canister is constructed using ANSYS [10] SHELL43 elements. The overall finite element model of the basket, peripheral rails and canister is shown in Figure 3.9.1-1. For conservatism, the strength of aluminum plates in the basket was neglected by excluding these from the finite element model. However, their weight was accounted for by increasing the basket plates and peripheral rail material densities. Because of the large number of plates in the basket and large size of the basket, certain modeling approximations were necessary. In view of continuous support of fuel compartment tubes by the peripheral rails along the entire basket length during a side drop, only a 15.0" long slice of the basket and rail was modeled. At the two cut faces of the model, symmetry boundary conditions were applied (UZ = ROTX = ROTY=0). The fuel compartment tubes, structural plates, support rails, square bars, and canister shell are included in the model and are shown individually in Figures 3.9.1-2 through 3.9.1-6.

Radial gap elements (CONTAC 52) are used to simulate the interface between the basket peripheral rails and inner side of the canister and between canister outer radius and cask inner radius. Each gap element contains two nodes; one on each surface of the structure. The gap nodes specified at the inner side of cask are restrained in x, y and z directions. The gap size at each gap element is determined by the difference between the basket rails radius and the inside radius of the canister (rail outer radius = 34.25" and canister inner radius = 34.375" to give 0.125 inch mean radial gap at room temperature) and between the canister outer radius and the inside radius of the cask (canister outer radius = 34.875" and cask inner radius = 35.25" to give 0.375 inch mean gap). Radial gap (and link) elements are generated using a small ANSYS macro. Actual gap sizes for the gap element, at each radial location, were determined and input into the model as real constants using another small ANSYS macro. This macro accepts the drop orientation and model geometry as inputs and determines the circumferential position of each gap element. The macro then computes the appropriate real constants and applies to appropriate gap elements. At the operating temperatures, the initial gap sizes will be lower. Thus use of room temperature gap sizes is conservative. Figure 3.9.1-6 shows the locations of both sets of gap elements.

During drops on cask rails (180° side drop), the initial gaps between the canister and the cask are modified using the ANSYS macro. Two 3 inch wide and 0.12 inch thick rails are welded to the cask inside at 12° on both sides of vertical center line and another set of two rails are welded at 38° on both sides of the vertical center line. For the 180° side drop onto the rails, the initial gaps at the two inner rail locations are assumed closed. In-between these two rail locations, the initial gaps are set to 0.12 inches. On the other two rail locations, the gap statuses are initially set to open, and the gap sizes are generated by macro and decreased by the rail thickness.

The connections between the stainless steel fuel compartment tubes (with intermediate aluminum plates), between the tubes and stainless steel plates as well as between the tubes and rails are made with node couplings. The nodes of various plates are coupled together in the out-of-plane direction so that they will bend in unison under surface pressure or other lateral loading to simulate the through-the-thickness support provided by the aluminum plates. The fusion welds, connecting the fuel compartment and plates, are modeled by coupling nodes in all directions. The bolts connecting the peripheral rails with the plates are also modeled by coupling nodes (in x, y and z directions).

Material Nonlinearities

The basket fuel compartments, structural plates, peripheral rails and canister shell are constructed from SA-240, type 304 stainless steel. A bilinear stress-strain relationship, with kinematic hardening, was used for each component to simulate a correct nonlinear material behavior at the maximum operating temperature. The following elastic and inelastic material properties are used in the analysis:

Material Property	Basket Fuel Compartments, and Center Plates at 700° F	Peripheral Rails, Sq. Bars and Outer Plates at 550° F	Canister at 500° F
Modulus of Elasticity, E (psi)	24.8×10^6	25.6×10^6	25.8×10^6
Yield Strength, F_y (psi)	17,600	18,900	19,400
Tangent Modulus, E_t (psi)	5% of $E = 1.24 \times 10^6$	5% of $E = 1.28 \times 10^6$	5% of $E = 1.29 \times 10^6$

Gap Element Nonlinearities

Gap elements (CONTAC 52) are used to model the actual surface clearance between the basket rails and canister inside as well as between canister outside and cask inside. The gap elements introduce nonlinearities in the analysis depending whether they are open or closed. Initially, at the contact surface, gaps are closed. For the remaining periphery, the gaps are open. Actual gap size at each rail nodal location was computed using ANSYS macro for each basket orientation. The gap element spring constant, K_n, is calculated as:

$$K_n = f E h \ [10]$$

Where f is a factor usually between 0.01 and 100, E is the material modulus of elasticity (25.8×10^6 psi), and h and is a typical "target length" or typical element size [typical element length ≈ 1.2 in., typical element width ≈ 1.9 in typical target length $h = (1.2\times1.9)^{0.5} = 1.51$ in.]. Therefore,

$$K_n = 25.8\times10^6 \times 1.51 \times f = 0.39\times10^6 \text{ to } 3{,}900\times10^6 \text{ lb./in.}$$

Thus, there is very wide range for K_n value. The structure responded well for a spring constant value of 1.0×10^6 lb/in. and is used in previous similar analyses. Further, to help convergence, ANSYS elements LINK8 were inserted coincident with the CONTAC52 elements. To assure that these elements do not transfer a substantial load between the surfaces, a very low elastic modulus (E = 1000 psi), a small area (0.1 in^2) and zero density (to zero their inertial loading contribution to the structure) are used in the analysis.

B. Fuel Basket Stress Analysis for Transfer Loads

B.1. Dead Weight, 1g Down (Cask Vertical)

During 1g down loading, the fuel assemblies and fuel compartment are forced against the bottom of the canister/transfer cask. It is important to note that, for any vertical or near vertical loading, the fuel assemblies react directly against the bottom of the cask and not through the basket structure as in lateral loading. It is the dead weight of basket only that causes axial compressive stress during an end drop. Axial compressive stresses are computed as if only the compartment tubes will withstand all the weight. A conservative basket weight of 30.0 kips. (Actual weight is 29.854 kips) is used in the end drop stress calculations.

Total weight of the basket assembly = 30.0 kips

Compressive Stress in Fuel Compartments

Section area of compartment tubes = $32\times[9.075^2 - 8.7^2] = 213.3$ in^2

Area of 1 in × 1 in drain slots = 32 × 4 slots/compartment × [1.0" × 0.1875"] = 24.0 in^2
Stress due to 1g = -30.0 / (213.3 - 24.0) = - 0.1585 ksi

Shear Stress in Rail Stud

During vertical basket orientation, the rail will support its own weight. Therefore, there will be no load on rail studs.

B.2. Handling Loads (Cask Horizontal)

During normal conditions of transfer, the basket and canister shell is resting on two rails in OS187H Transfer Cask (3 in wide × 0.12 in thick) at 12° on either side of basket centerline. The radial contact elements between the pads and canister are assumed to be initially closed. The canister and rail nodes at one location are held in circumferential direction to avoid rigid-body motion of the model. In-between the two rail locations, initial gaps are set to 0.12 inches. On the other two rail locations (at 38° on either side of basket centerline); the macro generated gaps are decreased by rail thickness and are initially open.

Loading

A fuel assembly weight of 1,585 lb and 144 inch effective length are used in the analysis. A uniform fuel weight distribution is conservatively assumed over the 144 inches of basket length (the actual basket length is 162 inches).

The vertical and transverse loads, resulting from the fuel assembly weight, are applied as pressure on horizontal and vertical faces of plates (see Figure 3.9.1-7). The pressure for 2g acceleration is calculated below:

Pressure, P_V, on horizontal plates = P_h on vertical plates
= Fuel wt. × acceleration / (Panel span × Panel length)
= 1,585 lb. × 2.0 / (8.8874 in × 144 in) = 2.477 psi

The inertia load due to fuel compartments, rails and canister dead weight is simulated using the density and appropriate acceleration in vertical and transverse directions.

The steel panels and peripheral rail material density is modified to account for the aluminum plates, which are not modeled. The equivalent densities of these components are the following:

Fuel Compartments:	Equivalent Density, ρ = 0.424 lb/in^3
Large Peripheral Rails:	Equivalent Density, ρ = 0.50 lb/in^3
Small Peripheral Rails:	Equivalent Density, ρ = 0.488 lb/in^3

Since only a 15 inch length of the basket is modeled, the acceleration in axial direction is increased to account for the entire 144 inch length.

Axial acceleration for $2g$ load = $2.0g \times 144/15.0 = 19.2g$

To simulate the axial stress due to the above acceleration, only one side of basket is restrained in z-direction.

For ($2g$ axial + $2g$ Transverse + $2g$ Vertical) handling load, the pressures and accelerations are applied simultaneously:

Therefore the acceleration applied to the model is: acel,-2.0, 2.0, 19.2

Analysis and Results

A nonlinear elastic stress analysis was conducted for computing the elastic stresses in the basket model. The nonlinearity of analysis resulted from the gaps in the model. The total load was applied in small steps. The automatic time stepping program option "Autots" was activated. This option lets the program decide the actual size of the load-substep for a converged solution. The shell middle surface nodal stress intensity is the membrane stress intensity and top or bottom surface stress intensity is the membrane plus bending stress intensity.

Analysis of Fusion Welds for Handling Loads

The maximum fusion weld load was computed using the finite element model side drop load case (see page 3.9.1-16). The finite element model is modified by replacing the fusion weld couplings with PIPE20 elements.

A static nonlinear stress run is made and results of the run are post-processed in order to extract the axial (FX) and shear (FY and FZ) forces in the pipe elements. Reviewing the details of pipe element forces (at '*i*' node and '*j*' node of each pipe element) show that the axial (FX) and shear (FZ) loads are not significant. The maximum force in anyplace of the pipe elements (F_y) for the 2g handling load is 246 lb. The thermal loads in the fusion welds are calculated in Section B.3 below (page 3.9.1-14). The maximum combined shear force due to handling load and thermal load is 1.5 kips. The fusion weld load capacity is qualified for 75g accident load by test and is 16.5 kips. The transfer loads (2g) are much smaller than the 75g load. Thus by comparison, fusion welds capacity is judged to be much higher than the combined handling and thermal loads.

B.3. Thermal Loads during Transfer

Generally, thermal stresses develop in the basket if its free thermal expansion is constrained by the peripheral rails or canister. The thermal expansion calculations in Section 3.9.1.4.4 show that the basket rails are free to grow due to the maximum operating temperature in the canister. The rails are attached to the basket with bolts in slotted holes. Thus rails also permit free thermal growth of basket boxes. Aluminum and poison plates are sandwiched by the compartment tubes. A thermal expansion gaps are provided to allow the aluminum/poison plates free to grow in the axial direction while oversize slots are provided to allow aluminum/poison plates free to grow in the radial direction (see TN drawing 10494-72-8). However, some thermal stresses in basket and rails can develop due to radial gradients (hot at center and cooler at periphery) for normal thermal conditions. Basket and Rail thermal stresses are therefore calculated for the 115° F (hot normal), -20° F (cold normal) ambient and vacuum drying process.

Thermal Stresses in Basket Fuel Compartments during Transfer

A three-dimensional finite element model of the basket is used for thermal and stress analyses of the basket, using ANSYS. This finite element model is described in Section 3.9.1.2.3A. Due to symmetry of temperature distribution, only a ¼ model (see Figure 3.9.1-8) is used in this analysis. The rails and canister are removed from the model, as they have no effect on the fuel compartment thermal stresses. The support rails are analyzed separately for the thermal loads.

In order to model realistic contact between the fuel compartments, the couplings are replaced by contact elements. The couplings at the fusion weld locations are replaced by pipe elements.

Two finite element analyses are required to compute the thermal stresses in the fuel basket. The first analysis is a thermal analysis that computes the temperature distribution at each node of the structural model, given the temperature distribution in the thermal model described in Chapter 4. The second finite element analysis computes the thermal stress distribution caused by the temperature distribution computed in the first analysis.

The four-node element SHELL57 (Thermal Shell) and LINK33 (Thermal Conduction Bar) are used in the thermal analysis. These elements are replaced by stress elements SHELL43 and PIPE20 in the stress analysis.

Thermal Analyses

Thermal analyses of a gross model of 32PTH DSC Canister, Basket, and OS187H Cask is conducted for hot and cold normal ambient conditions and for vacuum dry procedures in Chapter 4. Steady-state thermal analyses of the basket structural model are conducted to obtain the nodal temperatures by impressing the temperatures computed in the Chapter 4 analyses as the boundary conditions for 115° F, -20° F ambient and vacuum drying cases.

In Chapter 4, the basket and rail temperatures are computed for three separate vacuum drying procedures. The table below provides the maximum basket and rail temperatures and thermal gradients for each vacuum drying procedure.

	Procedure A At 34 hours, Max. Temp (°F)	Procedure B At 32 hours, Max. Temp (°F)	Procedure C At 40 hours, Max. Temp (°F)
Basket Fuel Compartments	697	693	704
Basket Rails	531	575	579
Thermal Gradient, ΔT	166	118	125

From the above table, it is judged that 'Procedure A' case will be critical for stresses due to the highest thermal gradient and is selected for the analysis.

Thermal material properties for material (type 304 stainless steel), taken from Reference 1, are reproduced in Table 3.9.1-2.

The thermal analysis resulting temperature distributions for -20° F and 115° F ambient and vacuum drying conditions closely match the temperature distributions presented in Chapter 4.

Thermal Stress Analysis

Elastic stress analyses of the basket structure are conducted in order to compute the thermal stresses. The nodal temperature distribution from the thermal analysis results is applied to obtain the thermal stresses in the model. The resulting nodal stress intensity distribution in the basket fuel compartments reveals that the maximum thermal stress occurs during the vacuum drying load case, and is 9,566 psi.

Thermal Stresses in Support Rails during Transfer

The temperature distribution and the thermal stresses in peripheral rails are computed using the same methodology as given above for the fuel compartments. The finite element model of the rails is taken from the full basket model described in Section 3.9.1.2.3.A and is shown in Figure 3.9.1-9.

The resulting nodal stress intensity distribution in the support rails reveals that the maximum thermal stress occurs during the vacuum drying load case, and is 15,073 psi.

Thermal Load in Fusion Welds during Transfer

The forces in *x*, *y* and *z* global directions in the PIPE elements (modeling the fusion welds) for the critical vacuum dry thermal load case are tabulated for stress computation. The thermal analysis results show that the thermal loads in the fusion welds are quite small. Conservatively summing the maximum absolute values of F_x, F_y and F_z yields a combined load, F_{weld} of,

$$\underline{F_{weld}} = 294 + 505 + 454 = 1{,}253 \text{ lbs.}$$

This force is combined with the force generated from 2g handling load calculated above, therefore, the total load is:

1253 lb + 246 lb = 1,499 lb ≈ 1.5 kips

This load is much smaller compared to the weld capacity of 16.5 kips from test.

B.4. Summary of Fuel Basket Stresses for Normal Condition Transfer Loads

Table 3.9.1-3 summarizes basket stress analysis results and compares them with the Code allowable stresses. For the Normal thermal condition allowable stresses, the fuel compartment temperature is taken to be 700° F uniform, the peripheral rail temperature is taken to be 600° F uniform and canister temperature is taken to be 500° F uniform. Based on the results of these analyses, the basket and rails are structurally adequate for normal transfer condition loads.

B.5. 75g Hypothetical Accident Condition Side Drop during Transfer

In this section, stresses are evaluated in 32PTH DSC Basket for 75*g* transfer accident condition side and end drop impact loads.

The basket temperature is taken as 700° F, uniform. The rail temperature is taken as 550° F, uniform, and the canister temperature is taken as 500° F uniform. These temperatures conservatively bound the temperature distributions computed in Chapter 4.

The 3-dimensional finite element model described in Section 3.9.1.2.3.A is also used to compute the stresses due to the 75*g* accident drop.

Loadings

The basket structure is analyzed for 75g side drops in 0°, 30°, 45°, and 180° (on rails) orientations. Due to the basket structure symmetry, these orientations are assumed to bound all possible maximum stress cases.

The load resulting from the fuel assembly weight is applied as pressure on the fuel compartment plates of the basket. For the 180° (drop on support rails) and 0° orientations, the pressure act only on the horizontal plates while for all other orientations, the pressure is divided into components to act on both horizontal and vertical plates of the basket. The pressures for the different orientations are calculated below for 1*g* and 75*g* accelerations:

0° and 180° drops

Pressure, p for $1g$ = Fuel assembly weight / (Panel span × Panel length)
= 1585 lb./ (8.8874 in × 144 in) = 1.238489 psi.

Pressure, p for $75g$ = 75 × 1.238489 = 92.8867 psi.

30° drop

p_h on vertical plates for $1g = p \sin 30° = 1.238489 \times 0.5 = 0.619244$ psi.

p_v on horizontal plates for $1g = p \cos 30° = 1.238489 \times 0.86603 = 1.072563$ psi

p_h on vertical plates for 75g = 75 × 0.619244 = 46.4433 psi

p_v on horizontal plates for 75g = 75 × 1.072563 = 80.4422 psi

45° drop

p_v on horizontal plates $= p \cos 45° = 1.238489 \times 0.7071 = 0.875736$ psi

p_h on vertical plates $= p \sin 45° = 1.238489 \times 0.7071 = 0.875736$ psi

$p_v = p_h$ for $75g = 75 \times 0.875736 = 65.680$ psi

The inertia loads due to the basket and peripheral rail dead weights are simulated by applying the density and appropriate acceleration in the runs.

The aluminum plate weight is accounted for by increasing the densities of stainless steel basket fuel compartments, large rails and small rails, as in Section 3.9.1.2.3.B.2.

Finite Element Analysis

A nonlinear static stress analysis of the structural basket is conducted for computing the stresses for 0°, 30°, 45° and 180° (drop rails) drop orientations. The maximum load of 75*g* was applied in each analysis. The automatic time stepping program option "Autots" was activated. This option lets the program decide the actual size of the load-substep for a converged solution. Displacements, Stresses and Forces at each node of model (for each converged substep load) were written in ANSYS result files. The program stops at the load substep when it fails to result in a converged solution. In all side drop cases, the program gave converged solutions up to 75*g* load. Results were extracted at the last load sub-step of 75*g* for evaluation.

Shear Load in Fusion Welds due to 75g Side Drop

The maximum fusion weld load was computed using the finite element model. The finite element model is modified by replacing the fusion weld couplings with PIPE20 elements.

A static nonlinear stress run is made and results of the run are post-processed in order to extract the axial (FX) and shear (FY and FZ) forces in the pipe elements. Reviewing the details of pipe element forces (at '*i*' node and '*j*' node of each pipe element) show that the axial (FX) and shear (FZ) loads are not significant.

Conservatively, the maximum shear load in a fusion weld is computed by vectorially adding the maximum FY and maximum FZ (irrespective of their locations in the finite element model) as follows.

Maximum Force, FY = 6,893 lb.
Maximum Force, FZ = 228 lb.
Maximum Shear Force = $[6,893^2 + 228^2]^{1/2}$ = 6,897 lb.

From the above, it is seen that the maximum shear load on a fusion weld is 6,897 lb. The fusion weld capacity (by test) is to match or exceed this maximum weld load.

For the fusion weld load capacity test at room temperature, it is determined to include a safety factor of 2 and a correction for material strength for room temperature testing. Therefore, the Required Minimum Fusion Weld Test Load = 6,897 x (2) x (F_{tu} at room temperature / F_{tu} at 700°F) = 6,897 (2.0) (75.0 ksi/63.4 ksi) = 16,318 lbs ≈ 16.5 kips

Square Bar Stresses due to 75g Side Drop

Four 7/8 inch square bars are welded to the small rails. These square bars are modeled with SOLID45 Elements in finite element model shown in Figure 3.9.1-5.

From the nodal stress intensity distribution in the bars for 0°, 30°, 45° and 180° (on rails) side drops, it is seen that the maximum nodal stresses are local and secondary (surrounded by a low stress area) in nature and can be neglected in the accident stress evaluation. However, for conservatism, stresses are considered primary and are linearized at the highest nodal stress intensity locations to compute P_m and $P_m + P_b$.

Summary of Side Drop Results

The Table 3.9.1-4(a) and Table 3.9.1-4(b) summarizes the accident condition basket structural analysis maximum stress intensities for the various side drops and compares them with the material code allowable stresses at 700° F for the basket, 550° F for the support rails and 500° F for the canister shell. The shell element middle surface nodal stress intensity is the membrane stress intensity and the maximum of the top or bottom surface stress intensity is the membrane plus bending stress intensity. All stresses in the basket components are below the allowables.

Rail Stud Stresses due to 75g Side Drop

It is observed from the stress summary table (Table 3.9.1-4(a)) that the maximum rail stresses occurred during the 180° side drop (drop on rails). Stresses in other basket components are also quite high during this drop orientation. In other side drop orientations stresses were somewhat lower. Therefore, the maximum shear stresses in the rail studs are expected to occur during a 180° side drop. The rail stud stresses are therefore computed for this side drop. These stresses will bound the stud stresses for the other basket drop orientations.

For the small rails, due to the enlarged holes in rails mounting spacers, the top small rail will slide and will be supported by the fuel compartment. The bottom small rail will also slide and be supported by the canister and in term supported by the transfer cask. There are no loads to be carried by the studs. Therefore stress evaluations of the small rail studs are not critical.

For the large rail (A180) studs, the rails also will slide during the drop and will be supported by the canister and in term will be supported by the transfer cask. However, for conservatism, the shear stresses are considered in the rail plate/stud weld (O.D. = 0.75 in, I.D. = 0.75 – (2 × 0.31) = 0.13 in) by assuming all the weight of the rail will be carried by the studs. There are 11 stud rows over the entire basket length.

Weight of 8 half-rails = 10378 lb. (Section 3.2.1)
Weight of one half-rail = 10378/8 = 1297.3 lb

Number of Studs supporting the half-rail = 11 rows × 2 studs/row = 22
Weld area of one stud, $A = \pi/4\ (0.75^2 - 0.13^2) = 0.4285\ in^2$

Weld Shear Stress for $75g = (1297.3 \times 75) / (22 \times 0.4285) = 10{,}321$ psi ≈ 10.3 ksi

Allowable Shear Stress (at 550° F) $= 1.2\ S_m = 1.2 \times 16.95 = 20.34$ ksi. > 10.3 ksi.

The design is adequate.

B.6. 75g Hypothetical Accident Condition End Drop during Transfer

During an end drop, the fuel assemblies and fuel compartments are forced against the bottom of the canister/transfer cask. It is important to note that, for any vertical or near vertical loading, the fuel assemblies react directly against the bottom or top end of the canister/transfer cask and not through the basket structure as in lateral loading. It is the dead weight of the basket only that causes axial compressive stress during an end drop. Axial compressive stresses are conservatively computed assuming that all loads act on the compartment tubes during an end drop. A conservative basket weight of 30.0 kips. (Actual weight is 29.85 kips from Section 3.2.1) is used in the end drop stress calculations.

Compressive Stress in Fuel Compartments due to 75g End Drop

Section area, A, of the fuel compartment tubes is,

$$A = 32\times [9.075^2 - 8.7^2] = 213.3 \text{ in}^2$$

Area of 1 × 1 inch drain slots, A_s, is,

$$A_s = 32 \times 4 \text{ slots/compartment} \times [1.0 \text{ in} \times 0.1875 \text{ in}] = 24.0 \text{ in}^2$$

Stress due to $1g$ load, σ_{1g}, is,

$$\sigma_{1g} = -30.0 / (213.3 - 24.0) = -0.1585 \text{ ksi}$$

At 75g the compressive stress, σ_{75g}, is,

$$\sigma_{75g} = -0.1585 \text{ ksi} \times 75 = -11.89 \text{ ksi}$$

Stress in Rail Stud due to 75g End Drop

During the 75g end drop, the rail will support its own weight by contact with the bottom or top of canister / transfer cask. Therefore, there will not be any stresses in rail studs during an end drop.

C. Fuel Basket Stress Analysis for Storage Loads

This section evaluates the elastic stresses in the 32PTH DSC fuel basket due to HSM-H storage loads including, dead weight, seismic and thermal loads. The basket stresses are then compared with the code allowable stresses.

During normal condition of storage, the fuel compartment temperature is taken to be 700° F, uniform, the rail temperature is taken as 550° F, uniform, and the canister temperature is taken as 500° F, uniform. These temperatures are considered conservative based on the thermal evaluation presented in Chapter 4.

During the blocked vent hypothetical accident case, the fuel compartment and support rail temperatures are taken to be 800° F and 650°F, respectively. These temperatures are also conservatively based on the thermal evaluation in Chapter 4.

C.1. Dead Weight Analysis during Storage

The three-dimensional finite element model of the basket fuel compartments, rails and canister shell used to compute stresses for storage loads is described in Section 3.9.1.2.3.A. The finite element model, boundary conditions, and real constants are described in detail in that section.

The canister shell is resting on two support pads (3 inches wide) inside the HSM at 30° on either side of canister/basket centerline (canister/basket in 180° orientation). The radial contact elements at the pad locations are assumed closed. The canister shell nodes at one location coincident with the pad are held in circumferential direction to avoid rigid-body motion of the model. The gap elements between inside surface of the canister shell and basket rails are assumed to be closed at 180° (bottom) orientation and the remaining initial gaps were suitably modified (from 0.0 inch gaps at the bottom, to 0.25 inch gaps at the top) using an ANSYS macro. The node couplings, boundary conditions and loading are shown in Figure 3.9.1-10. For clarity, only the front view of the model is shown and rotational displacement boundary conditions are not shown in this figure.

Dead Weight Applied Load

Dead Weight Load (1g) is applied as pressure on the horizontal faces of fuel compartment plates (see Figure 3.9.1-10). The pressures for the 1g acceleration are calculated below.

Pressure, p_v on horizontal plates is,

$$p_v = \text{Fuel assembly weight} \times \text{acceleration} \;/\; (\text{Panel span} \times \text{Panel length})$$
$$= 1{,}585 \text{ lb.} \times 1.0g \;/\; (8.8874 \text{ in} \times 144 \text{ in}) = 1.2385 \text{ psi}$$

The inertial load due to the basket rails and canister dead weight is simulated applying density and 1*g* acceleration in vertical direction. The aluminum plate weights are accounted for by increasing the basket fuel compartment plate and rail densities.

Dead Weight Finite Element Analysis

An ANSYS elastic nonlinear stress analysis is conducted to compute stresses in the basket and canister model. The nonlinearity of analysis results from the gaps elements in the model. The total load is applied in small steps. The automatic time stepping program option "Autots" is activated. This option lets the program decide the actual size of the load sub-step for a converged solution. Displacements, stresses and forces at the final load sub-step are written to ANSYS result files.

C.2. Seismic Load Analysis for Storage

An elastic static analysis of seismic loads is conducted using the finite element model described in Section 3.9.1.2.3.A (page 3.9.1-7). The lowest natural frequency of the fuel basket is computed by modal finite element analyses to be 82.7 Hz. This shows that the basket structure is relatively rigid and therefore, the ground seismic loads will not be appreciably amplified. However, 0.65*g* axial + 0.65*g* transverse + 1.3*g* vertical loads are conservatively used in the static stress analysis. These loads translate into the following applied pressures in horizontal and vertical plates and accelerations.

The vertical and transverse loads, resulting from the fuel assembly weight, are applied as pressure on horizontal and vertical faces of plates. The pressures on the horizontal and vertical plates of the fuel compartments are calculated below as follows.

The pressure, p_v on the horizontal plates due to the vertical 1.3*g* acceleration is,

$$p_v = 1.3 \times (1.2385) = 1.610 \text{ psi}$$

The pressure, p_h on the vertical plates due to the transverse 0.65*g* acceleration is,

$$p_h = 0.65 \times (1.2385) = 0.805 \text{ psi}$$

The inertia load due to the fuel compartments, rails and canister dead weights are simulated using the density and appropriate acceleration in the vertical, transverse and axial directions. As described in Section 3.9.1.2.3.A., the steel panels and peripheral rail material density are modified to account for the aluminum plates, which are not modeled. Since only a 15 inch length of the basket is modeled, the acceleration in axial direction is increased to account for the entire 144 inch length.

The axial acceleration for the 0.65g load, a_{axial}, is,

$$a_{axial} = 0.65g \times 144/15.0 = 6.24g$$

To simulate the axial stress due to the above acceleration, only one side of the basket is restrained in z-direction. The above loads and boundary conditions are applied to the finite element model using a method similar to that described for the dead weight load case.

For (1.3g vertical + 0.65g transverse + 0.65g axial) seismic load, the calculated pressures and the following accelerations are applied simultaneously.

Acceleration applied to the model: 1.3g (vertical), -0.65g (transverse), and 6.24g (axial)

Seismic Load Finite Element Analysis

A nonlinear elastic stress analysis was conducted for computing the elastic stresses in the basket model. The nonlinearity of analysis results from the gap elements in the model. The total load is applied in small steps. Again, the automatic time stepping program option "Autots" is activated. The detailed displacements, stresses and forces for each computer run, at the final load sub-step, are written to ANSYS result files. The shell middle surface nodal stress intensity is the membrane stress intensity and top or bottom surface stress intensity is the membrane plus bending stress intensity.

C.3. Thermal Loads during Storage

For the thermal stress analysis, temperatures are assumed to be symmetric about a 90° basket model.

Generally, thermal stresses develop in the basket if its free thermal expansion is constrained by the peripheral rails or canister. The thermal expansion calculations in Section 3.9.1.4.4 (page 3.9.1-79) show that the basket rails are free to grow due to the maximum operating temperature in the canister. Also, the peripheral rails are attached to the fuel compartments with bolts in slotted holes. Thus rails also permit free thermal growth of the fuel compartments boxes. Aluminum and poison plates are sandwiched by the compartment tubes. A thermal expansion gaps are provided to allow the aluminum/poison plates free to grow in the axial direction while oversize slots are provided to allow aluminum/poison plates free to grow in the radial direction (see TN drawing 10494-72-8). However, some thermal stresses in fuel compartments and rails can develop due to radial gradients (hot at center and cooler at periphery) due to normal and accident condition thermal conditions. The fuel compartments and support rail thermal stresses are therefore calculated for the 115° F (hot environment), -20° F (cold environment) ambient and HSM blocked vent accident conditions.

Thermal Stresses in the Basket Fuel Compartments

A 3-dimensional finite element model of the basket is used for the thermal and stress analyses of the fuel basket. This finite element model is described in Section 3.9.1.2.3A. Due to symmetry of temperature distribution, only a ¼ model (see Figure 3.9.1-8) is used in this analysis. The rails and canister are removed from the model, as they have no effect on the fuel compartment thermal stresses. The support rails are analyzed separately.

In order to model realistic contact between the fuel compartments, the couplings are replaced by contact elements. The couplings at the fusion weld locations are replaced by pipe elements.

Two finite element analyses are required to compute the thermal stresses in the fuel basket. The first analysis is a thermal analysis that computes the temperature distribution at each node of the structural model, given the temperature distribution in the thermal model described in Chapter 4. The second finite element analysis computes the thermal stress distribution caused by the temperature distribution computed in the first analysis.

The four-node element SHELL57 (Thermal Shell) and LINK33 (Thermal Conduction Bar) are used in the thermal analysis. These elements are replaced by stress elements SHELL43 and PIPE20 in the stress analysis.

Thermal Analyses

Thermal analyses of the gross finite element model of the 32PTH DSC and fuel basket is conducted for both hot and cold normal ambient conditions as well as for the HSM blocked vent accident in Chapter 4. Steady-state thermal analyses of the detailed basket model, shown in Figure 3.9.1-8, are conducted to obtain the nodal temperatures by impressing the temperature distribution (computed in Chapter 4) as the boundary conditions for hot, cold and vent blockage cases. In Chapter 4, the fuel basket and support rail temperatures are computed for four normal condition cases. Below are given the maximum basket and rail temperatures and thermal gradients for each case.

	Case 1 115° F Ambient With Fins Max. Temp. (°F)	Case 2 115° F Ambient Without Fins Max. Temp. (°F)	Case 3 -20° F Ambient With Fins Max. Temp. (°F)	Case 4 -20° F Ambient Without Fins Max. Temp. (°F)
Basket Fuel Compartments	656	666	565	568
Basket Rails	511	523	418	422
Thermal Gradient, ΔT	145	143	147	146

From the above table, it is seen that the 'with fins' cases give higher thermal gradient for both hot and cold ambient conditions. These two normal condition cases are expected to result in higher stresses and are selected for the analyses.

Similarly the basket and rail temperatures are computed for four blocked vent condition cases. Below are the maximum basket and rail temperatures and thermal gradients for each case.

	Case 1 34 hours after blockage, With Fins Max. Temp.(°F)	**Case 2** 48 hours after blockage, With Fins Max. Temp.(°F)	**Case 3** 31 hours after blockage, Without Fins Max. Temp.(°F)	**Case 4** 48 hours after blockage, Without Fins Max. Temp. (°F)
Basket Fuel Compartments	749	774	750	780
Basket Rails	605	631	606	637
Thermal Gradient, ΔT	144	143	144	143

From the above table, it is seen that the 'with fins' cases give slightly higher thermal gradients but 'without fins' case give higher temperatures. Enveloping these conditions, thermal and stress runs are made for Case 1 (because of the higher ΔT), but material allowable stresses are taken at the higher temperatures from Case 4.

Thermal material properties for material (type 304 stainless steel), taken from Reference 1, are reproduced in Table 3.9.1-2.

The thermal analysis resulting temperature distributions for each thermal load case closely match the temperature distributions presented in Chapter 4.

Thermal Stress Analyses

Elastic stress analyses of the fuel compartment structure are conducted for computing the thermal stresses in the fuel basket. The nodal temperature distribution from the above thermal analysis results are applied to obtain the thermal stresses in the model.

The resulting displacements and stresses in the model are written to ANSYS result files. The critical stresses are summarized in Table 3.9.1-5. It is seen from Table 3.9.1-5, that the maximum thermal stress intensities in fuel compartments are developed in -20° F ambient case.

Thermal Stresses in Peripheral Rails during Storage

Temperature distribution and thermal stresses in peripheral rails are computed using the same methodology as given above for the fuel compartments. The finite element model of the support rails is extracted from the model described in Section 3.9.1.2.3.A (page 3.9.1-7) and is shown in Figure 3.9.1-9.

The resulting thermal stress intensities for the 115° F, -20° F, and blocked vent cases are summarized in Table 3.9.1-5.

Fusion Welds Evaluation for Thermal Storage Loads

The forces in the X, Y and Z global directions, in the PIPE elements (modeling the fusion welds), for the critical -20° F case are post processed from the ANSYS result files.

Review of the ANSYS results files reveal that the thermal loads are quite small. Even if the maximum absolute values of FX, FY and FZ are algebraically added,

Combined Load = 224 + 313 + 304 = 841 lbs.

This load is small.

The fusion weld load capacity, qualified by load test (for 75*g* horizontal drop accident) is 16.5 kips. The storage seismic and thermal loads are much smaller than the test load. Thus by comparison, fusion welds capacity is judged to be adequate for the storage loads.

C.4. Summary of Fuel Basket Stresses for Storage Loads

Table 3.9.1-5 summarizes the fuel basket stress analysis results and compares them with the code allowable stresses.

For the normal condition thermal load cases, the fuel compartment temperature is taken to be 700° F uniform, the peripheral rail temperature is taken to be 600° F uniform and the canister shell temperature is taken to be 500° F uniform. For the HSM vent blockage hypothetical accident condition the fuel compartment temperature is taken to be 800° F uniform and the peripheral rail temperature is taken to be 650° F.

Based on the results of these analyses, the basket and rails are structurally adequate for the normal and accident condition storage loads.

3.9.1.2.4 32PTH DSC Fuel Basket Buckling Analysis

This Section evaluates the 32PTH DSC basket design with respect to buckling. The basket fuel compartments and support rails are evaluated separately in the following sections.

A. Basket Fuel Compartment Buckling Analysis

A.1. Approach

Only the most critical fuel basket section is analyzed in detail here. The critical basket section is depicted in Figure 3.9.1-11. This approach is then validated by performing a buckling evaluation for the entire fuel basket cross section for the worst case loading condition.

The entire basket is 162.00 inches tall. However, the inertial load of the basket plates and fuel assemblies is conservatively assumed to be distributed over only a 144 inch length (effective fuel assembly length).

A.2. Material Properties used for Buckling Evaluation

All structural members of the 32PTH DSC fuel basket are constructed from SA-240 Type 304 stainless steel. A bilinear stress-strain curve for SA-240 Type 304 stainless steel is used for the elastic-plastic analysis.

The material properties used for the basket plates are taken from ASME Code, Section II, Part D [1], at 611° F. This temperature represents an average temperature for the fuel basket section analyzed and depicted in Figure 3.9.1-11.

Fuel Basket Material Properties at 611° F

Stainless Steel (SA-240 Type 304)

$E = 25.26 \times 10^6$ psi [1]

$S_y = 18.31$ ksi [1]

$S_u = 63.4$ ksi [1]

Tangent Modulus, E_T, is taken to be 5% of E,

$\Rightarrow E_T = 0.05 \times 25.26 \times 10^6 = 1.262 \times 10^6$ psi

A.3. Finite Element Buckling Analysis

Nonlinear stress analyses are conducted in order to evaluate the buckling loads for the 32PTH DSC basket plates at location shown in Figure 3.9.1-11. The three critical azimuth drop orientations analyzed are:

i) 0° (load applied in the direction parallel to the basket plates)
ii) 30° (load applied at 30° relative to the basket plate direction)
iii) 45° (load applied at 45° relative to the basket plate direction)

In order to calculate the buckling load, a small three-dimensional ANSYS finite element model is constructed using plastic large strain shell elements SHELL43 and large displacement option. This model is shown in Figure 3.9.1-12. The small model is constructed by selecting the appropriate elements and nodes from full basket cross section model as described in Section 3.9.1.2.3.A (page 3.9.1-7). As described in Section 3.9.1.2.3.A., the stiffness from aluminum plates is conservatively neglected but their weight is accounted for in the applied loads.

The loading on the small model are appropriately transferred from the full size basket loading. A maximum load of 200*g* is applied in each analysis. The automatic time stepping option "AUTOTS" is activated. This option lets the program decide the actual size of the load sub-step for a converged solution. The program stops at the load sub-step that fails to result in a converged solution. The last load step with a converged solution is used to compute the allowable collapse load for the fuel basket grid.

Applied Load Calculations

The vertical load due to the weight of top basket compartments is computed as follows (see Figure 3.9.1-11).

Weight of 5 fuel assemblies = $1585\times5\times15.0/144 = 825.52$ lb.
Weight of 5 SS square tubes = $5\times(9.075^2 - 8.70^2)\times15.0 \times 0.29 = 144.98$ lb.
Weight of 10 aluminum, 0.5 inch thick and 1 aluminum, 0.31 in thick plates
$= (8.70 + 2\times0.1875 + 0.5) \times 13.18 \times (10\times0.5 + 1\times0.31) \times 0.1 = 67.01$ lb.
Weight of SS plates (10 plates) = $10\times(8.70 + 2\times0.1875 + .5)\times1.75\times0.5 \times 0.29 = 24.30$ lb.
Weight of rail (1/2 of large rail) = (1297 lbs.) × 15.0 /144 = 135.1 lb.

Total weight = 825.52 + 144.98 + 67.01 + 24.30 + 135.1 = 1196.9 lb. ... say 1197 lb.

The horizontal load due to the adjacent fuel assembly is computed as follows.

Weight of fuel = 1,585 × 15.0 /144 = 165.10 lb.
Weight of aluminum 0.5 inch thick plate = (8.70 + 2 × 0.1875 + .5) ×13.18×0.5×0.1
= 6.31 lb.

Total weight = 165.10 + 6.31 = 171.41 lb. ... say 172 lb.
Side pressure due to W = 172 /(8.888 × 15.0) = 1.290 psi

For 200g,

Vertical Load = 200 × 1197 cosθ = 239,400 cosθ lb.
Pressure, applied to the vertical panel due to fuel weight = 200 (1.290) sinθ
= 258 sinθ psi.

Finite Element Analysis Results

The following table summarizes the input load and last converged load for all three load cases:

Basket Orientation	200g loads (psi)			Last converged Load (g)	Max. Deflection u_x (in)
	Vertical Load (lb)	Lateral Pressure (psi)	Applied Acceleration		
Vertical	239,400	0	(0, 200, 0)	107	0.005
30°	207,326	129.0	(-100, 173, 0)	84	0.1090
45°	169,281	182.4	(-141, 141, 0)	84	0.1277

As per paragraph F-1340 [3], the acceptability of a component may be demonstrated by collapse load analysis. The allowable collapse load shall not exceed 100% of plastic analysis collapse load ([3] F-1341.3). The plastic analysis collapse load is defined as that determined by plastic analysis according to the criteria given in II-1430 ([3] F-1321.6(c)).

It is seen from the above table that the 45° drop orientation is critical with respect to buckling. Using the methodology described in II-1430 ([3] F-1321.6(c)) the allowable collapse load has been determined for the critical 45° azimuth drop orientation, and is 81g. The collapse load analysis is shown in Figures 3.9.1-13. For the 0° drop orientations, collapse load will be higher than 81g, and for the 30° drop orientation, collapse load will be of the same order of magnitude.

A.4. Validation of the Small Model Analysis

In order to validate the small model buckling analysis technique, a buckling analysis of the full size basket was conducted for a 45° basket orientation drop. The finite element model, boundary conditions, real constants, material properties and loadings are the same as those described in Section 3.9.1.2.3. As described in Section 3.9.1.2.3.A., the stiffness from aluminum plates is conservatively neglected but their weight is accounted for in the applied loads. The loading applied to the model corresponds to a maximum load of 100*g*.

A nonlinear inelastic finite element analysis, with the large displacement option, is conducted to calculate the buckling load for the basket. The automatic time stepping option "AUTOTS" is activated. This option lets the program decide the actual size of the load sub-step for a converged solution. The program stops at the load sub-step that fails to result in a converged solution.

The inelastic finite element analysis converged up to a load of 86.4*g*. The deformed shape of basket, at 86.4g load, is shown in Figure 3.9.1-14. The 86.4*g* buckling load from the large basket buckling analysis is quite comparable to the small model buckling load of 84g. Thus, the large model analysis validates the use of a small model and applied boundary and loading conditions for the basket buckling load calculations.

A.5. Validation by Analytical Buckling Analysis

As an order of magnitude check, the 32PTH DSC basket plate allowable buckling load and interaction equations of paragraph NF-3322.1 (e) [5] are evaluated for the 75*g* side drop. The basket plates are evaluated for the 0° and 45° drop orientations, at a temperature of 611° F, on the most critically loaded panel.

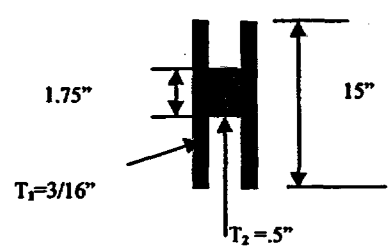
0°drop

According to ASME Code, Subsection NF, Paragraph NF-3322-1(c)(2)(a) [5] (Level A Condition) and modified as per Appendix F, Paragraph F-1334 [3] (Level D Condition), the compressive stress limit under accident conditions (Level D) when KL/r is less than 120 and $S_u >$ 1.2 S_y is,

$$F_a = 2 \times S_y\,[0.47 - (KL/r)/444]$$

Where,

K = 0.65 as recommended by AISC ([6], Table C1.8.1). Since the plate is continuously supported, the column is assumed to have fixed ends.
Plate height, L = 8.70 inches
Plate width, b = 15.0"
$E = 25.26 \times 10^6$ psi [1]
S_y = 18.31 ksi [1]
S_u = 63.4 ksi [1]



The moment of inertia, I, of the structure is,

$$I = 15.0 \times (0.875^3 - 0.5^3) / 12 + 1.75\ (0.5)^3/12 = 0.6812 + 0.0182 = 0.6994 \text{ in.}^4$$

The area, A,

$$A = 15.0 \times 2 \times 0.1875 + 1.75 \times 0.5 = 6.5 \text{ in.}^2$$

Therefore,

$$r = (I/A)^{1/2} = 0.328 \text{ in.}$$

$$KL/r = 0.65 \times 8.70 / 0.328 = 17.24$$

Substituting the values given above, the compressive stress limit, F_a, is,

$$F_a = 2 \times 18{,}310\ [0.47 - (17.24)/444] = 15{,}789 \text{ psi}$$

Therefore the limiting force, and limiting g load, is,

Limiting force = $F_a \times A$ = 15,789 × 6.5 = 102,629 lb.

Limiting g load = Limiting force / W = 102,629 /1,197 = 85.7g.

45° drop

The plate span is treated as a beam-column with fixed ends under axial compression and uniform transverse load ([15], Table VI, Case 10).

During a 45° side drop, the axial load (75g), P, is,

$$P = 75g \times 1{,}197 \cos(45) = 63{,}481 \text{ lb.}$$

The transverse pressure load (75g), P_t and the distributed transverse load, w, are,

$$P_t = 75g \times 1.290 \sin(45) = 68.41 \text{ psi.}$$

$$w = 68.41 \text{ psi} \times 15.0 \text{ in.} = 1026.19 \text{ lb./in}$$

The moment at the beam center is then,

$$M = wj^2\left[\frac{U/2}{\sin(U/2)} - 1\right]$$

Where,

$j = [EI/P]^{1/2} = [25.26\times10^6 \times 0.6994/63{,}481]^{1/2} = 16.68$
$U = L/j = 8.7/16.68 = 0.5216 \text{ rad.} = 29.89°$

Therefore,

$$M = 1026.19 \times 16.68^2 \, [\{.5216(0.5) / \sin(29.89/2)\} - 1] = 3{,}220 \text{ in. lb.}$$

The corresponding bending stress, f_b, is,

$$f_b = Mc/I = 3{,}220 \times 0.4375 / (0.6994) = 2{,}014 \text{ psi.}$$

Axial compressive stress, f_a,

$$f_a = P/A = 63{,}481/6.5 = 9{,}766 \text{ psi.}$$

From Subsection NF, NF 3322.1(e)(1)(a),

$$C_{mx} = 0.85$$

From Appendix F, F-1334.4(b),

$$F_b = 1.5\, S_y = 1.5 \times 18{,}310 = 27{,}465 \text{ psi.}$$

The value of F_e is calculated by the formula below as per Paragraph F-1334.5(b).

$$F_e = \pi^2 E / [1.3(kl/r)^2] = \pi^2\, 25.26\times10^6 / [1.3(17.24)^2] = 645{,}231$$

$$f_a / F_a + C_{mx} f_b / [(1 - f_b/ F_e) F_b] = 9{,}766 / 15{,}789 + 0.85\times2{,}014/[(1-2{,}014/ 645{,}231)\, 27{,}465] = 0.681 \leq 1$$

$$f_a / 2.0\,(0.6)\, S_y + f_b / F_b = 9{,}766 / 2.0\,(0.6)18{,}310 + 2{,}014 / 27{,}465 = 0.518 \leq 1$$

The results of the analytical calculations confirm that allowable buckling loads in basket plates due to the 75g side drop are within acceptable limits.

A.6. Summary of Basket Fuel Compartment Buckling Analysis Results

Since the critical collapse load for the 32PTH DSC basket fuel compartments (81*g* for the 45° Orientation) is greater than the maximum applied acceleration of 75*g*, the basket will not fail in buckling during the accident condition side drop event. The small finite element model technique used for the buckling analyses of the basket fuel compartment is verified by a full basket cross section finite element analysis as well as classical hand calculation methods.

B. Support Rail Buckling Analysis

This section evaluates the 32PTH DSC fuel basket support rail design with respect to buckling.

B.1. Approach

The fuel basket rail cross section geometry critical to the buckling evaluation is shown in Figure 3.9.1-15. The entire 32PTH DSC fuel basket is 162.0 inches in length, however, only a 15.0 inch long section of the support rails is analyzed in this section due to the symmetry of the basket design.

The maximum applied lateral acceleration analyzed, caused by accident condition transfer cask drop, is 75*g*, which is computed in Appendix 3.9.7.

B.2. Material Properties used for Buckling Evaluation

The material properties used for the basket support ail buckling evaluation are taken from ASME Code, Section II, Part D, at 600° F [1]. The maximum normal condition basket plate temperature is less than this (see Chapter4). A bilinear stress-strain curve for SA-240 Type 304 stainless steel is used for the elastic-plastic buckling analysis.

Fuel Basket Support Rail Material Properties at 600° F

Stainless Steel (SA-240 Type 304)

$E = 25.3\times10^6$ psi. [1]
$S_y = 18.4$ ksi. [1]
$S_u = 63.4$ ksi. [1]
Tangent Modulus, $E_T = 5\%$ of E = 1.265×10^6 psi.

B.3. Support Rail Finite Element Buckling Analysis

Nonlinear stress analyses are conducted to evaluate the plastic buckling loads for the 32PTH DSC fuel basket support rails at locations 1 and 2 (see Figure 3.9.1-15). A nonlinear finite element analysis, with large deflection option, is conducted to evaluate the plastic buckling loads for the support rails.

In order to calculate the buckling load for both support rail types, small three-dimensional ANSYS finite element models are constructed using SHELL43 plastic large strain shell elements. The small finite element models are extracted from large basket cross section model described in Section 3.9.1.2.3.A.

The vertical loading on the rail models are appropriately transferred from the full size basket loads. A maximum load of 200*g* is applied in each analysis. The automatic time stepping option AUTOTS is also activated. This option lets the program decide the actual size of the load sub-step for a converged solution. The program stops at the load sub-step that fails to result in a converged solution. The last load step with a converged solution is used to compute the allowable collapse load for each model.

Location 1 Applied Load Calculation (Large Rail, see Figure 3.9.1-15)

The vertical load due to weight of top compartments is computed as follows.

Weight of 10 fuel assemblies = 1,585 × 10 × 15.0/144 = 1,651 lb.
Weight of 12 SS square tubes = 12/32 [10,021 × 15.0/162] = 348 lb.
Weight of aluminum plates = 3 vertical plates × 56.95 × 13.18 × 0.5 × 0.1 + 6 horizontal plates × 18.65* × 13.18 × 0.5 × 0.1 = 112.6 + 73.7 = 186.3 lb.
Weight of SS plates = 3 vertical plates × 56.95 × 1.75 × 0.5 × 0.29 + 6 horizontal plates × 18.65 × 1.75 × 0.5 × 0.29 = 43.4 + 28.4 = 71.8 lb.
Weight of rail = 1297 × 15.0/162 = 120.1 lb.

Total Vertical weight, W_1 = 1651 + 348 + 186.3 + 71.8 + 120.1 = 2377.2 lb.
say 2,400 lb.

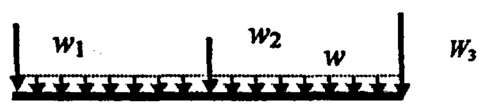
*2[8.7+2(.1875)] + 0.5 = 18.15 + 0.5 = 18.65 in.

The distributed vertical load on the top of the support rail is as follows.

Weight of 2 fuel assemblies = 2 × 1,585 × 15.0/144 = 330.2 lb.
Weight of 2 steel plate (box bottom) = 18.15 × 15.0 × 0.1875 × 0.29 = 14.8 lb.
Weight of aluminum plate = 18.65 × 13.18 × 0.3125 × 0.1 = 7.68 lb.
Weight of SS plate = 18.65 × 1.75 × 0.3125 × 0.29 = 3.0 lb.

Total weight, W_2 = 330.2 + 14.8 + 7.68 + 3.0 = 355.68 lb. ... say 365 lb.
Pressure due to distributed load, w = 365 / (2 × 8.8875 × 15.0) = 1.36896 psi

The applied nodal loads due to W_1 (2,400 lb.) are,



$w_1 = w_3$ = 2400/4 = 600 lb.
w_2 = 2400/2 = 1200 lb.

For the maximum applied acceleration of 200g, the loads computed above are scaled as follows.

Pressure Load, w = 200g × 1.36896 = 273.79 psi.
Nodal Force due to w_1 and w_3 = 200g × 600/9 (nodes) = 13,333 lb.
Nodal Force due to w_2 = 200g × 1200/9 (nodes) = 26,667 lb.

<u>Location 2 Applied Load Calculation (Small Support Rail, see Figure 3.9.1-15)</u>

The vertical load due to the weight of the top compartments is the following:

Weight of 3 fuel assemblies = 1,585 × 3 × 15.0/144 = 495.3 lb.
Weight of 4 SS square tubes = 4/32 [10021 × 15.0/162] = 116.0 lb.
Weight of aluminum plates = vertical plates (56.95 + 37.8) ×13.18 × 0.5 × 0.1
+ 4 horizontal plates × (9.575*) × 13.18 × 0.5 × 0.1 = 62.4 + 25.2 = 87.6 lb.
Weight of SS plates = vertical plates 94.75 × 1.75 × 0.5 × 0.29
+ 4 horizontal plates × 9.575 × 1.75 × 0.5 × 0.29 = 24.1 + 9.7 = 33.8 lb.
Weight of support rail = 809 × 15.0/162 = 74.9 lb.

Total vertical weight, W_1 = 495.3 + 116 + 87.6 + 33.8 + 74.9 = 807.6 lb. ... say 820 lb.

*8.7 + 2×(0.1875) + 0.5 = 9.075 + 0.5 = 9.575 in.

The distributed vertical load on the support tail top is computed as follows:

Weight of 1 fuel assembly = 1,585 × 15.0/144 = 165.1 lb.
Weight of 1 steel plate (box bottom) = 9.075 × 15.0 × 0.1875 × 0.29 = 7.4 lb.
Weight of aluminum plate = 9.575 × 13.18 ×0.5 × 0.1 = 6.3 lb.
Weight of SS plate = 9.575 × 1.75 × 0.5 × 0.29 = 2.4 lb.

Total distributed weight, W_2 = 165.1+ 7.4 + 6.3 + 2.4 = 181.2lb. ... say 185 lb.

Pressure due to W_2 = 185 / [(8.7 + .1875) × 15.0] = 1.39 psi.

The applied nodal loads due to W_1 (820 lb.), are,

$w_1 = w_2$ = 820/2 = 410 lb.

For the maximum applied acceleration of 200*g*, the loads computed above are scaled as follows:

Pressure Load, w = 200 × 1.39 = 278 psi
Nodal Force due to w_1 and w_2 = 200 × 410/9 (nodes) = 9,111 lb.

B.4. Summary of Support Rail Buckling Analysis Results

For both the large and small support rails, the finite element analysis converged up to the maximum 200*g* applied load. Therefore, both support rails are stable up to 200*g*. The results of the analysis show that the buckling loads of support rails are much higher than the maximum applied acceleration of 75*g*. Consequently, there is no potential for the basket support rails to buckle during the hypothetical accident condition side drop events.

3.9.1.3 32PTH DSC Shell Structural Evaluation

3.9.1.3.1 Approach

This section evaluates the structural adequacy of the 32PTH DSC Canister under all applicable normal and hypothetical accident condition loads. Evaluation of the stresses generated in the canister is presented in Section 3.9.1.3.2, and the DSC canister shell buckling evaluation is presented in Section 3.9.1.3.3.

3.9.1.3.2 DSC Canister Shell Stress Analysis

A. Methodology

An enveloping technique of combining various individual loads in a single analysis is used in this evaluation for several load combinations. This approach greatly reduces the number of computer runs while remains to be conservative. However, for some load combinations, the stress intensities under individual loads are added to obtain resultant stress intensities for the specified combined loads. This stress addition at the stress intensity level for the combined loads, instead of at component stress level, is also a conservative way to reduce numbers of analysis runs.

The ANSYS calculated stresses are the total stresses of the combined membrane, bending, and peak stresses. These total stresses are conservatively taken to be membrane stresses (P_m) as well as membrane plus bending stresses ($P_L + P_b$) and are evaluated against their corresponding ASME code stress limits. In the case where the total stresses, evaluated in this manner, exceed the ASME allowable stresses, a detailed stress linearization is performed to separate the membrane, bending, and peak stresses. The linearized stresses are then compared to their proper Code allowable stresses. ASME B&PV Code Subsection NB [8] is used for evaluation of loads under normal conditions and Appendix F [3] for evaluation of loads under hypothetical accident conditions.

The thermal stress intensities are classified as secondary stress intensities, Q, for code evaluations. The hypothetical accident blocked vent and flood loads (Service Level D) are conservative evaluated as normal loads.

B. Canister Material Properties

Material properties obtained from Reference 1 for the NUHOMS® 32 PTH canister materials, taken at the highest metal temperature of 500° F (from thermal evaluation presented in Chapter 4), are summarized here.

Elastic Material Properties

1. Canister Shell, Support Ring of Shield Plug, Outer Top Cover, and Bottom Grapple Ring (SA-240 Type 304) @ 500° F.

$E = 25.8\times10^6$ psi. $S_y = 19.4$ ksi.
$S_u = 63.4$ ksi. $S_m = 17.5$ ksi.
$\nu = 0.3$ $\rho = 0.29\ lb_f/in^3$

Temperature dependent coefficients of thermal expansion are as follows.

Temperature °F	$\alpha_{(T)}$ (in./in. °F^{-1})
70	8.5×10^{-6}
100	8.6×10^{-6}
150	8.8×10^{-6}
200	8.9×10^{-6}
250	9.1×10^{-6}
300	9.2×10^{-6}
350	9.3×10^{-6}
400	9.5×10^{-6}
500	9.7×10^{-6}

2. Top Shield Plug and Shell Bottom (SA-182 F304, plate thickness > 5 in) @ 500° F.

$E = 25.8\times10^6$ psi.
$S_u = 59.2$ ksi.
$\nu = 0.3$
$S_y = 19.4$ ksi.
$S_m = 17.5$ ksi.
$\rho = 0.29\ lb_f/in^3$

Temperature dependent coefficients of thermal expansion are as follows.

Temperature °F	$\alpha_{(T)}$ (in./in. °F^{-1})
70	8.5×10^{-6}
100	8.6×10^{-6}
150	8.8×10^{-6}
200	8.9×10^{-6}
250	9.1×10^{-6}
300	9.2×10^{-6}
350	9.3×10^{-6}
400	9.5×10^{-6}
500	9.7×10^{-6}

Elastic-Plastic Material Properties

The ANSYS Multilinear Kinematic Harding material option of inelastic analysis is employed in the analyses of all canister accident side drops. A multi-linear stress-strain curve for Type 304 stainless steel at 500°F is constructed using the yield and tensile stress values taken from Reference 1 and the elongation value from Reference 9. The stress-strain curve used for all canister materials is as follows.

Point	1	2	3	4	5
Strain (in/in)	0.0004845	0.000768	0.001164	0.00275	0.46
Stress (psi)	12,500	14,660	17,120	19,400	63,400

C. DSC Canister Stress Criteria

Allowable stresses given in ASME B&PV Code Subsection NB [8] and Appendix F [3] are used to evaluate the calculated stresses in the canister under normal, off-normal, and accident conditions, respectively. The allowable stresses for the transfer load case, evaluated at 500° F for both normal and accident conditions are summarized in Table 3.9.1-7. The allowable stresses for the storage load cases, evaluated at 450° F for normal conditions and 550° F for accident condition, are summarized in Table 3.9.1-8.

D. DSC Shell Stress Analysis for Transfer Loads

The evaluation of the stresses generated in the NUHOMS® 32 PTH canister during transfer operations is presented here. During fuel transfer, the canister is oriented horizontally inside the OS187H Transfer Cask. The OS187H Transfer Cask is mounted to the transfer skid and transferred from the fuel building to the ISFSI.

The maximum temperature in the canister under vacuum drying operation is calculated to be 511 °F in the thermal stress (see Chapter 4). This temperature occurs in the shell center where stresses are low. The maximum temperature in critical stress areas (top and bottom canister regions) are below 500° F. However, the stress evaluations are conservatively performed at 500° F.

D.1. Canister Transfer Load Cases

Elastic and elastic-plastic analyses are performed to calculate the stresses in the NUHOMS® 32 PTH canister under the transfer loads. These load cases are summarized in Table 3.9.1-9 and Table 3.9.1-10. All side drop loads are analyzed by elastic-plastic analyses and the rest by elastic analyses.

D.2. DSC Canister Finite Element Model Descriptions

Canister Thermal Model

There are three thermal load cases considered in this section. They are: a) vacuum drying, b) decay heat with ambient temperatures at 115° F, and c) at -20° F. An ANSYS 2-D thermal model is created using Thermal SOLID55 elements to calculate the metal temperatures in the canister for the vacuum drying load case.

For the cases with decay heat loads, no thermal models are created. The canister metal temperatures which are calculated in Chapter 4 are extracted and directly applied as temperature loads to the 2-D stress model using ANSYS macros.

2-D Canister Stress Models

A two-dimensional (2-D) axisymmetric ANSYS finite element model, constructed from PLANE42 elements, is used for the elastic analyses of all axisymmetrical loading on the canister. Only elastic properties of the canister materials are used in the analysis. The Canister Lifting Blocks are not simulated in the model. The effect of the omitted weight of the lifting blocks is negligible. ANSYS contact elements CONTAC12 are generated by connecting the nodes of two adjacent solids along their boundary. The real constant of each contact element is defined by the actual initial gap at each contact element.

At the weld locations between two joined solids, the contacting nodes are coupled in all directions without creation of contact elements. These coupled-nodes are applied to the welds between the shell and the support ring of the top shield plug and between the shell and the top shield plug. The larger ½ inch weld between the shell and the top cover is fully modeled with PLANE42 elements. The normal stiffness of all contact elements are calculated using guidelines in the ANSY 6.0 manual [10]. The applied boundary conditions for this 2-D model under each load case are described in the following sections. Figures 3.9.1-16, 3.9.1-17, and 3.9.1-18 show the ANSYS 2-D finite element model, which includes the canister shell, bottom, top shield plug and its support ring, and outer top cover. This model is used for analyses of all axisymmetric loads during the transfer operations of the canister.

The normal stiffness, K_N, for the contact elements ware estimated according to the ANSYS manual [10] as follows.

$$K_N \approx f E h$$

Where, f = Factor that controls contact compatibility (ranging between 0.01 to 100), use 1
E = Young's Modulus, use 25.8×10^6 psi
h = Average radius where contact to occur (for 2-D axisymmetrical model), use 34 in.

$K_N = 1 \times 25.8\times10^6 \times 34 = 8.8 \times10^8$ lb/in

3-D Canister Stress Model

A three-dimensional (3-D) ANSYS stress model is created from the axisymmetrical 2-D stress model using ANSYS elements SOLID45 and CONTAC52. The 3-D model is used for the analysis of accident side drops. To help reduce the ANSYS run time and assure numerical convergence, the whole canister is split into two portions, namely, the top and the bottom end sections. These two sections are represented by two different ANSYS models. Each end model includes the canister shell at a length beyond which the un-modeled shell will have no significant impact on the stress levels at the junction between the shell and its end closures. The DSC canister top end assembly finite element model is shown in Figure 3.9.1-19 and the canister bottom end assembly model is shown in Figure 3.9.1-20.

These 3-D models are used for analyses of side drops only. The postulated side drops will occur when the canister is nesting inside the OS187H transfer cask during transfer. Two side drops with the impact points located at 0° (i.e. the cask drops onto a target at 180° opposite to its four canister support pads) and at 180° (i.e. the cask drops onto a target between its two bottom canister support pads) are analyzed.

Load cases 6, 7, 10, and 11 consider the side drop loads at 0° and load cases 8, 9, 12, and 13 at 180° (see table 3.9.1-9). Elastic-plastic analyses, using multi-linear hardening material properties, are performed for both side drops. In addition to the contact elements generated from the 2-D model, new contact elements radially connecting the inner diameter of the cask and the outer diameter of the canister are generated. The nodes of these contact elements are located

either on the inner diameter of the cask or on the outer diameter of the canister at the moment when the cask hits the side drop target. The actual gaps between these contact element nodes at this moment were calculated and input for the contact element real constants. The contact element nodes located on the inner diameter of the cask are held fixed in all directions, simulating a rigid cask on which the canister drops.

A weak link element with a cross-section area of 0.1 in^2 and a Young's modulus of 1 psi is added to each contact element in the model to help numerical convergence. Zero density of these link elements is used to avoid adding any non-existing weights. This model does not calculate the stress levels in the middle section of the canister shell, which are calculated and evaluated in Section 3.9.1.2.3.

Only half of the canister in circumferential direction is included in the 3-D model. Symmetry boundary conditions are applied to the plane of symmetry (global Cartesian *x-y* plane) during a side drop. Symmetry boundary conditions are also applied to the cut-off plane at the canister shell to provide proper diametrical rigidity of the shell during side drops.

During the 75*g* side drop, the canister internals are accounted for by applying a cosine varying pressure distribution on the inside surface of the canister shell. Assuming that the canister internals react upon 90° arc of the inside surface, then the inertial load of the internals, $P_{(\theta)}$, which varies with angle, θ, ($\theta = 0$ is at the impact point), is governed by the following expression.

$$P_{(\theta)} = P_{max}\cos(2\theta) \qquad (0° < \theta < 45°)$$

Where P_{max} is the maximum pressure at the impact point ($\theta = 0$). Assuming the axial length of the applied load is L, the inside radius of the canister shell is R, and the load distribution, $P_{(\theta)}$ above, then the total inertial load generated by the internals, F, is the following.

$$F = \int_{\frac{-\pi}{4}}^{\frac{\pi}{4}} P_{\max}\cos(2\theta)\cos(\theta)LR\,d\theta$$

or,

$$F = \frac{P_{\max}LR}{2}\int_{\frac{-\pi}{4}}^{\frac{\pi}{4}} \cos((2+1)\theta)+\cos((2-1)\theta)d\theta$$

By integrating we get the following.

$$F = \left[\frac{P_{\max}LR}{2}\right]\left[\frac{\sin(3\theta)}{3}+\sin(\theta)\right]_{\frac{-\pi}{4}}^{\frac{\pi}{4}}$$

Therefore,

$$F = \left[\frac{P_{max}LR}{2}\right]\left[\frac{\sin\left(\frac{3\pi}{4}\right)}{3} + \sin\left(\frac{\pi}{4}\right) - \frac{\sin\left(\frac{-3\pi}{4}\right)}{3} - \sin\left(\frac{-\pi}{4}\right)\right]$$

$$F = P_{max}LR\left[\frac{\sin\left(\frac{3\pi}{4}\right)}{3} + \sin\left(\frac{\pi}{4}\right)\right]$$

The canister shell inner diameter, $R = 34.375$ in., the axial length of the applied load, $L = 164.88$ in. (Max. canister length 185.75 in – top 12.12 in – bottom 8.75 in). The total applied force, F, is equal to the inertial load of the canister internals, which is the following.

Basket weight = 29,854 lb,
Fuel assembly weight = 50,720 lb

Total weight of canister internals = 29,854 lb + 50,720 lb = 80,574 lb, ... say 81,000 lb

Then,

$$F = 81{,}000 \times 75g = 6{,}075{,}000 \text{ lb.}$$

Therefore, P_{max} is the following.

$$P_{max} = \frac{6{,}075{,}000}{(164.88)(34.375)}\left[\frac{\sin\left(\frac{3\pi}{4}\right)}{3} + \sin\left(\frac{\pi}{4}\right)\right]^{-1} = 1136.87 \text{ psi.}$$

The equivalent pressure applied on the canister inside shell surface is therefore,

$$P_{(\theta)} = 1136.87\cos(2\theta),$$

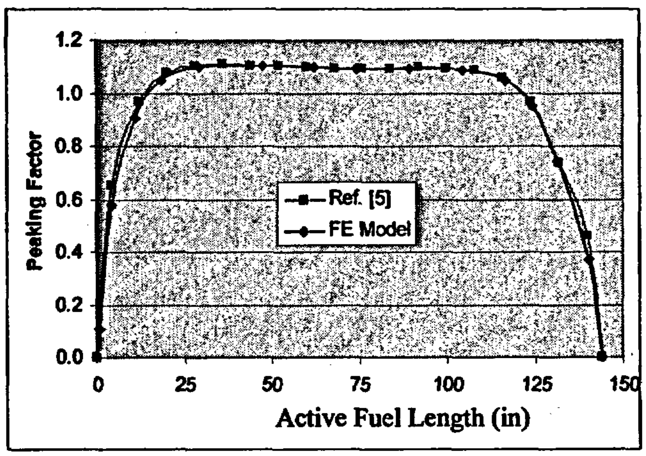
Where, θ is the angle from the bottom ($\theta = 0$) of the horizontal canister shell to the center of the shell element, up to 45°.

D.3. Canister Finite Element Analysis for Transfer Loads

All analyzed load cases in this section are identified in Tables 3.9.1-9 and 3.9.1-10 and are described in detail in the following sections.

Transfer Load Case 1: Deadweight + 15 psig external pressure + Thermal (Vacuum Drying)

The metal temperature profile in the canister shell is assumed to be of the same shape as that of the decay heat peaking factor reported in Chapter 4. The distribution of the decay heat along the fuel effective length for normal condition is shown in the following figure.



Three vacuum drying procedures, Procedure A, B, and C, are studied in Chapter 4. It shows that Procedure C, among all three procedures, produces the maximum metal temperature of 511° F in the canister. A steady-state thermal calculation using a 2-D canister thermal model is performed to calculate the temperature distribution throughout the canister. In this model the maximum temperature of 511° F is applied to the canister shell in locations corresponding to that between 26 inches and 125 inches of the active fuel length, where the maximum decay heat peaking factor occurs. Also an ambient temperature of 100° F is applied to the outer surfaces of the canister top and bottom plates. A steady-state thermal analysis is conducted to calculate the temperature profile in the canister. This temperature profile is then used as the thermal load for the stress analysis. The stress analysis of this load case contains two load steps. Load step 1 includes the primary loads of 1g down deadweight and an external pressure of 15 psig. Load step 2 includes these primary loads plus the secondary thermal loads from the thermal analysis.

For load step 1, the maximum stress intensity in the canister shell is 1,637 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is 1,341 psi., and the maximum stress intensity in the area of closure weld between the shell and the top cover plate is 410 psi.

For load step 2, the maximum stress intensity in the canister shell is 18,720 psi. The maximum stress intensity in the area of closure weld between the shell and the top is 2,114 psi., and the maximum stress intensity in the area of closure weld between the shell and the top cover plate is 413psi.

Transfer Load Case 2: **Handling, 2*g* Axial + 2*g* Transverse + 2*g* Vertical + 30 psig Int. Pressure + Thermal (115° F ambient)**

The handling loads applied to the canister in the horizontal orientation are analyzed in Section 3.9.1.2.3. It is judged that under the relatively light handling loads the maximum stresses in the canister will occur in the shell section and can be obtained from the results calculated in Section 3.9.1.2.3. Therefore only the axisymmetric loads, internal pressure of 30 psig and the 115° F ambient environment loads are analyzed in this section. The calculated stress intensities from these two computations are then conservatively added for comparison with the corresponding ASME Code allowable stresses.

The maximum primary membrane stress intensity and primary membrane plus bending stress intensity in the canister shell under the handling load of 2*g* are calculated, in Section 3.9.1.2.3, to be 1,516 psi and 13,239 psi, respectively.

The stress analysis of this load case contains two load steps. Load step 1 includes the primary loads of 30 psig internal pressure. Load step 2 includes this primary load plus the secondary thermal load from the thermal analysis.

The maximum primary stress intensity in the canister was calculated to be 3,332 psi in Load Step 1 analysis. The maximum primary stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,134 psi. The maximum primary stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 656 psi.

The maximum primary plus secondary stress intensity in the canister is calculated to be 36,219 psi under load step 2. The maximum primary plus secondary stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,646 psi. The maximum primary plus secondary stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 1,292 psi.

The maximum primary stress intensities in the canister shell calculated in Section 3.9.1.2.3 are to be added to these maximum primary and primary plus secondary stress intensities calculated under this Load Case for combined load evaluation per ASME stress limits. The direct addition of stresses at the stress intensities level, in stead of at the component level, as well as the addition of the maximum stress intensities at different locations is very conservative. This enveloping technique is used to minimize the computer runs.

Transfer Load Case 3: **Handling 2*g* Axial + 2*g* Transverse + 2*g* Vertical + 15 psig Ext. Pressure + Thermal (-20° F ambient)**

The same methodology described for load case 2 is used in this load case.

The maximum stress intensity in the canister for the primary load of 15 psig external pressure in load step 1, is calculated to be 1,666 psi. The maximum stress intensity in the area of the closure weld between the shell and the top shield plug is calculated to be 1,232 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 344 psi.

The maximum stress intensity in the canister for the primary load of 15 psig internal pressure plus the secondary temperature load in load step 2, is calculated to be 35,001 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 2,247 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate s calculated to be 1,514 psi.

Transfer Load Case 4: **120 psig internal pressure and hypothetical accident fire.**

Stresses in the canister under an internal pressure of 120 psig are calculated in this Load Case. ASME code [3] requires only primary stresses be evaluated under accident conditions. The secondary thermal stresses are therefore not calculated. The ANSYS 2-D model is used for analysis of this axisymmetrical pressure load.

The maximum calculated stress in the entire canister for the pressure load is 13,329 psi. This maximum stress intensity is conservatively treated both as primary membrane stress intensity and as primary membrane plus bending stress intensity and so evaluated against ASME code limits at the maximum metal temperature of the canister.

The maximum metal temperature in the canister during fire accident is calculated to be 790° F (see Chapter 4). Canister material properties at 800° F are used for the ANSYS model. The maximum stress intensity in the area of the closure weld between the shell and the shield plug is calculated to be 12,379 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 2,613 psi.

Transfer Load Case 5: **25 psig external pressure and flood hypothetical accident**

The external pressure of 25 psig on the canister is analyzed using material properties taken at 500° F for the entire model.

The maximum stress intensity in the canister for this load case is calculated to be 2,777 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 2,051 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 573 psi.

Transfer Load Case 6: **Accident Condition 75*g* side drop at 0° (No Rail) at ambient temperature of 115° F (75*g* side drop + 30 psig internal pressure) – Top End Portion of Canister**

The canister internal pressure of 30 psig plus a side acceleration of 75*g* is analyzed in this load case. A multi-linear elastic-plastic stress-strain curve for material 304 SS at 500° F is applied to all materials. The stress-strain curve is obtained from Reference 9. ASME code requires only primary stresses be evaluated under accident conditions. The values of the thermal expansion coefficients for all materials are therefore set to 0 to eliminate any secondary thermal stresses in the canister.

The maximum stress intensity in the canister for this load case is calculated to be 27,990 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 25,841 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 27,566 psi.

Transfer Load Case 7: **Accident condition 75*g* side drop at 0° (No Rail) at ambient temperature of 115° F (75*g* side drop + 30 psig internal pressure) – Bottom End Portion of Canister**

The methodology of the analysis and stress evaluation used in this load case is the same as that described for Load Case 6.

The maximum stress intensity in the canister for this load case is calculated to be 19,976 psi.

Transfer Load Case 8: **Accident 75*g* side drop at 180° (drop between two transfer cask bottom support pads) at ambient temperature of 115° F (75*g* side drop + 30 psig internal pressure) – Top End Portion of Canister**

The same methodology of the analysis and stress evaluation used for Load Case 6 is used for this load case except that the gaps between the canister and the rigid cask are different due to the orientation of the transfer cask support pads.

The maximum stress intensity in the canister for this load case is calculated to be 28,869 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 23,242 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 27,220 psi.

Transfer Load Case 9: **Accident 75*g* side drop at 180° (drop between two cask bottom rails) at ambient temperature of 115° F (75*g* side drop + 30 psig internal pressure) – Bottom End Portion of Canister**

The same methodology of the analysis and stress evaluation used for Load Case 7 is used for this load case except that the gaps between the canister and the rigid cask are different.

The maximum stress intensity in the canister for this load case is calculated to be 22,666 psi.

Transfer Load Case 10: **Accident 75*g* side drop at 0° (drop at no cask rail) at ambient temperature of -20° F (75*g* side drop + 15 psig external pressure) – Top End Portion of Canister**

The same methodology of the analysis and stress evaluation used for Load Case 6 is used for this load case except that external pressure instead of internal pressure is applied.

The maximum stress intensity in the canister for this load case is calculated to be 28,402 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 25,618 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 27,493 psi.

Transfer Load Case 11: **Accident 75*g* side drop at 0° (drop at no cask rail) at ambient temperature of -20° F (75*g* side drop + 15 psig external pressure) – Bottom End Portion of Canister**

The same methodology of the analysis and stress evaluation used for Load Case 7 are used for this load case except external pressure instead of internal pressure is applied.

The maximum stress intensity in the canister for this load case is calculated to be 19,381 psi.

Transfer Load Case 12: **Accident 75*g* side drop at 180° (drop between two cask bottom rails) at ambient temperature of -20° F (75*g* side drop + 15 psig external pressure) – Top End Portion of Canister**

The same methodology of the analysis and stress evaluation used for Load Case 8 is used for this load case except that external pressure instead of internal pressure is applied.

The maximum stress intensity in the canister for this load case is calculated to be 29,354 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 23,073 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 27,306 psi.

Transfer Load Case 13: **Accident 75*g* side drop at 180° (drop between two cask bottom rails) at ambient temperature of -20° F (75*g* side drop + 15 psig external pressure) – Bottom End Portion of Canister**

The same methodology of the analysis and stress evaluation used for Load Case 9 is used for this load case except that the external pressure instead of the internal pressure is applied.

The maximum stress intensity in the canister is calculated to be 22,650 psi.

Transfer Load Case 14: **Accident 75*g* Top End Drop (75*g* + Internal pressure of 30 psig)**

The weight of the canister internals (basket and fuel assemblies) during end drop is accounted for by applying equivalent pressures on canister components that support them. The actual weights of the canister basket and fuel assemblies are 29,854 lb and 50,720 lb. (see Section 3.2.1). Therefore, the total actual weight of the canister internals is 80,574 lb. The weight of the canister internals used in this analysis is conservatively increased to 81,000 lb.

The canister cavity inner radius at the top end is 34.375 in. The pressure load equivalent to the inertial load of the internals at 75*g* under accident condition, P_{ia}, is,

$$P_{ia} = [81{,}000 / (\pi \times 34.375^2)] \times 75g = 1636.481 \text{ psi.}$$

The top face of the canister outer top cover is held in the axial direction in order to simulate the rigid support provided by the transfer cask top cover. An inertial load of 75*g* in the negative *y*-direction is applied to the model. An internal pressure of 30 psig and the metal temperatures from the 115° F ambient condition are also included in this analysis. The temperatures in the canister are only applied so that the proper temperature dependent material properties are used. However, the values of thermal expansion coefficients for all materials are set to 0 to eliminate the secondary thermal stresses, which are not required for evaluation under an accident condition per Reference 3.

The maximum stress intensity in the canister for this load case is calculated to be 15,539 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 4,907 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 5,641 psi.

Transfer Load Case 15: **Accident 75*g* Bottom End Drop (75*g* + Internal pressure of 30 psig)**

The weight of the canister internals used in this analysis is 81,000 lb. The canister cavity inner radius at the bottom end is 34.375 in. The pressure load equivalent to the weight of the internals under the accident condition 75*g* drop, P_{ia}, is,

$$P_{ia} = [81{,}000 / (\pi \times 34.375^2)] \times 75g = 1636.481 \text{ psi.}$$

The bottom face of the canister is held in the axial direction in order to simulate the rigid support provided by the transfer cask bottom. An inertial load of 75*g* in the positive *y*-direction is applied to the model. An internal pressure of 30 psig and the metal temperatures from the 115 °F ambient condition are included in this analysis. The temperatures in the canister are only applied so that the proper temperature dependent material properties are used. However, the values of thermal expansion coefficients for all materials are set to 0 to eliminate the secondary thermal stresses, which are not required for evaluation under an accident condition per Reference 3.

The maximum stress intensity in the canister for this load case is calculated to be 16,492 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 5,295 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 8,714 psi.

Transfer Load Case 16: **Accident 75*g* Top End Drop (75g + External pressure of 15 psig)**

This load case is similar to Load Case 14 with different pressure loadings and metal temperatures. An external pressure of 15 psig and the metal temperatures from the -20° F ambient condition are applied in this analysis. The temperatures in the canister are only applied so that the proper temperature dependent material properties are used. However, the values of thermal expansion coefficients for all materials are set to 0 to eliminate the secondary thermal stresses, which are not required for evaluation under an accident condition per Reference 3.

The maximum stress intensity in the canister for this load case is calculated to be 18,545 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 7,916 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 6,135 psi.

Transfer Load Case 17: **Accident 75*g* Bottom End Drop in Accident Condition (75g + External Pressure of 15 psig)**

This load case is similar to Load Case 15 with different pressure loadings and metal temperatures. An external pressure of 15 psig and the metal temperatures from the -20° F ambient condition are applied in this analysis. The temperatures in the canister are only applied so that the proper temperature dependent material properties are used. However, the values of thermal expansion coefficients for all materials are set to 0 to eliminate the secondary thermal stresses, which are not required for evaluation under an accident condition per Reference 3.

The maximum stress intensity in the canister for this load case is calculated to be 19,956 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 12,319 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 6,279 psi.

Transfer Load Case 18: **Fabrication Test Condition (DW + 18 psig Internal Pressure + 155 kips Axial Load)**

After the canister bottom is welded to the shell a pressure test is conducted by applying an internal pressure of 18 psig(**) with a top seal plate being held by an axial force of 155 kips. The canister bottom may be made, as an option, of composite plates. For each of these options the inner plate, which is to be first welded to the shell and tested, has a minimum thickness of 1.69 inches. An ANSYS model, shown in Figures 3.9.1-21, is generated that simulates the canister shell with this thin bottom inner plate for analysis of pressure and axial loads under the test condition. The deadweight load on the horizontal canister is manually analyzed using Roark's formulas [7]. The stresses calculated from both manual and ANSYS analyses are conservatively added for ASME Code stress evaluation.

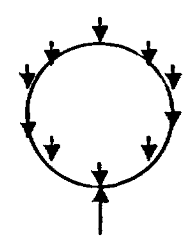
1) 1g deadweight load

It is conservatively assumed that the horizontal shell's own weight is line supported at its base.

From Case 15 of Table 17 in Roark's Formulas for Stress & Strains, 6th Edition.

R (mean radius) = ½ (69.75 in. – 0.5 in.) = 34.625 in.
t (wall thickness) = 0.5 in.
ρ (density) = 0.29 lb/in^3

Take unit length (L = 1 in.) of shell,
The weight per unit length of circumference of shell, w, is,



$2\pi\, R\, w$ (Line support)

$$w = (2 \times \pi \times R \times t \times L \times \rho) / (2 \times \pi \times R)$$
$$= t \times L \times \rho = 0.5 \times 1\text{in.} \times 0.29 \text{ lb/in}^3 = 0.145 \text{ lb/in}$$

For a thin ring, $I = \dfrac{t^3}{12(1-\upsilon^2)} = 0.01145$, where $\upsilon = 0.3$

$$K_T = 1 + \frac{I}{AR^2} \approx 1 \qquad K_2 = 1 - \alpha = 1 - \frac{I}{AR^2} \approx 1$$

$$\text{Max. } -M = -wR^2(1.6408\text{-}K_2) = -0.145 \times 34.625^2\,(1.6408 - 1) = -111.4 \text{ in-lb/in}$$

or,

$$\text{Max. } +M = (3/2)\,wR^2 = 1.5 \times .145 \times 34.625^2 = 260.76 \text{ in-lb/in}$$

$$\text{Max. bending stress, } \sigma_b = (6M)/(t^2) = (6 \times 260.76) / (0.5^2) = 6{,}258 \text{ psi}$$

$$N = N_A\text{Cos}(x) + V_A\text{ Sin}(x) + LT_N$$
$$V_A = 0$$

$$LT_N = -\,Wr(x)(\text{Sin}(x))$$

$$N_A = w\,R/2 = 2.51 \text{ lb/in}$$

$$N = 2.51 \text{ Cos}(x) - 0.145 \times 34.625 \times (x) \times \text{Sin}(x) \text{ lb/in}$$

$$N_{max} = 2.51 \text{ lb/in at } x = 0°.$$

$$\text{Max. membrane stress, } \sigma_m = N_{max} / t = (2.51 \text{ lb/in}) / (0.5 \text{ in}) = 5 \text{ psi}$$

2) 18 psig internal pressure + 155 kips axial load

An internal pressure of 18 psig is applied while an axial force of 155 kips is applied to a seal plate on the top of the shell. The net force applied to the entire circumference of the shell at the top will be 88,180 lb (155,000 lb – 18 lb/in^2 × [π/4 × 68.75^2] in^2 = 88,180 lb). A nodal force of 29,393 lb (88,180 / 3 = 29,393 lb) is applied at each node on the top end of the shell. Figure 3.9.1-21 shows the model with the applied pressure of 18 psig and the nodal forces of 88,180 lb.

The maximum stress intensity in the canister is calculated to be 14,143 psi under these testing loads.

The resultant stresses calculated in (1) and (2) are conservatively added and evaluated against ASME Code allowable stresses in Table 3.9.1-11.

Transfer Load Case 19: **Normal 80 kip Push Hydraulic Load (Internal Pressure of 30 pisg + 80 kip Push + Thermal load of 115° F ambient)**

During transfer of the canister from the transfer cask to the HSM a normal maximum push force of 80 kip is applied by a hydraulic ram over an area of 9 inch diameter on the canister bottom. A uniform pressure of 1,257.52 psig [= 80,000 lb / $((\pi/4) \times 9^2)$] is applied over this area. The periphery of the top cover outer surface is held as boundary condition. The sustained loads of an internal pressure of 30 psig plus the equivalent push load pressure of 1,257.5 psi are applied in load step 1. The sustained loads plus the temperature load from fuel decay heat are applied in load step 2.

The maximum stress intensity for load step 1 is calculated to be 7,238 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,123 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 669 psi.

The maximum stress intensity in the canister for load step 2 is calculated to be 34,916 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,602 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 1,267 psi.

Transfer Load Case 20: **Normal 60 kip Pull Hydraulic Load (Internal Pressure of 30 pisg + 60 kip Pull + Thermal of 115° F ambient)**

During retrieval of the canister from the HSM into the transfer cask a normal maximum pull force of 60 kips is applied by a hydraulic ram over an annulus area of 12.62 inches outer diameter and 10 inches inner diameter on the inside surface of grapple ring. A uniform pressure of 1,289.04 psig [= 60,000 lb / $((\pi/4) \times (12.62^2 - 10^2))$] is applied over this area. The periphery of the top cover outer surface is held as boundary condition. The sustained loads of an internal pressure of 30 psig plus the equivalent pull load pressure of 1,289.04 psi are applied in load step 1. The sustained loads plus the temperature load from fuel decay heat are applied in the load step 2.

The maximum stress intensity in the canister is calculated to be 11,484 psi for load step 1. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,134 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 656 psi.

The maximum stress intensity in the canister is calculated to be 36,753 psi for load step 2. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,646 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 1,292 psi.

Transfer Load Case 21: **Off-Normal 80 kip Push Hydraulic Load (Internal Pressure of 30 pisg + 80 kip Push + Thermal load of 115° F ambient)**

The same 80 kip push hydraulic load analyzed in load case 19 is also designated as an off-normal condition. Evaluation of this load in load case 19 as normal condition covers this off-normal condition.

Transfer Load Case 22: **Off-Normal 80 kip Pull Hydraulic Load (Internal Pressure of 30 pisg + 80 kip Pull + Thermal of 115° F ambient)**

During retrieval the canister from the HSM into the transfer cask a normal maximum pull force of 80 kips is applied by a hydraulic ram over an annulus area of 12.62 inches outer diameter and 10 inches inner diameter on the inside surface of grapple ring. A uniform pressure of 1,718.72 psig [$= 80{,}000 \text{ lb} / ((\pi/4) \times (12.62^2 - 10^2))$] is applied over this area. The periphery of the top cover outer surface is held as boundary condition. The sustained loads of an internal pressure of 30 psig plus the equivalent pull load pressure of 1,718.72 psi are applied in load step 1. The sustained loads plus the temperature load from fuel decay heat are applied in the load step 2.

The maximum stress intensity in the canister, other than the grapple ring and its support, is calculated to be 3,134 psi for load step 1. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 657 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 1,794 psi.

The maximum stress intensity in the grapple ring and its support is calculated to be 15,301 psi for load step 1. The calculated total stress intensities in the grapple ring and its support are further linearized to separate the membrane, bending, from the peak stress intensities. The maximum primary membrane stress in the grapple ring and its support is calculated, by stress linearization, to be 5,249 psi and the membrane plus bending stress intensity is 13,790 psi.

The maximum stress intensity in the canister is calculated to be 36,931 psi for load step 2. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,646 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 1,292 psi.

Transfer Load Case 23: **Accident 110 kip Push Hydraulic Load (Internal Pressure of 30 pisg + 110 kip Push)**

The maximum accident hydraulic force applied by the ram to push the canister from its transfer cask to the HSM is set at 110 kips. The load will be applied over an area with a 9 inch diameter on the canister bottom. A uniform pressure of 1,729.1 psig [$= 110{,}000 \text{ lb} / ((\pi/4) \times 9^2)$] is applied over this area in the 2-D ANSYS canister model. The periphery of the canister top cover outer surface is held as boundary condition. The sustained loads of an internal pressure of 30 psig plus

the equivalent push force pressure of 1,729.1 psi are applied as the loading. The secondary temperature load is not required by ASME code for an accident condition analysis.

The maximum stress intensity in the canister for this load case is calculated to be 9,854 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,132 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 686 psi.

Transfer Load Case 24: Accident 110 kip Pull Hydraulic Load (Internal Pressure of 30 psig + 110 kip Pull)

The maximum accident condition hydraulic force applied by the ram to pull the canister out of the HSM into the transfer cask is set at 110 kips. This pull force is applied over an annulus area of 12.62 inches outer diameter and 10 inches inner diameter on the inside surface of grapple ring. A uniform pressure of 2363.25 psig [$=110{,}000 \text{ lb} / ((\pi/4) \times (12.62^2 - 10^2))$] is applied over this area in the 2-D ANSYS canister model. The periphery of the top cover outer surface is held as a boundary condition. The sustained loads of an internal pressure of 30 psig plus the equivalent pull force pressure of 2363.25 psi are applied as loading. The secondary temperature load is not required by ASME code for an accident condition analysis.

The maximum calculated stress intensity in the canister for this load case is calculated to be 21,025 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,134 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 656 psi.

Transfer Load Case 25: Canister Lifting

ANSI N16.4 [11] is used as guide for evaluation of the canister lifting blocks. It requires that a load bearing member of a lifting device shall be capable of lifting six and ten times the weight of the canister without exceeding the material yield and ultimate strengths respectively. The combined shear stress or maximum tensile stress at any point in the device shall not exceed the material minimum tensile yield strength and the material ultimate tensile strength for 6*g* and 10*g* lifting loads, respectively. Since the ultimate tensile strength of the canister and its lifting block material exceeds 10/6 times their minimum tensile yield strength (i.e. $S_u / S_y > 10/6$), the 6*g* lifting load is the critical load for stress evaluation.

The empty canister weight is 28,191 lb. Therefore, the total lifting load is 28,191 lb × 6 = 169,146 lb.

Lifting Block Weld Stresses

It is conservatively assumed that the full lift load acts on the throat area of the 5/16 in. all-around weld of each 2.5 in. × 2.5 in. square lifting block. There are four lifting blocks welded to the inside surface of canister bottom. The total throat area of the welds in all four blocks is,

$$\text{Weld throat area} = (5/16 \text{ in} \times \cos 45°) \times 2.5 \text{ in} \times 4 \times 4 = 8.84 \text{ in}^2$$

Therefore, the maximum shear stress in the weld, τ_w, is,

$$\tau_w = 169{,}146 \text{ lb} / 8.84 \text{ in}^2 = 19{,}134 \text{ psi} < S_y$$

Lifting Block thread stresses

Each lifting block has internal threads of 1-1/4-7UNC-2B and a minimum thread length of 2 inches.

For each block, the thread stripping shear area = 2.9441 in^2 / in of engagement [12].
For an engaging length of 2 inches, the total stripping shear area for 4 blocks is,

$$A = 2.9441 \text{ in}^2 \times 2 \text{ in.} \times 4 = 23.55 \text{ in}^2$$

Since the thread stripping shear area is greater than area of the weld shear area in above, the design thread shear area is acceptable.

Max. Tensile stress in Lift Block

For each block, the cross-section area, A_b, is,

$$A_b = 2.5 \text{ in} \times 2.5 \text{ in} = 6.25 \text{ in}^2$$

The major diameter of a 1-1/4-7UNC-2B thread is 1.25 in. [13]

The minimum cross-section area, A_{min}, is,

$$A_{min} = 6.25 \text{ in}^2 - \pi/4 \times (1.25 \text{ in})^2 = 5.02 \text{ in}^2$$

Therefore, the maximum membrane tensile stress, P_m, is,

$$P_m = 169{,}146 \text{ lb} / (5.02 \text{ in}^2 \times 4) = 8{,}424 \text{ psi} < S_y$$

Where, S_y = 30,000 psi for SA-240 Gr.304 at 70° F.

Therefore, it is concluded that the design lift blocks for the canister is structurally adequate.

Stress intensities at the canister end closure welds

There are two closure welds in the canister design. One weld joins the canister shell and the top shield plug. Another weld joins the canister shell and its outer cover. Since the radiographic examination of these two closure welds is not feasible, ASME B&PV Code Case N-595-2 [14] requires a design stress reduction factor be used in the design calculation. The design stress reduction factor of 0.8 from Table 1 of Reference 14 is used in evaluation of the ANSYS calculated stresses at these two welds.

From the entire ANSYS model a set of elements surrounding these two end closure welds is selected for ANSYS postprocessing to calculate the element nodal stress intensities. This nodal stress postprocessing is performed for each load case results. The maximum calculated nodal stress intensity at the closure welds within this set of elements is reported in the above individual load case Sections. Tables 3.9.1-15 through 3.9.1-18 summarize these reported stress intensities at the welds and their evaluation.

D.4. Summary of the Stress Calculation Results for All Transfer Load Cases

Tables 3.9.1-11 through 3.9.1-18 summarize the calculated stress intensities in the canister and their corresponding ASME code evaluations for all transfer load cases.

Based on the results of these analyses, the design of the 32PTH DSC Canister is structurally adequate under transfer loads of testing, normal (Service Level A), and accident (Level D) conditions.

E. DSC Shell Stress Analysis for Storage Loads

This section evaluates the structural adequacy of the 32PTH DSC Canister when it is in the horizontal storage position within a HSM-H. This section considers storage loads on the canister under both normal and hypothetical accident conditions.

The evaluation of the stresses in the canister for storage loads employs an ANSYS 2-D axisymmetrical model to analyze three thermal conditions specified for the canister during storage. This 2-D model is the same model described in Section 3.9.1.3.2.D.2 used to compute stresses due to axisymmetric transfer loads. The analyses of axisymmetric loads such as internal and external pressure loads for transfer conditions are also valid for a horizontal storage canister. Their results are therefore used in this section for stress combinations and evaluations.

The fuel basket stress analysis for storage loads (Section 3.9.1.2.3.C) uses an ANSYS 3-D model, which includes the DSC canister shell, to calculate the non-axisymmetrical seismic and deadweight loads. The calculated stress intensities in the canister under the seismic and deadweight loads from Section 3.9.1.2.3.C are used in this section for stress combinations and evaluations.

The temperatures in the canister under 115° F and -20° F ambient conditions of and under HSH-H blocked vent conditions for 48 hours are computed in Chapter 4. These temperatures are imposed on the stress model in this evaluation for thermal stress calculations.

E.1. Canister Storage Load Cases

The storage load cases considered in this section are summarized in Table 3.9.1-19.

E.2. DSC Canister Finite Element Model Descriptions

The 2-D axisymmetrical thermal and stress models described in Section 3.9.1.3.2.D.2 for the canister transfer load analysis are also used for the storage load analysis. Figures 3.9.1-16, 3.9.1-17 and 3.9.1-18 show this model. This model is used in this section to evaluate the three specified thermal cases for storage, which are the -20° F and 115° F ambient conditions, and the blocked vent hypothetical accident condition. The temperature profiles in the canister for the three storage thermal cases are calculated in Chapter 4.

E.3. Canister Finite Element Analysis for Storage Loads

All individual load cases specified in Table 3.9.1-19 are described in details in the following sections.

Storage Load Case 1: Deadweight (1*g* Down)

The canister shell and fuel basket containing the fuel assemblies, resting horizontally on the rails of a HSM-H is analyzed in Section 3.9.1.2.3.C. for storage loads. The maximum primary membrane and membrane plus bending stress intensities in the canister shell due to the deadweight load are calculated to be 0.4 ksi, and 4.86 ksi, respectively (see Table 3.9.1-5).

Storage Load Case 2: Internal Pressure of 30 psig

The internal pressure of 30 psig applied on the canister is analyzed in load step1 of transfer load case 2 in Section 3.9.1.3.2.D. The maximum membrane plus bending stress intensities in the canister, calculated in Section 3.9.1.3.2.D is 3.33 ksi (see page 3.9.1-44).

Storage Load Case 3: Seismic Loads (0.65*g* Axial + 0.65*g* Transverse + 1.3*g* Vertical Down)

The seismic loads on the canister, containing the basket and the fuel assemblies and resting on the rails of a HSM-H, are analyzed in Section 3.9.1.2.3.C (page 3.9.1-21). The maximum primary membrane and membrane plus bending stress intensities are calculated in Section 3.9.1.2.3.C. to be 1.52ksi, and 11.03 ksi, respectively (see Table 3.9.1-5)). This specified seismic load includes a 1*g* deadweight load.

Storage Load Case 4: Thermal Load at -20° F Ambient

The analysis presented in Chapter 4 shows that the canister stored in the HSM-H without fins experiences higher temperatures than in the HSM-H with fins. The case of HSM without fins is therefore selected for analysis in this evaluation. The maximum temperature in the canister for this thermal case is calculated in Chapter 4 to be 324° F. The temperatures in the canister calculated in Chapter 4 are applied to the stress model in order to compute the thermal stress intensities in the canister. The maximum secondary thermal stress intensity is calculated to be 20.60 ksi.

Storage Load Case 5: Thermal Load at 115° F Ambient

The thermal load case with the canister stored in the HSM-H without fins, described in Chapter 4, is selected for this evaluation. The maximum temperature in the canister for this thermal case is calculated in Chapter 4 to be 434° F. The same procedure used for calculating the thermal stress intensities for the load case 4 is repeated for the 115° F ambient thermal load. The secondary thermal stress intensity is calculated to be 18.48 ksi.

Storage Load Case 6: Blocked Vent Thermal Accident Condition

The thermal evaluation presented in Chapter 4 reports four thermal cases for the canister stored in the HMS with blocked vent. The maximum temperature of 544° F in the canister is reached after 48 hours of complete vent blockage in a HSM without fins. The 48 hour vent blockage is a conservative scenario, since the vent is visually checked at least every 24 hours. However, this case is reported in thermal evaluation and is therefore selected for analysis in this section. The same procedure used for obtaining the thermal load in load case 4 is used in this load case. The secondary thermal stress intensity is calculated to be 13.84 ksi.

Storage Load Case 7: Accident Internal Pressure of 70 psig (In the Event of Blocked Vent)

The internal pressure of 70 psig in the canister is analyzed for enveloping the accident condition internal pressures during the blocked vent scenario. The maximum primary membrane plus bending stress intensity in the canister is calculated to be 7.77 ksi.

Storage Load Case 8: Accident Flood Load (Enveloped by External Pressure of 30 psig)

The hypothetical accident condition flood load is enveloped by an external pressure of 30 psig. The maximum primary membrane plus bending stress intensity in canister is calculated to be 3.33 ksi.

Stresses in the Canister End Closure Welds due to Storage Loads

Since the radiographic examination of the canister end closure welds is not feasible, Reference 14 requires a design stress reduction factor be used in the design calculation. There are two closure welds in the canister design. One is the weld between the canister shell and the top shield plug, and the other is the weld between the canister shell and its outer cover. The design stress reduction factor of 0.8 from Table 1 of Reference 14 is used to evaluate of the calculated stresses in these two welds. The maximum stress intensities in the entire canister calculated for all load cases are conservatively used for the stress evaluation at the welds. Table 3.9.1-21 summarizes the results of this the evaluation.

E.4. Summary of the Stress Calculation Results for All Storage Load Cases

Tables 3.9.1-20 and 3.9.1-21 summarize the calculated stresses in the entire canister and their corresponding ASME code evaluations.

Based on the results of this calculation, the 32PTH DSC canister is structurally adequate under all normal (Service Level A), off-normal (Service Level C), and hypothetical accident (Service Level D) conditions during storage.

3.9.1.3.3 DSC Shell Buckling Evaluation

This section evaluates the structural adequacy of 32PTH DSC canister against buckling during a vertical end drop during transfer operations.

A. Approach

A finite element plastic analysis with large displacement option is performed to monitor occurrence of canister shell buckling under the specified loads.

The thermal evaluation presented in Chapter 4 show that the metal temperatures of the entire canister are below 500° F during the transfer operations. The material properties of canister at 500° F are therefore conservatively used in this calculation.

B. Material Properties used for Canister Buckling Evaluation

The material properties of the canister materials, SA-240 Gr. 304 and SA-182 F304 (with thickness > 5 inches) stainless steel, at 500° F are as follows.

Property	@ 500° F
S_m (ksi)	17.5
S_y (ksi)	19.4
S_u (ksi)	59.2*
E (psi)	25.8×10^6

*Lesser value of S_u for materials SA-240 Gr. 304 and SA-182 F304 is used.

For the plastic finite element analysis, a multilinear modulus of elasticity is used. The following material stress-strain relation for SA-240 Gr. 304 is used:

Stress (at 500° F) (psi)	Strain (in /in)
12,500	0.0004845
14,660	0.0007680
17,120	0.001164
19,400	0.002750
59,200	0.46

C. Finite Element Buckling Analysis

The following two hypothetical accident load cases for the canister are considered in this buckling analysis.

Buckling Load Case 1: 15 psig external pressure and 75*g* axial acceleration due to 30 foot hypothetical accident condition drop

Buckling Load Case 2: 30 psig internal pressure and 75*g* axial acceleration due to 30 foot hypothetical accident condition drop

The two-dimensional axisymmetric finite element model of the canister described in Section 3.9.1.3.2.D.2 for the DSC canister stress analysis is used for this analysis. The gap element real constants, node couplings and displacement boundary conditions are also the same as those used in Section 3.9.1.3.2.D.2. The weight of canister's outer top cover plus the top shield plug and its support ring is 12,847 lb, and the bottom shield plug is 9,420 lb. Since the top end of the canister is heavier than the bottom end, it is a more severe case when the canister drops on its bottom end. A bottom end drop is therefore chosen for analysis in this calculation.

For each load case, a quasi-static plastic analysis consisting of two load steps is performed to monitor the buckling of canister. The first load step applies external pressure or internal pressure alone. A subsequent inertial load of 150*g* is added in the second load step. The outer surface of the canister bottom is held in order to simulate the case that the canister drops on a rigid cask bottom face.

In the load step 1, the stepped external or internal pressure is applied as a static load.

In the load step 2, the weight of the canister internals (basket and fuel assemblies) is accounted for by applying an equivalent internal pressure on the canister bottom. The actual total weight of the canister internals is 80,574 lb (basket 29,854 lb + fuel assemblies 50,720 lb). A total weight of 81,500 lb for the canister internals is conservatively used in this analysis. This inertial load is uniformly distributed over the bottom surface of the canister cavity with a radius of 34.375 in. This equivalent uniform pressure, P_{in}, exerted on the canister bottom by the weight of the internals under a 1*g* load is calculated as follows.

$$P_{in} = [81{,}500 / (\pi \times 34.375^2)] = 21.954 \text{ psi.}$$

An equivalent pressure of 3293.1 psig on the canister bottom corresponding to the 150*g* load ($P_{in} = 150 \times 21.954 = 3293.1$psi) is therefore applied to the canister bottom along with the 150*g* acceleration load in the load step 2.

A multilinear stress-strain relationship (with kinematic hardening) is used to obtain stresses and deflections beyond the elastic limit of the material. The large deflection option in ANSYS is activated to monitor the buckling response.

The ANSYS program stops at the first load sub-step that fails to result in a converged solution. The canister collapse load is determined using criteria in Paragraph F-1341.3 of Reference 3. It is constructed from a load-deflection curve developed from ANSYS converged results. Load Case 2 produces larger maximum lateral displacement than Load Case 1. Therefore Load Case 2 is selected for computation of the canister collapse load. The collapse load is determined to be 92g. Figure 3.9.1-22 shows the construction of the collapse load curve.

D. Summary Canister Buckling Analysis Results

The analysis shows that the collapse load for the canister end drop is 92g, which is well beyond the design 75*g* load. It is, therefore, concluded that buckling of the canister will not occur during a hypothetical accident end drop.

3.9.1.3.4 Evaluation of Alternate DSC Bottom Closure Assembly Design

This section evaluates the optional composite bottom assembly design of the canister. The same transfer loads and evaluation methodology used in Section 3.9.1.3.2.D are used in this evaluation, because the applied transfer loads bound all possible canister loads, including storage loads.

A. Approach

This analysis is performed to evaluate the structural adequacy of the optional composite canister bottom, relative to the one-piece canister bottom assembly. The same methodology used for the analysis of the solid canister bottom assembly, described in Section 3.9.1.3.2 is employed in this evaluation.

An enveloping technique of combining various individual loads in a single analysis is used. The ANSYS calculated total stresses are conservatively taken to be membrane stresses (P_m) as well as membrane plus bending stresses ($P_L + P_b$) or ($P_m + P_b$) and are accordingly evaluated against their corresponding ASME code stress limits. In case that the total stresses exceed the ASME allowable stresses, detailed stress linearization are then performed to separate the membrane, bending, and peak stresses from the total stresses, for their specific code evaluations. ASME B&PV Code Subsection NB [8] and NF [5], as designated for each canister component, are used to evaluate loads under normal conditions, and Appendix F [3] to evaluate loads under hypothetical accident conditions.

There are four optional composite bottom designs, designated Optional T1, T2, T3 and T4. The canister transfer loads specified in Table 3.9.1-9 are analyzed in this section by following the methodology used in Section 3.9.1.3.2. The analysis of the solid canister bottom assembly subject to side drop loads shows that the maximum stresses occurred at the junction between the canister shell and the solid bottom assembly. During the side drop of the canister, the stiff canister bottom prevents its connected flexible shell from flattening. This incompatible deformation between the two creates a maximum stress at their junction. Should the canister bottom become less stiff as in a composite design, it would better conform to the shell deformation and result in a less maximum stress at the junction. The side drop analysis of the canister with one-piece bottom design, presented in Section 3.9.1.3.2, will therefore bound the more flexible optional bottom designs. Therefore, the side drop load to the canister with the optional composite bottom designs are not analyzed in this section. All other transfer loads are analyzed for the canister with the optional composite bottom. The Optional T3 composite bottom design, which is structurally equivalent to the Optional T4 design, is selected for stress evaluations in this section, since it provides the least rigidity among all optional bottom designs.

B. Material Properties use for the Alternate Canister Design

Mechanical properties obtained from Reference 1 for 32PTH DSC canister materials, at the highest metal temperature of 500° F, from the transfer thermal analysis presented in Chapter 4, are used in this calculation, and are as follows.

1. Canister Shell, Support Ring of Shield Plug, Outer Top Cover, and Bottom Grapple Ring (SA-240 Type 304) @ 500° F.

$E = 25.8\times10^6$ psi. $S_y = 19.4$ ksi.
$S_u = 63.4$ ksi. $S_m = 17.5$ ksi.
$\nu = 0.3$ $\rho = 0.29\ lb_f/in^3$

Temperature dependent coefficients of thermal expansion are as follows.

Temperature °F	$\alpha_{(T)}$ (in./in. °F^{-1})
70	8.5×10^{-6}
100	8.6×10^{-6}
150	8.8×10^{-6}
200	8.9×10^{-6}
250	9.1×10^{-6}
300	9.2×10^{-6}
350	9.3×10^{-6}
400	9.5×10^{-6}
500	9.7×10^{-6}

2. Top Shield Plug and Shell Bottom (SA-182 F304, plate thickness > 5 in) @ 500° F.

$E = 25.8\times10^6$ psi. $S_y = 19.4$ ksi.
$S_u = 59.2$ ksi. $S_m = 17.5$ ksi.
$\nu = 0.3$ $\rho = 0.29\ \text{lb}_f/\text{in}^3$

Temperature dependent coefficients of thermal expansion are as follows.

Temperature °F	$\alpha_{(T)}$ (in./in. °F^{-1})
70	8.5×10^{-6}
100	8.6×10^{-6}
150	8.8×10^{-6}
200	8.9×10^{-6}
250	9.1×10^{-6}
300	9.2×10^{-6}
350	9.3×10^{-6}
400	9.5×10^{-6}
500	9.7×10^{-6}

3. Bottom Shield Plug Shield Plug in Optional Composite Bottom Design (SA-36) @ 500° F.

$E = 27.3\times10^6$ psi. $S_y = 29.3$ ksi.
$S_u = 58$ ksi. $S_m = 19.3$ ksi.
$\nu = 0.3$ $\rho = 0.29\ \text{lb}_f/\text{in}^3$

Temperature dependent coefficients of thermal expansion are as follows.

Temperature °F	$\alpha_{(T)}$ (in./in. °F^{-1})
70	6.4×10^{-6}
100	6.5×10^{-6}
200	6.7×10^{-6}
250	6.8×10^{-6}
300	6.9×10^{-6}
350	7.0×10^{-6}
400	7.1×10^{-6}
500	7.3×10^{-6}

C. Alternate Canister Design Stress Design Criteria

Allowable stresses given in ASME B&PV Code Subsection NB [8], NF [5], and Appendix F [3] are used to evaluate the calculated stresses in the canister under normal, off-normal and accident conditions. The allowable stresses for all materials, taken at 500° F, are summarized in Table 3.9.1-7 for NB components and Table 3.9.1-22 for NF components. Table 3.9.1-23 summarizes the allowable stresses for the weld between the canister shell and the bottom outer cover. This weld is designated a Subsection NF weld. Figure 3.9.1-23 shows the boundary between Code designations for Subsection NB and NF. All stresses are evaluated at metal temperature of 500° F for all load conditions except for the accident fire condition in Transfer Load Case 4.

D. Alternate Canister Design Evaluation Load Cases

Elastic finite element analyses are performed in order to compute the stresses in the 32PTH DSC canister subjected to the transfer loads. These load cases are tabulated in Tables 3.9.1-9 and 3.9.1-10.

E. Alternate Canister Design Finite Element Model Description

The 2-D ANSYS thermal and stress models described in Section 3.9.1.3.2.D.2 is modified with a composite bottom design, as shown in Figure 3.9.1- 23. The Optional T3 design is selected as the representative of the four composite bottom designs.

Thermal Model

There are three thermal load cases considered in this evaluation. They are vacuum drying and decay heat with ambient temperatures at 115° F and -20° F. The ANSYS 2-D thermal model is created using thermal SOLID55 elements to calculate the metal temperatures in the canister for the vacuum drying case.

For the thermal load cases with decay heat loads, no thermal models are created. The canister metal temperatures which are calculated in Chapter 4 are extracted and directly applied to the 2-D stress model.

Stress Models

A two-dimensional axisymmetric ANSYS finite element model, constructed from PLANE42 elements, is used for the elastic analyses of all axisymmetrical loads on the canister. Only elastic properties of the canister materials are used in the analysis. The canister lifting blocks are not simulated in the model. The effect of the omitted weight of the lifting blocks is negligible. ANSYS CONTAC12 elements are generated connecting the nodes of two adjacent solids along their boundary. The actual initial gap for each contact element is input by its real constant.

At the welds between two joined solids, the contacting nodes are coupled in all directions without creation of contact elements. These coupled-nodes are applied between the shell and the support ring of the top shield plug and between the shell and the top shield plug. The larger ½ inch weld between the shell and the top cover is fully modeled with PLANE42 elements. The normal stiffness of all contact elements is calculated using guidelines in the ANSYS 6.0 manual [10] in the following way.

$$K_n = f E h$$

Where f is a factor usually between 0.01 and 100, E is the material modulus of elasticity (25.8×10^6 psi), and h and is the average radius where contact occurs (34 inches is used). Therefore,

$$K_n = 1 \times 25.8\times10^6 \times 34 \times f = 8.8\times10^8 f$$

Thus, there is very wide range for K_n value. The structure responded well for a spring constant value of 8.8×10^6 lb/in ($f = 1$).

The applied boundary conditions for this 2-D model under each load case are described in the following sections.

F. Alternate Canister Design Finite Element Analysis

All analyzed load cases in this calculation are identified in Table 3.9.1-9 and 3.9.1-10 and are described in details in the following sections.

Transfer Load Case 1: **Deadweight + 15 psig external pressure + Thermal (Vacuum drying)**

The same methodology used to calculate the canister metal temperature profile in the standard canister design for the transfer load case described in Section 3.9.1.3.2.D.3 is used in this load case.

The stress analysis of this load case contains two load steps. Load step 1 includes the primary loads of 1g down deadweight and an external pressure of 15 psig. Load step 2 includes these primary loads plus the secondary thermal loads from the thermal analysis.

The maximum stress intensity in the canister for load case 1 is calculated to be 1,657 psi and 331 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 1,341 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 410 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 0 psi as it is compressive. The maximum stress intensity in the canister for load case 2 is calculated to be 14,668 psi and 6,369 psi in NB components and NF components, respectively. The maximum stress intensity in

the area of closure weld between the shell and the top shield plug is calculated to be 2,112 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 420 psi.

Transfer Load Case 2: **Handling 2*g* Axial + 2*g* Transverse + 2*g* Vertical + 30 psig Int. Pressure + Thermal (115° F ambient)**

The axisymmetrical loads due to an internal pressure of 30 psig and the temperature distribution due to the 115° F ambient environment are analyzed in this load case. The calculated stress intensities for this load case are added to the stress intensities in the canister shell computed in Section 3.9.1.2.3.B for the transfer loads. This is the same procedure used in Section 3.9.1.3.2.D for the standard canister design. The combined stress intensities are evaluated against ASME code allowable stresses.

The maximum primary membrane stress intensity and primary membrane plus bending stress intensity in canister shell due to the 2*g* handling load, computed in Section 3.9.1.3.2.D, are 1,516 psi and 13,239 psi, respectively.

The maximum stress intensity in the canister due to the primary load of 30 psig internal pressure in load step1 is calculated to be 3,831 psi and 4,067 psi in NB components and NF components respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,134 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 656 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 1,502 psi. These maximum stress intensities calculated in this load case are added directly to the maximum stress intensities in the shell calculated in Section 3.9.1.2.3.B for combined load evaluations. The direct addition of stresses at the stress intensities level, in stead of at component level, as well as addition of maximum stress intensities at different location is a conservative enveloping approach. This enveloping technique is used to minimize the computer runs.

The maximum stress intensities in the canister due to the primary load of 30 psig internal pressure plus the secondary temperature load in load step 2 are calculated to be 35,266 psi and 4,729 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,646 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 1,292 psi. These calculated maximum stress intensities are to be added to the maximum shell stress intensity calculated in Section 3.9.1.2.3.B for ASME code stress evaluation as described above for load step 1.

Transfer Load Case 3: **Handling 2*g* Axial + 2*g* Transverse + 2*g* Vertical + 30 psig Ext. Pressure + Thermal (-20° F ambient)**

The same methodology used in Transfer Load Case 2 is used for this load case.

The maximum stress intensity in the canister due to the primary load of external pressure of 15 psig in load step 1 is calculated to be 1,772 psi and 540 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 1,232 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 344 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 23 psi.

The maximum stress intensities in the canister due to the primary load of 15 psig internal pressure plus the secondary temperature load in load step 2 are calculated to be 27,619 psi and 12,448 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 2,247 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 1,514 psi.

Transfer Load Case 4: **120 psig internal pressure during the accident fire**

Stresses in the canister under an internal pressure of 120 psig are calculated in this load case. ASME code [3] requires only primary stresses be evaluated under accident conditions. The secondary thermal stresses are therefore not calculated. ANSYS 2-D model is used for analysis of this axisymmetrical pressure load.

The maximum stress intensity in the canister for this load case is calculated to be 15,389 psi and 16,570 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 12,502 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 2,628 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 6,249 psi. This maximum stress intensity is conservatively treated both as primary membrane stress intensity and as primary membrane plus bending stress intensity and so evaluated against ASME code limits at the maximum metal temperature of the canister. The maximum metal temperature in the canister during fire accident is calculated to be 790° F in Chapter 4. Canister material properties taken at 800° F are used in the ANSYS model.

Transfer Load Case 5: **25 psig external pressure during flood accident**

The external pressure of 25 psig on the canister is analyzed using material properties taken at 500° F for the entire model.

The maximum stress intensity in the canister for this load case is calculated to be 2,953 psi and 900 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 2,054 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 573 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 39 psi.

Transfer Load Cases 6 through 13:

Side drop Load cases 6 through 13 depicted in Table 3.9.1-9 are not analyzed in this calculation as explained in Section 3.9.1.3.4.A. These loads are bounded by Transfer Load Cases 6 through 13 analyzed in Section 3.9.1.3.2.D.

Transfer Load Case 14: **Accident 75*g* Top End Drop (75*g* + Internal pressure of 30 psig)**

The weight of the canister internals (basket and fuel assemblies) during an end drop is accounted for by applying equivalent pressures on the canister surfaces that support them. The actual weights of the canister basket and fuel assemblies are 29,854 lb and 50,720 lb, respectively. Therefore, the total actual weight of the canister internals is 80,574 lb. However, the weight of the canister internals used in this analysis is conservatively increased to 81,000 lb. The canister cavity inner radius at the top end is 34.375 in. The pressure load equivalent to the inertial load of the internals at 75*g*, P_{ia}, is,

$$P_{ia} = [81{,}000 / (\pi \times 34.375^2)] \times 75g = 1636.481 \text{ psi.}$$

The top end face of the canister outer top cover is held in the axial direction in order to simulate the rigid support provided by the transport cask lid. An inertial load of 75*g* in the negative *y*-direction is applied to the model. An internal pressure of 30 psig and the metal temperatures from the 155° F ambient thermal analysis performed in Chapter 4 are also included in this analysis. The temperatures in the canister are applied for proper use of material properties. However, the value of thermal expansion coefficients of all materials are set to 0 in order to eliminate secondary thermal stresses, which are not required for evaluation under accident conditions according to Reference 3.

The maximum stress intensity in the canister for this load case is calculated to be 15,645 psi and 6,337 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 4,814 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 5,578 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 483 psi.

Transfer Load Case 15: Accident 75*g* Bottom End Drop (75*g* + Internal pressure of 30 psig)

The weight of the canister internals used in this analysis is 81,000 lb. The canister cavity inner radius at the bottom end is 34.375 in. The pressure load equivalent to the weight of the internals during a 75*g* accident condition end drop, P_{ia}, is,

$$P_{ia} = [\ 81{,}000 / (\pi \times 34.375^2)\ ] \times 75g = 1636.481 \text{ psi.}$$

The bottom end face of the canister is held in the axial direction in order to simulate the rigid support provided by the transfer cask bottom. An inertial load of 75*g* in the positive *y*-direction is applied to the model. An internal pressure of 30 psig and the metal temperatures from the 115° F ambient thermal analysis presented in Chapter 4 are also included in this analysis. The temperatures in the canister are applied for proper use of material properties. However, the value of thermal expansion coefficients of all materials were set to 0 to eliminate the secondary thermal stresses, which are not required for evaluation under a accident condition per Reference 3.

The maximum stress intensity in the canister for this load case is calculated to be 16,584 psi and 9,024 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 8,582 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 5,283 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 596 psi.

Transfer Load Case 16: Accident 75*g* Top End Drop (75*g* + External pressure of 15 psig)

This load case is similar to Transfer Load Case 14 with different pressure loadings and metal temperatures. An external pressure of 15 psig and the metal temperatures from the -20° F ambient thermal analysis from Chapter 4 are applied in this analysis. The temperatures in the canister are applied for proper use of material properties. However, the value of thermal expansion coefficients of all materials are set to 0 to eliminate the secondary thermal stresses, which are not required for evaluation for accident conditions as per Reference 3.

The maximum stress intensity in the canister for this load case is calculated to be 18,685 psi and 6,992 psi in NB components and NF components, respectively. The maximum stress intensity in

the area of closure weld between the shell and the top shield plug is calculated to be 7,808 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 6,085 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 598 psi.

Transfer Load Case 17: Accident 75*g* Bottom End Drop in Accident Condition (75*g* + Extern. Press. of 15 psig)

This load case is similar to Transfer Load Case 15 with different pressure loadings and metal temperatures. An external pressure of 15 psig and the metal temperatures from the -20° F ambient thermal analysis in Chapter 4 are applied in this analysis. The temperatures in the canister are applied for proper use of material properties. However, the value of thermal expansion coefficients of all materials are set to 0 to eliminate the secondary thermal stresses, which are not required for evaluation for accident conditions as per Reference 3.

The maximum stress intensity in the canister for this load case is calculated to be 20,081 psi and 9,121 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 12,153 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 6,265 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 432 psi.

Transfer Load Case 18: Fabrication Test Condition (DW + 18 psig Internal Pressure + 155 kips Axial Load)

This load case is equivalent to the fabrication test load case analyzed for the standard canister design presented in Section 3.9.1.3.2.D, and is therefore not reevaluated.

Transfer Load Case 19: Normal 80 kip Push Hydraulic Load (Internal Pressure of 30 pisg + 80 kip Push + Thermal load of 115° F ambient)

In the event of transferring the canister from the transfer cask to the HSM-H a normal maximum push force of 80 kip will be applied by a hydraulic ram over an area with a 9 inch diameter on the canister bottom. A uniform pressure of 1,257.52 psig [$= 80{,}000 \text{ lb} / ((\pi/4) \times 9^2)$] is applied over this area. The periphery of the top cover outer surface is held as a boundary condition. The sustained loads of an internal pressure of 30 psig plus the equivalent push load pressure of 1,257.52 psi are applied in load step 1. The sustained loads plus the temperature load from fuel decay heat are applied in load step 2.

The maximum stress intensity in the canister for load step 1 is calculated to be 7,238 psi and 2,499 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,123 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 669 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 152 psi.

The maximum stress intensities in the canister for load case 2 are calculated to be 33,189 psi and 6,394 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,602 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 1,267 psi.

Transfer Load Case 20: **Normal 60 kip Pull Hydraulic Load (Internal Pressure of 30 pisg + 60 kip Pull + Thermal of 115° F ambient)**

In the event of retrieving the canister from the HSM into the transfer cask a normal maximum pull force of 60 kips will be applied by a hydraulic ram over an annulus area with a 12.62 inch outer diameter and a 10 inch inner diameter on the inside surface of grapple ring. A uniform pressure of 1,289.04 psig [$= 60{,}000 \text{ lb} / ((\pi/4) \times (12.62^2 - 10^2))$] is applied over this area. The periphery of the top cover outer surface is held as boundary condition. The sustained loads of an internal pressure of 30 psig plus the equivalent pull load pressure of 1289.04 psi are applied in load step 1. The sustained loads plus the temperature load from fuel decay heat are applied in the load step 2.

The maximum primary stress intensity in the canister for load step 1 is calculated to be 4,115 psi in NB components. The maximum primary stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,110 psi. The maximum primary stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 603 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 12,498 psi.

The maximum stress intensity in the canister NF components is calculated to be 30,046 psi at the junction between the shell and the bottom cover plate, which is a gross structural discontinuity according to NF-3121.14 of Reference 5. Bending stress at a gross structural discontinuity is a secondary stress as per NF-3121.3 of Reference 5. The allowable primary plus secondary stress intensity for loads under Normal condition (Service Level A) is $2S_y$ per Table NF-3522(b)-1 of Reference 5.

The calculated maximum total stress intensities in the grapple ring and its support for load case 1 are linearized to separate the membrane and bending stresses from the peak stress intensities. The maximum primary membrane stress in the grapple ring and its support is computed, by stress linearization, to be 10,410 psi and the membrane plus bending stress is computed to be 24,790 psi.

The maximum primary plus secondary stress intensities in the canister for load case 2 are calculated to be 35,029 psi and 30,340 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,660 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 1,311 psi.

Transfer Load Case 21: **Off-Normal 80 kip Push Hydraulic Load (Internal Pressure of 30 pisg + 80 kip Push + Thermal load of 115° F ambient)**

The Transfer Load Case 19 of normal 80 kip push hydraulic load is also designated as an off-normal load case. The stresses calculated in Transfer Load Case 19 are then used for evaluations against the allowable stresses of this off-normal load case.

Transfer Load Case 22: **Off-Normal 80 kip Pull Hydraulic Load (Internal Pressure of 30 pisg + 80 kip Pull + Thermal of 115° F ambient)**

In the event of retrieving the canister from the HSM into the transfer cask an off-normal maximum pull force of 80 kips will be applied by a hydraulic ram over an annulus area with a 12.62 inch outer diameter and 10 inch inner diameter on the inside surface of grapple ring. A uniform pressure of 1,718.72 psig [$= 80,000 \text{ lb} / ((\pi/4) \times (12.62^2 - 10^2))$] is applied over this area. The periphery of the top cover outer surface is held as boundary condition. The sustained loads of an internal pressure of 30 psig plus the equivalent pull load pressure of 1,718.72 psi are applied in load step 1. The sustained loads plus the temperature load from fuel decay heat are applied in the load step 2.

The maximum stress intensity in the canister for load step 1 is calculated to be 4,223 psi in NB components. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,110 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 603 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 16,407 psi.

The calculated total stress intensities in the grapple ring and its support are linearized to separate the membrane and bending stresses from the peak stress intensities. The maximum primary membrane stress in the grapple ring and its support is calculated, by stress linearization, to be 13,860 psi and the membrane plus bending stress is calculated to be 33,010 psi.

The primary plus secondary stress intensities calculated in the NF components under load step 2 are not required to be evaluated for Service Level C and D loads as per Table NF-3522(b)-1 of Reference 5. The maximum primary plus secondary stress intensity in the canister NB components for load step 2 is calculated to be 34,905 psi. The maximum primary plus secondary stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,660 psi. The maximum primary plus secondary stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 1,311 psi.

Transfer Load Case 23: **Accident 110 kip Push Hydraulic Load (Internal Pressure of 30 pisg + 110 kip Push)**

The hypothetical accident condition maximum hydraulic force applied by the ram to push the canister from the transfer cask to the HSM is set at 110 kips. The load is applied over an area with a 9 inch diameter on the canister bottom. A uniform pressure of 1,729.1 psig [= 110,000 lb / $((\pi/4) \times 9^2)$] is applied over this area in the 2-D ANSYS canister model. The periphery of the canister top cover outer surface is held as a boundary condition. The sustained loads of an internal pressure of 30 psig plus the equivalent push force pressure of 1,729.1 psi are applied as the loading. The secondary temperature load is not required by ASME code for analysis in an accident condition.

The maximum stress intensity in the canister for this load case is calculated to be 9,854 psi and 3,565 psi in NB components and NF components, respectively. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,132 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 688 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 135 psi.

Transfer Load Case 24: **Accident 110 kip Pull Hydraulic Load (Internal Pressure of 30 pisg + 110 kip Pull)**

The accident maximum hydraulic force applied by the ram to pull the canister out of the HSM into the transfer cask is set at 110 kips. This pull force is applied over an annulus area with a 12.62 inch outer diameter and a 10 inch inner diameter on the inside surface of grapple ring. A uniform pressure of 2363.25 psig [=110,000 lb / $((\pi/4) \times (12.62^2 - 10^2))$] is applied over this area in the 2-D ANSYS canister model. The periphery of the top cover outer surface is held as a boundary condition. The sustained loads of an internal pressure of 30 psig plus the equivalent pull force pressure of 2363.25 psi are applied as loading. The secondary temperature load is not required by ASME code for analysis in an accident condition.

The maximum stress intensity in the canister NB components for this load case is calculated to be 4,393 psi. The maximum stress intensity in the area of closure weld between the shell and the top shield plug is calculated to be 3,132 psi. The maximum stress intensity in the area of closure weld between the shell and the top cover plate is calculated to be 656 psi. The maximum tensile stress normal to the effective throat of the weld between the shell and the bottom outer cover plate is calculated to be 21,928 psi.

The calculated total stress intensities in the grapple ring and its support are linearized to separate the membrane and bending stresses from the peak stress intensities. The maximum primary membrane stress in the grapple ring and its support is calculated, by stress linearization, to be 19,090 psi and the membrane plus bending stress is calculated to be 45,480 psi.

Transfer Load Case 25: Canister Lifting

During the lifting of the canister the most weights, except the weight of the bottom inner plate to which the lifting blocks are attached, are transmitted through the shell and its connected bottom inner plate to the lifting blocks. Since the four 2.5" square lifting blocks are very close to the canister shell (1.5 inch from the edge of the block to the inner diameter of the shell), the load path of the bottom inner plate between the shell and the blocks is relatively rigid. It is therefore judged that the lifting blocks are the weakest link in the composite bottom design, as in the one-piece bottom design case, during a lifting operation. The total canister weight is the same for both composite and one-piece bottom canisters. Analysis of the lifting blocks in Transfer Load Case 25 of Section 3.9.1.3.2.D for canister with one-piece bottom is therefore also valid for the canister with the optional composite bottom designs.

Stresses in the Canister End Closure Welds due to Storage Loads

There are two closure welds in the alternate canister design. One weld joins the canister shell and the top shield plug. Another weld joins the canister shell and its outer cover. Since the radiographic examinations of these two closure welds are not feasible, Reference 14 requires a design stress reduction factor be used in the design calculation. The design stress reduction factor of 0.8 from Table 1 of Reference 14 is used in evaluation of the ANSYS calculated stresses at these two welds. Tables 3.9.1-27, 3.9.1-31, and 3.9.1-35 summarize these reported stress intensities at the welds and their evaluations. Only the higher of the calculated stress intensities in these two closure welds is reported in the summary Tables.

G. Summary of the Stress Calculation Results for the Alternate Canister Design

Tables 3.9.1-24 through 3.9.1-35 summarize the calculated stress intensities in the alternate canister design and their corresponding ASME code evaluations for all bounding transfer load cases.

Based on the results of these analyses, the 32PTH DSC alternate canister composite bottom assembly is structurally adequate with respect to normal, off-normal, and hypothetical accident conditions.

3.9.1.4 32PTH DSC and OS187H Transfer Cask Thermal Expansion Evaluation

3.9.1.4.1 Introduction

The purpose of this section is to determine the thermal growths among components of fuel cladding, basket, canister, and transfer cask in the 32PTH DSC. This thermal expansion calculation covers events of Vacuum Drying, Transfer, Storage, and Storage with Blocked Vent.

3.9.1.4.2 Approach

The temperatures of the fuel cladding, basket, canister, and transfer cask under various events are calculated in the thermal analyses of Chapter 4. Transient thermal analyses are conducted for the vacuum drying and blocked vent events. Steady-state thermal analyses are conducted for the normal and off-normal conditions during transfer and Storage. This section computes the thermal expansions at the steady-state temperatures in the events of Transfer and Storage.

In the vacuum drying load case, the profiles of transient temperature versus time computed in Chapter 4 are studied for selection of the critical time points at which the corresponding component temperatures would generate a minimum clearance between two nested components. For the blocked vent load case, the maximum temperatures from Chapter 4 are used in this calculation.

The cold dimensions of each pair of nested components are so determined, based on design tolerances, which generates a minimum cold clearance between the two components.

Unless otherwise stated, nominal dimensions of basket, canister, and cask are used for the thermal expansion calculations.

3.9.1.4.3 Mechanical Properties of Materials

The coefficient of thermal expansion of structural materials used for the fuel basket, canister shell, and transfer cask are provided in Table 3.9.1-6 as a function of temperature. The properties of SA-240 Gr.304 are taken from Reference 1, and the zircaloy properties are taken from Reference 4.

3.9.1.4.4 Thermal Expansion Computation

A. Thermal Expansion between the Length of Fuel Assembly and Canister Cavity

The extreme metal temperatures for the fuel cladding and canister under different cases are obtained from Chapter 4 for computation of the differential length growth. These temperatures are conservatively rounded and used in this calculation as listed in the following table.

Cases \ Component Temperature	Length Growth Between Fuel Cladding and Canister	
	Fuel Cladding Temp. (°F)	Canister (DSC Shell) Temp. (°F)
Vacuum Drying	750	210
Transfer	730	390
Storage – Off Normal	700	310
Storage – Blocked Vent	810	500

The spent fuel type was assumed to be Westinghouse 17×17 Standard.

Vacuum Drying Case (750° F Fuel Cladding with 210° F Canister)

The length of the spent fuel assembly when hot is,

$$L_F = L_T + (L_Z \times \alpha_Z + L_S \times \alpha_S)\,\Delta T$$

Where,

L_F = Hot length of PWR fuel assembly, in.

L_T = Total length of fuel assembly at room temperature = 162.4 in.

L_Z = Length of Zircaloy guide tube ≅144 in.

α_Z = Zircaloy axial coefficient of thermal expansion = 2.79×10^{-6} in./in.°F at 750° F

L_S = Length of stainless steel per fuel assembly ≅18.4 in.

α_S = Stainless steel coefficient of thermal expansion = 10×10^{-6} in./in.°F at 750° F

ΔT = 750° F – 70° F = 680° F

Therefore,

$L_F = 162.4 + (144 \times 2.79 + 18.4 \times 10) \times 10^{-6} \times 680 = 162.80$ in.

Allowing 1.25 inch for irradiation growth of the spent fuel assembly, the total assembly length including thermal expansion is 164.05 inches. The minimum length of the canister cavity at room temperature is 164.5 inches. The minimum length of the canister cavity at 210° F is,

$$L_{CH} = L_{CC} + L_{CC} \times \alpha_C \times \Delta T$$

Where,

L_{CH} = Hot length of canister cavity, in.

L_{CC} = Minimum canister cavity length at room temperature = 164.5 in.

α_C = Stainless steel coefficient of thermal expansion = 8.94×10^{-6} in./in.°F at 210° F

ΔT = 210° F – 70° F = 140° F

Therefore,

$L_{CH} = 164.5 + 164.5 \times 8.94 \times 10^{-6} \times 140 = 164.7 \text{ in} > 164.05 \text{ in}$

Thus, adequate clearance has been provided between the PWR spent fuel assemblies and the canister cavity length to permit free thermal expansion.

<u>Transfer Case</u> (730 °F Fuel Cladding with 390 °F Canister)

The length of the spent fuel assembly when hot is,

$$L_F = L_T + (L_Z \times \alpha_Z + L_S \times \alpha_S)\,\Delta T$$

Where,

L_F = Hot length of PWR fuel assembly, in.

L_T = Total length of fuel assembly at room temperature = 162.4 in.

L_Z = Length of Zircaloy guide tube ≅144 in.

α_Z = Zircaloy axial coefficient of thermal expansion = 2.73×10^{-6} in./in.°F at 730° F

L_S = Length of stainless steel per fuel assembly ≅18.4 in.

α_S = Stainless steel coefficient of thermal expansion = 10×10^{-6} in./in.°F at 730° F

ΔT = 730° F – 70° F = 660° F

Therefore,

$$L_F = 162.4 + (144 \times 2.73 + 18.4 \times 10) \times 10^{-6} \times 660 = 162.78 \text{ in.}$$

Allowing 1.25 inches for irradiation growth of the spent fuel assembly, the total assembly length including thermal expansion is 164.03 inches. The minimum length of the canister cavity at room temperature is 164.5 inches. The minimum length of the canister cavity at 390° F is,

$$L_{CH} = L_{CC} + L_{CC} \times \alpha_C \times \Delta T$$

Where,

L_{CH} = Hot length of canister cavity, in.

L_{CC} = Minimum canister cavity length at room temperature = 164.5 in.

α_C = Stainless steel coefficient of thermal expansion = 9.46×10^{-6} in./in.°F at 390° F

ΔT = 390° F -70° F = 320° F

Therefore,

$$L_{CH} = 164.5 + 164.5 \times 9.46\times10^{-6} \times 320 = 165 \text{ in} > 164.03 \text{ in}$$

Thus, adequate clearance has been provided between the PWR spent fuel assemblies and the canister cavity length to permit free thermal expansion.

Storage – Off-Normal Case (700 °F Fuel Cladding with 310 °F Canister)

The length of the spent fuel assembly when hot is,

$$L_F = L_T + (L_Z \times \alpha_Z + L_S \times \alpha_S)\,\Delta T$$

Where,

L_F = Hot length of PWR fuel assembly, in.

L_T = Total length of fuel assembly at room temperature = 162.4 in.

L_Z = Length of Zircaloy guide tube ≅144 in.

α_Z = Zircaloy axial coefficient of thermal expansion = 2.70×10^{6} in./in.° F at 700° F

L_S = Length of stainless steel per fuel assembly ≅18.4 in.

α_S = Stainless steel coefficient of thermal expansion = 10×10^{-6} in./in.°F at 700° F

ΔT = 700° F – 70° F = 630° F

Therefore,

$$L_F = 162.4 + (144 \times 2.70 + 18.4 \times 10) \times 10^{-6} \times 630 = 162.76 \text{ in.}$$

Allowing 1.25 inch for irradiation growth of the spent fuel assembly, the total assembly length including thermal expansion is 164.01 inches. The minimum length of the canister cavity at room temperature is 164.5 inches. The minimum length of the canister cavity at 310° F is,

$$L_{CH} = L_{CC} + L_{CC} \times \alpha_C \times \Delta T$$

Where,

L_{CH} = Hot length of canister cavity, in.

L_{CC} = Minimum canister cavity length at room temperature = 164.5 in.

α_C = Stainless steel coefficient of thermal expansion = 9.22×10^{-6} in./in.°F at 310° F

ΔT = 310° F -70° F = 240° F

Therefore,

$$L_{CH} = 164.5 + 164.5 \times 9.22 \times 10^{-6} \times 240 = 164.86 \text{ in} > 164.01 \text{ in}$$

Thus, adequate clearance has been provided between the PWR spent fuel assemblies and the canister cavity length to permit free thermal expansion.

Storage – Blocked Vent Case (810° F Fuel Cladding with 500° F Canister)

The length of the spent fuel assembly when hot is,

$$L_F = L_T + (L_Z \times \alpha_Z + L_S \times \alpha_S)\,\Delta T$$

Where,

L_F = Hot length of PWR fuel assembly, in.

L_T = Total length of fuel assembly at room temperature = 162.4 in.

L_Z = Length of Zircaloy guide tube ≅144 in.

α_Z = Zircaloy axial coefficient of thermal expansion = 2.7×10^{-6} in./in.°F at 810° F

L_S = Length of stainless steel per fuel assembly ≅18.4 in.

α_S = Stainless steel coefficient of thermal expansion = 10.1×10^{-6} in./in.°F at 810° F

ΔT = 810° F – 70° F = 740° F

Therefore,

$$L_F = 162.4 + (144 \times 2.7 + 18.4 \times 10.1) \times 10^{-6} \times 740 = 162.83 \text{ in.}$$

Allowing 1.25 inch for irradiation growth of the spent fuel assembly, the total assembly length including thermal expansion is 164.08 inches. The minimum length of the canister cavity at room temperature is 164.5 inches. The minimum length of the canister cavity at 500° F is,

$$L_{CH} = L_{CC} + L_{CC} \times \alpha_C \times \Delta T$$

Where,

L_{CH} = Hot length of canister cavity, in.

L_{CC} = Minimum canister cavity length at room temperature = 164.5 in.

α_C = Stainless steel coefficient of thermal expansion = 9.7×10^{-6} in./in.°F at 500° F

ΔT = 500° F – 70° F = 430° F

Therefore,

$$L_{CH} = 164.5 + 164.5 \times 9.7\times10^{-6} \times 430 = 165.19 \text{ in } > 164.08 \text{ in}$$

Thus, adequate clearance has been provided between the PWR spent fuel assemblies and the canister cavity length to permit free thermal expansion.

B. Thermal Expansion between the Outer Diameter of the Basket and the Inner Diameter of the Canister Cavity

The following spreadsheet is used to compute the relative thermal expansions between the fuel basket outer diameter and the DSC canister cavity inner diameter. All maximum and minimum component temperatures are taken fro the thermal evaluation performed in Chapter 4.

The maximum outer diameter of the basket at room temperature, D_b, is computed as follows.

$$D_b = 68.53 \text{ in.} - 0.16 \text{ in. min. gap} = 68.370 \text{ in.}$$

The minimum inner diameter of the DSC canister cavity at room temperature, D_c, is computed as follows.

$$D_c = (69.75 \text{ in.} - 0.12 \text{ in.}) - 2 \times (0.50 \text{ in.} + 0.05 \text{ in.}) = 68.530 \text{ in.}$$

Event	Load Case	T_{CNH} (°F)	α_{CN} (in/in-°F)	T_{BKH} (°F)	α_{BK} (in/in-°F)	D_{CNH} (in)	D_{BKH} (in)	D_{CNH} - D_{BKH} (in)
Vacuum Drying	Procedure A	218	8.972E-06	614	9.828E-06	68.621	68.736	-0.115
	Procedure A (1)	218	8.972E-06	377	9.408E-06	68.621	68.567	0.054
	Procedure B	393	9.472E-06	634	9.868E-06	68.740	68.751	-0.011
	Procedure C (2)	180	8.860E-06	390	9.460E-06	68.597	68.577	0.020
	Procedure C	504	9.708E-06	642	9.884E-06	68.819	68.757	0.062
Transfer	115° F Amb. # 1	475	9.650E-06	627	9.854E-06	68.798	68.745	0.053
	115° F Amb. # 2	475	9.650E-06	613	9.826E-06	68.798	68.735	0.063
	115° F Amb. # 3	475	9.650E-06	616	9.832E-06	68.798	68.737	0.061
	115° F Amb. # 4	475	9.650E-06	624	9.848E-06	68.798	68.743	0.055
	-20° F Amb. # 1	398	9.492E-06	554	9.800E-06	68.743	68.694	0.049
	115° F Amb. # 1	475	9.650E-06	629	9.858E-06	68.798	68.747	0.051
Storage	115° F Amb. w/ Fins	422	9.544E-06	584	9.800E-06	68.760	68.714	0.046
	115°F Amb. w/o Fins	434	9.568E-06	595	9.800E-06	68.769	68.722	0.047
	-20° F Amb. w/ Fins	319	9.238E-06	492	9.684E-06	68.688	68.649	0.038
	-20°F Amb. w/o Fins	324	9.248E-06	495	9.690E-06	68.691	68.652	0.039
Storage Blocked Vent	34 hr w/ Fins	511	9.722E-06	677	9.954E-06	68.824	68.783	0.041
	48 hr w/ Fins	537	9.774E-06	703	1.000E-05	68.843	68.803	0.040
	31 hr w/o Fins	512	9.724E-06	678	9.956E-06	68.825	68.784	0.041
	48 hr w/o Fins	544	9.788E-06	709	1.000E-05	68.848	68.807	0.041

Notes:

1. Temperatures are taken from the thermal analysis case with zero gaps between the basket and canister in Chapter 4.
2. Temperatures are taken at time = 14 hour, which has the largest temperature difference between the Canister and the Basket throughout Procedure C in Chapter 4.

Nomenclature,

$D_{CNH} = 68.53 \times [1+ \alpha_{CN} \times (T_{CNH} - 70)]$

$D_{BKH} = 68.37 \times [1+ \alpha_{BK} \times (T_{BKH} - 70)]$

T_{CNH} = Temperature of the hot canister

α_{CN} = Thermal expansion coefficient of the canister at T_{CNH} temperature

T_{BKH} = Average of temperatures of the hot basket and basket rail.

α_{BK} = Thermal expansion coefficient of the basket at T_{BKH} temperature

D_{CNH} = Inner diameter of the hot canister cavity at T_{CNH} temperature

D_{BKH} = Outer diameter of the hot basket rail at T_{BKH} temperature

$D_{CNH} - D_{BKH}$ = diametrical hot clearance between the inner diameter of the canister and the outer diameter of the basket rail

C. Thermal Expansion between the Length of Basket and Canister Cavity

The maximum basket length at room temperature, L_b =162.120 inches.

The minimum canister cavity length at room temperature, L_c = 164.50 inches.

Event	Case	T_{CNH} (°F)	α_{CN} (in/in-°F)	T_{BKH} (°F)	α_{BK} (in/in-°F)	L_{CNH} (in)	L_{BKH} (in)	L_{CNH} - L_{BKH} (in)
Vacuum Drying	Procedure A	218	8.972E-06	614	9.828E-06	164.718	162.987	1.732
	Procedure B	393	9.472E-06	634	9.868E-06	165.003	163.022	1.981
	Procedure C(1)	180	8.860E-06	390	9.460E-06	164.660	162.611	2.050
	Procedure C	504	9.708E-06	642	9.884E-06	165.193	163.037	2.157
Transfer	115° F Amb. # 1	475	9.650E-06	627	9.854E-06	165.143	163.010	2.133
	115° F Amb. # 2	475	9.650E-06	613	9.826E-06	165.143	162.985	2.158
	115° F Amb. # 3	475	9.650E-06	616	9.832E-06	165.143	162.990	2.153
	115° F Amb. # 4	475	9.650E-06	624	9.848E-06	165.143	163.004	2.138
	-20° F Amb. # 1	398	9.492E-06	554	9.800E-06	165.012	162.889	2.123
	115° F Amb. # 1	475	9.650E-06	629	9.858E-06	165.143	163.013	2.130
Storage	115° F Amb. w/ Fins	422	9.544E-06	584	9.800E-06	165.053	162.937	2.116
	115° F Amb. w/o Fins	434	9.568E-06	595	9.800E-06	165.073	162.954	2.119
	-20° F Amb. w/ Fins	319	9.238E-06	492	9.684E-06	164.878	162.783	2.096
	-20° F Amb. w/o Fins	324	9.248E-06	495	9.690E-06	164.886	162.788	2.099
Storage Blocked Vent	34 hr w/ Fins	511	9.722E-06	677	9.954E-06	165.205	163.100	2.106
	48 hr w/ Fins	537	9.774E-06	703	1.000E-05	165.251	163.146	2.105
	31 hr w/o Fins	512	9.724E-06	678	9.956E-06	165.207	163.101	2.106
	48 hr w/o Fins	544	9.788E-06	709	1.000E-05	165.263	163.156	2.107

Notes:

1. Temperatures are taken at time = 14 hour, which has the largest temperature difference between the canister and the basket throughout Procedure C in Chapter 4.

Nomenclature,

L_{CNH} = 164.5× [1+ α_{CN} × (T_{CNH} -70)]
L_{CNH} = 162.12 × [1+ α_{BK} × (T_{BKH} -70)]
T_{CNH} = Temperature of the canister
α_{CN} = Thermal expansion coefficient of canister at T_{CNH} temperature
T_{BKH} = Average of temperatures of hot basket and basket rail
α_{BK} = Thermal expansion coefficient of the basket at T_{BKH} temperature
L_{CNH} = Length of hot canister cavity at T_{CNH} temperature
L_{BKH} = Length of hot basket at T_{BKH} temperature
$L_{CNH} - L_{BKH}$ = Hot clearance between the length of the canister cavity and the length of the basket

D. Thermal Expansion between the Outer Diameter of the Canister and the Inner Diameter of the Cask Body

The maximum outer diameter of the canister at room temperature, D_{cmax} = 69.87 inches.

The minimum inner diameter of the canister cavity at room temperature, D_{cmin} = 70.35 inches.

Event	Case	T_{CKH} (°F)	α_{CK} (in/in °F)	T_{CNH} (°F)	α_{CN} (in/in °F)	D_{CKH} (in)	D_{CNH} (in)	D_{CKH} - D_{CNH} (in)	Hot gap (with Cask Rail)[1]
Vacuum Drying	Procedure A	186	8.858E-06	218	8.972E-06	70.422	69.963	0.460	0.340
	Procedure B	174	8.822E-06	393	9.472E-06	70.415	70.084	0.331	0.211
	Procedure C	213	8.952E-06	504	9.708E-06	70.440	70.164	0.276	0.156
Transfer	115°F Amb. # 1	280	9.160E-06	475	9.650E-06	70.485	70.143	0.342	0.222
	115°F Amb. # 2	280	9.160E-06	475	9.650E-06	70.485	70.143	0.342	0.222
	115°F Amb. # 3	280	9.160E-06	475	9.650E-06	70.485	70.143	0.342	0.222
	115°F Amb. # 4	280	9.160E-06	475	9.650E-06	70.485	70.143	0.342	0.222
	-20°F Amb. # 1	178	8.834E-06	398	9.492E-06	70.417	70.088	0.330	0.210
	115°F Amb. # 1	280	9.160E-06	475	9.650E-06	70.485	70.143	0.342	0.222

Notes:

1. The hot diametrical gap between the canister outer diameter and the inner surfaces of the two 0.12 inch high rails welded to the cask inner diameter

Nomenclature,

$D_{CKH} = 70.35 \times [1 + \alpha_{CK} \times (T_{CKH} - 70)]$
$D_{CNH} = 69.87 \times [1 + \alpha_{CN} \times (T_{CNH} - 70)]$
T_{CKH} = Temperature of the hot cask outer structural shell
α_{CK} = Thermal expansion coefficient of the cask inner liner at T_{CKH} temperature
T_{CNH} = Temperature of the hot canister shell
α_{CN} = Thermal expansion coefficient of the canister shell at T_{CNH} temperature
D_{CKH} = Inner diameter of the hot cask inner liner at T_{CKH} temperature
D_{CNH} = Outer diameter of the hot canister shell at T_{CNH} temperature
$D_{CKH} - D_{CNH}$ = diametrical hot clearance between the inner diameter of the cask inner liner and the outer diameter of the canister shell

E. Thermal Expansion between the Length of the Canister and the Transfer Cask Cavity

The maximum length of the canister at room temperature, L_{can} = 185.75 inches.

The minimum length of the transfer cask cavity at room temperature, L_{tc}, is computed as follows.

$$L_{tc} = 186.60 \text{ in.} - 0.05 = 186.55 \text{ inches.}$$

Event	Case	T_{CKH} (°F)	α_{CK} (in/in-°F)	T_{CNH} (°F)	α_{CN} (in/in-°F)	L_{CKH} (in)	L_{CNH} (in)	L_{CKH} - L_{CNH} (in)
Vacuum Drying	Procedure A	186	8.858E-06	218	8.972E-06	186.742	185.997	0.745
	Procedure B	174	8.822E-06	393	9.472E-06	186.721	186.318	0.403
	Procedure C	213	8.952E-06	504	9.708E-06	186.789	186.533	0.256
Transfer	115° F Amb. # 1	280	9.160E-06	475	9.650E-06	186.909	186.476	0.433
	115° F Amb. # 2	280	9.160E-06	475	9.650E-06	186.909	186.476	0.433
	115° F Amb. # 3	280	9.160E-06	475	9.650E-06	186.909	186.476	0.433
	115° F Amb. # 4	280	9.160E-06	475	9.650E-06	186.909	186.476	0.433
	-20° F Amb. # 1	178	8.834E-06	398	9.492E-06	186.728	186.328	0.400
	115° F Amb. # 4	280	9.160E-06	475	9.650E-06	186.909	186.476	0.433

Nomenclature,

L_{CKH} = 186.55 × [1+ α_{CK} × (T_{CKH} - 70)]
L_{CNH} = 185.75 × [1+ α_{CN} × (T_{CNH} - 70)]
T_{CKH} = Temperature of the hot transfer cask structural shell
α_{CK} = Thermal expansion coefficient of the transfer cask structural shell at T_{CKH} temperature
T_{CNH} = Temperature of the hot canister
α_{CN} = Thermal expansion coefficient of the canister at T_{CNH} temperature
L_{CKH} = Length of the hot transfer cask cavity at T_{CKH} temperature
L_{CNH} = Length of the hot canister at T_{CNH} temperature
$L_{CKH} - L_{CNH}$ = diametrical hot clearance between the length of the transfer cask cavity and the length of the canister

3.9.1.4.5. Evaluation of Thermal Expansion Results

This evaluation shows that in all load cases, except Procedures A and B of the vacuum drying event, the minimum design cold gaps between each pair of nested components are adequate. The computation spreadsheet in Section 3.9.1.4.4.B shows that the basket will expand diametrically and make contact with the canister during Procedures A and B of the vacuum drying event. A diametrical interference of 0.115 inches between the basket and the canister will occur in Procedure A and 0.011 inches in Procedure B. Procedure A creates a more severe interference than Procedure B. The interference during Procedure A, as shown in Section 3.9.1.4.4.B is therefore selected for further analysis.

The fuel basket and canister temperatures used in the evaluation in Section 3.9.1.4.4.B are computed in Chapter 4 with a hot gap of 0.1 inches between them. However, since the hot basket expands towards the colder canister, the air gap between the basket and the canister will reduce and the temperature in the basket will drop. As this hot gap gets smaller, the average temperature of the basket will continue to drop until equilibrium is reached between the hot gap and its corresponding basket temperature.

The cold gap between the basket and the canister required for both to expand, without interference, is therefore equal to the differential expansion between the two when they are at their temperatures during contact. Chapter 4 conducts a thermal analysis of Procedure A for a case in which the basket comes into contact with the canister (no hot gap between the basket and the canister). In this case the average temperature of the basket is calculated to be 377° F and the canister 219 °F at 34 hours of operation of Procedure A. The cold canister inner diameter of 68.53 inches will expand to a hot inner diameter of 68.621 inches at the temperature of 219° F. For a hot basket outer diameter at a temperature of 377° F to expand to match this hot canister inner diameter of 68.621 inches, the cold basket outer diameter will be 68.423 inches.

Since,

$$OD_{cold\ basket} \times [1+\alpha_{377F} \times \Delta T_{basket}] = ID_{cold\ canister} \times [1 + \alpha_{219F} \times \Delta T_{canister}]$$

Or,

$$OD_{cold\ basket} \times [\ 1 + 9.408\times10^{-6}\ \text{in/in/°F} \times (377\text{-}70)°\ \text{F}]$$
$$= 68.53\ \text{in.} \times [\ 1 + 8.976\times10^{-6}\ \text{in/in/°F} \times (219\text{-}70)]\ °\text{F} = 68.621\ \text{in.}$$

$$OD_{cold\ basket} = 68.423\ \text{in.}$$

Therefore, the cold diametrical gap between the outer diameter of the cold basket and the inner diameter of the cold canister is,

$$ID_{cold\ canister} - OD_{cold\ basket} = 68.53\ \text{in.} - 68.423\ \text{in.} = 0.107\ \text{in.}$$

Therefore, in order to prevent the hot fuel basket from binding with the colder canister, a minimum cold diametrical gap of 0.107 inches between the two will have to be provided. The existing design minimum cold diametrical gap between the basket and the canister is 0.16 inches which is greater than 0.107 inches. It is concluded that a cold minimum canister inner diameter of 68.53 inches and a cold maximum basket outer diameter of 68.37 inches will not bind each other due to thermal expansions during vacuum drying Procedure A.

The temperature difference between the hot fuel basket and the hot canister during Procedure B, 241° F (= 634° F – 393° F), is much less than that during Procedure A, 396 °F (= 614° F – 218° F). It is therefore concluded that the design minimum cold gap of 0.16 inches between the canister and the fuel basket, which is adequate for Procedure A, is also adequate for vacuum drying Procedure B.

3.9.1.4.6. Thermal Expansion Analysis Conclusions

This evaluation demonstrates that adequate clearance is provided between the 32PTH DSC fuel basket and canister shell, and between the 32PTH DSC canister and the OS87H Transfer Cask to permit free thermal expansions among these components due to all specified design and service conditions.

3.9.1.5 References

1. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section II, Part D, 1998, including 2000 addenda.

2. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section III, Subsection NG, 1998 with 2000 Addenda.

3. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section III, Appendix F, 1998 with 2000 Addenda.

4. NUREG/CR-0497-Rev 2,"MATPRO-Version 11 (Revision 2), A handbook of materials properties for use in the analysis of light water reactor fuel rod behavior.

5. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section III, Division 1, Subsection NF, 1998, including 2000 Addendum.

6. Manual of Steel Construction, Eighth Edition, American Institute of Steel Construction, Inc., 1980.

7. Roark, Formulas for Stress and Strain, Sixth Edition.

8. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section III, Division 1, Subsection NB, 1998, including 2000 Addendum.

9. NUREG/CR-0481 SAND77-1872 R-7, "An Assessment of Stress-Strain Data Suitable for Finite-Element Elastic-Plastic Analysis of Shipping Containers", September 1978.

10. ANSYS User's Manual, Rev 6.0.

11. American National Standard ANSI N14.6 – 1993, "American National Standard for Radioactive Materials – Special Lifting Devices for Shipping Containers Weighing 10000 pounds (4500 kg) or More"

12. Table speeds calculation of Strength of Threads", R. C. Boucher, Product Engineering, November 27, 1961.

13. Machinery's Handbook, 24th Edition.

14. ASME B&PV Code Case N-595-3.

15. Roark, Formulas for Stress and Strain, Fourth Edition.

Table 3.9.1-1
Temperature Dependent Material Properties

Component	Material	Temp. °F	Ultimate S_u (ksi)	Yield S_y (ksi)	Allow. S_m (ksi)	E (10^6 psi)	α (10^{-6} °F^{-1})
Basket, Plates, Rails And Canister	SA-240 Stainless Steel 304	70	75.0	30.0	20.0	28.3	8.5
		200	71.0	25.0	20.0	27.6	8.9
		300	66.2	22.4	20.0	27.0	9.2
		400	64.0	20.7	18.7	26.5	9.5
		500	63.4	19.4	17.5	25.8	9.7
		600	63.4	18.4	16.4	25.3	9.8
		700	63.4	17.6	16.0	24.8	10.0

Table 3.9.1-2
SA-240, Type 304, Thermal Material Properties

Temperature (°F)	Conductivity (Btu/hr-in-°F)	Specific Heat (Btu/lbm-°F)	Density (lb/in^3)
200	0.775	0.1224	0.29
300	0.817	0.1258	0.29
400	0.867	0.1294	0.29
500	0.908	0.1317	0.29
600	0.942	0.1334	0.29
700	0.983	0.135	0.29
800	1.017	0.136	0.29
900	1.058	0.137	0.29
1000	1.100	0.139	0.29

Table 3.9.1-3
Summary of Stresses in Fuel Compartments, Rails and Canister for Transfer Loads

Loading	Component	Service Level	Stress Classification	Loads	Stress (ksi)	Allow. Stress (ksi)
Dead Weight (Cask Vert.)	Fuel Comp. & Plates	A	P_m	1g Axial	0.16	16.0
		A	$P_m + P_b$		0.16	24.0
	Rail	A	P_m		0.16	16.4
		A	$P_m + P_b$		0.16	24.6
Handling Loads (Cask Horiz.)	Fuel Comp. & Plates	A	P_m	2g Vert. + 2g Trans. + 2g Axial	8.10	16.0
		A	$P_m + P_b$		13.42	24.0
	Rail	A	P_m		4.38	16.4
		A	$P_m + P_b$		13.42	24.6
	Canister	A	P_m		1.52	17.5
		A	$P_m + P_b$		13.24	26.25
Thermal	Fuel Comp. & Plates	A	Q	115° F Amb.	4.75	48.0
	Rail	A	Q		11.16	49.2
	Fuel Comp. & Plates	A	Q	-20° F Amb.	5.15	48.0
	Rail	A	Q		11.62	49.2
	Fuel Comp. & Plates	A	Q	Vacuum Drying (Proc. A)	9.57	48.0
	Rail	A	Q		15.07	49.2
Handling Load + Normal Thermal	Fuel Comp. & Plates	A	$P_m + P_b + Q$	Primary plus Secondary	18.57	48.0
	Rails	A	$P_m + P_b + Q$		25.04	49.2
DW + Vacuum Drying Thermal	Fuel Comp. & Plates	A	$P_m + P_b + Q$	Primary plus Secondary	9.73	48.0
	Rails	A	$P_m + P_b + Q$		15.23	49.2

Table 3.9.1-4(a)
Summary of Calculated Stresses in the Fuel Basket and Canister Shell due to 75g Drop Loads

Drop Orientation	Component	Stress Category	Max. Stress (ksi)	Allowable Stress (ksi)
0° Side Drop	Fuel Compartment and Plates	P_m	29.23	44.38
		$P_m + P_b$	29.97	57.06
	Rails	P_m	25.17	44.38
		$P_m + P_b$	26.76	57.06
	Canister	P_m	4.53	44.38
		$P_m + P_b$	15.29	57.06
30° Side Drop	Fuel Compartment and Plates	P_m	35.17	44.38
		$P_m + P_b$	35.77	57.06
	Rails	P_m	24.99	44.38
		$P_m + P_b$	29.55	57.06
	Canister	P_m	5.77	44.38
		$P_m + P_b$	22.32	57.06
45° Side Drop	Fuel Compartment and Plates	P_m	21.19	44.38
		$P_m + P_b$	24.66	57.06
	Rails	P_m	20.57	44.38
		$P_m + P_b$	26.07	57.06
	Canister	P_m	4.86	44.38
		$P_m + P_b$	21.72	57.06
180° Side Drop (on Rails)	Fuel Compartment and Plates	P_m	28.26	44.38
		$P_m + P_b$	28.32	57.06
	Rails	P_m	19.89	44.38
		$P_m + P_b$	43.89	57.06
	Canister	P_m	10.36	44.38
		$P_m + P_b$	39.43	57.06
End Drop	Fuel Comp.	P_m	11.89	44.38

Table 3.9.1-4(b)
Summary of Linearized Stresses in 7/8 inch Square Bars due to 75g Side Drop Loads

Drop Orientation	Max. Nodal Stress Intensity (ksi)	Stress Category	Linearized Stress (ksi)	Allowable Stress (ksi)
0° Side Drop	32.96	P_m	17.80	44.38
		$P_m + P_b$	19.71	57.06
30° Side Drop	50.81	P_m	32.06	44.38
		$P_m + P_b$	36.42	57.06
45° Side Drop	27.61	P_m	11.68	44.38
		$P_m + P_b$	16.97	57.06
180° (on rails) Side Drop	90.01	P_m	40.53	44.38
		$P_m + P_b$	48.20	57.06

Table 3.9.1-5
Summary of Stresses in Fuel Compartments, Support Rails and Canister Shell due to Storage Loads

Loading	Component	Service Level	Stress Classification	Applied Loads	Calculated Stress (ksi)	Allowable Stress (ksi)
Dead Weight (Cask Horiz.)	Fuel Compartment	A	P_m	1g Down	1.33	16.0
		A	$P_m + P_b$		2.81	24.0
	Rail	A	P_m		0.77	16.4
		A	$P_m + P_b$		4.66	24.6
	Canister Shell	A	P_m		0.40	17.5
		A	$P_m + P_b$		4.86	26.25
Seismic Loads (Cask Horiz.)	Fuel Compartment	A	P_m	0.65g Axial + 0.65g Trans.+ 1.30g Vertical	11.28	16.0
		A	$P_m + P_b$		16.04	24.0
	Rail	A	P_m		4.55	16.4
		A	$P_m + P_b$		10.98	24.6
	Canister Shell	A	P_m		1.52	17.5
		A	$P_m + P_b$		11.03	26.25
Thermal	Fuel Compartment	A	Q	115° F Amb. (with fins)	5.11	48.0
	Rail	A	Q		12.87	49.2
	Fuel Compartment	A	Q	-20° F Amb. (with fins)	5.52	48.0
	Rail	A	Q		13.58	49.2
	Fuel Compartment	A	Q	HSM Vent Blockage	5.26	45.6
	Rail	A	Q		11.34	48.6
Seismic Load + Normal Thermal	Fuel Compartment	A	$P_m + P_b + Q$	Primary plus Secondary	21.56	48.0
	Rails	A	$P_m + P_b + Q$		24.56	49.2
DW + Vent Blockage Thermal	Fuel Compartment	A	$P_m + P_b + Q$	Primary plus Secondary	8.07	45.6
	Rails	A	$P_m + P_b + Q$		16.0	48.6

Table 3.9.1-6
Temperature Dependent Coefficients of Thermal Expansion

Component	Material	Temperature (°F)	α (10^{-6} in/in/°F)
Basket, Canister, And Transfer Cask	SA-240 Gr. 304	70	8.50
		100	8.60
		150	8.80
		200	8.90
		250	9.10
		300	9.20
		350	9.30
		400	9.50
		450	9.60
		500	9.70
		550	9.80
		600	9.80
		650	9.90
		700	10.0
		750	10.0
		800	10.1
		850	10.1
Fuel Cladding	Zircaloy	700	2.70*
		730	2.73*
		750	2.79*
		810	2.70*

*Axial thermal expansion coefficient is taken from Reference 4.

Table 3.9.1-7
ASME Subsection NB Code Allowable Stresses for the 32PTH DSC Canister (for Transfer Loads)

Loading Condition	Stress Category		Stress Limits	Material	Allowable Stress (ksi.)
Normal and Off-Normal Conditions[1]	Elastic Analysis	Membrane Stress, P_m	S_m	SA-240 Type 304	17.5
				SA-182 Type F304	17.5
		Membrane + Bending Stress, $P_L + P_b$	1.5 S_m	SA-240 Type 304	26.25
				SA-182 Type F304	26.25
		Membrane + Bending Stress + Secondary stress $P_L + P_b + Q$	3 S_m	SA-240 Type 304	52.5
				SA-182 Type F304	52.5
Accident Conditions	Elastic Analysis	Membrane Stress, P_m	Lesser of 2.4 S_m and 0.7 S_u	SA-240 Type 304	42
				SA-182 Type F304	41.44
		Membrane + Bending Stress, $P_L + P_b$	Lesser of 3.6 S_m and S_u	SA-240 Type 304	63
				SA-182 Type F304	59.2
	Elastic / Plastic Analysis	Membrane Stress, P_m	Greater of 0.7 S_u and $[S_y + 1/3 (S_u - S_y)]$	SA-240 Type 304	44.38
				SA-182 Type F304	41.44
		Membrane + Bending Stress, $P_L + P_b$	0.9 S_u	SA-240 Type 304	57.06
				SA-182 Type F304	53.28

Notes : 1. Normal condition allowable stresses are conservatively used for off-normal loads.

Table 3.9.1-8
ASME Code Allowable Stresses for the 32PTH DSC Canister for Storage Loads

Loading Condition	Stress Category	Stress Limits	Material	Allowable Stress (ksi.)
Normal Conditions [1] (At max. temperature of 450° F for both cases of ambient at -20° F and 115° F)	Membrane Stress (P_m)	S_m	SA-240 Type 304	18.1
			SA-182 Grade F304	18.1
	Membrane + Bending Stress ($P_L + P_b$)	1.5 S_m	SA-240 Type 304	27.15
			SA-182 Grade F304	27.15
	Membrane + Bending Stress + Secondary stress ($P_L + P_b + Q$)	3 S_m	SA-240 Type 304	54.3
			SA-182 Grade F304	54.3
Accident Conditions [2] (At max. temperature of 550° F for case of Blocked Vent)	Membrane Stress (P_m)	Lesser of 2.4 S_m and 0.7 S_u	SA-240 Type 304	40.68
			SA-182 Grade F304	40.68
	Membrane + Bending Stress ($P_L + P_b$)	Lesser of 3.6 S_m and S_u	SA-240 Type 304	61.02
			SA-182 Grade F304	61.02

Notes:
1. Stress limit per Reference 8
2. Stress limits per Reference 3

Table 3.9.1-9
32PTH DSC Canister Load Combinations during Transfer

Loading	Canister w/Transfer Cask Orientation	Service Level	Load for Analysis	Load Combinations	Analyzed Load Case No.	ANSYS Model
Dead Weight	Vertical[1]	A	1g Down (Axial)	1g Down + 15 psig Ext. Press. + Thermal (Vacuum Dry)	1	2-D
External Pressure	Vertical[1]	A	15 psig			
Thermal	Vertical[1]	A	Vacuum Dry			
Dead Weight	Horizontal[2]	A	2*g* Axial + 2*g* Trans. + 2*g* Vertical	A = 2*g* Axial + 2*g* Trans. + 2*g* Vertical A+ 30 psig Int. Pressure + Thermal (115° F)	2	2-D
Handling load in Transfer Cask	Horizontal[2]	A		A+ 15 psig Ext. Pressure + Thermal (-20° F)	3	2-D
Internal Pressure	Horizontal[2]	A	30 psig[6]	Pressure Stress	[2][5]	2-D
Ext. Press.	Horizontal[2]	A	15 psig	Pressure Stress	[3][5]	2-D
Thermal	Horizontal[2]	A	Thermal Stress (-20° F Ambient)	Thermal Stress	[3][5]	2-D
Thermal	Horizontal[2]	A	Thermal Stress (115° F Ambient)	Thermal Stress	[2][5]	2-D
Internal Pressure	Horizontal	D	120 psig[3]	Pressure Stress	4	2-D
External Pressure	Horizontal	D	25 psig[4]	Pressure Stress	5	2-D
Side Drop	Horizontal	D	75*g* Multiple Orientations (0°, 30°, 45°, impact on two rails, impact on one rails) Drop angles are enveloped by 0° (no rail) and 180° (two rails)	75*g* side drop at 0° (no rail) + 30 psig Int. Press. of Top / Bottom ends	6 / 7	3-D
				75*g* side drop at 180° (two rails) + 30 psig Int. Press. of Top / Bottom ends	8 / 9	3-D
				75*g* side drop at 0° (no rail) + 15 psig Ext. Press. of Top / Bottom ends	10 / 11	3-D
				75*g* side drop at 180° (two rails) + 15 psig Ext. Press. of Top / Bottom ends	12 / 13	3-D
Corner Drop	Horizontal	D	Enveloped by 75*g* Side Drop and 75*g* End Drop			
End Drop	Vertical	D	75*g* End Drop	75*g* Top/Bottom + 30 psig Int. Pressure	14 / 15	2-D
				75*g* Top/Bottom + 15 psig Ext. Pressure	16 / 17	2-D

Notes:
1. Transfer cask supported at the bottom.
2. Transfer cask supported at 4 trunnion location.
3. Under accident fire condition.
4. Under accident flood condition.
5. [#] indicates this individual load case is enveloped in the analyzed load case No. #
6. From Chapter 4, Table 4-10, the maximum normal operating pressure is 6.4 psig during transfer operation. However, for the canister stress analysis, 15 psig is conservatively used for the structural evaluation of the canister.

Table 3.9.1-10
32PTH DSC Canister Load Combinations during Lifting, Testing, and Hydraulic Loads

Loading	Canister w/Transfer Cask Orientation	Service Level	Load for Analysis	Load Combinations	Analyzed Load Case No.	ANSYS Model
Dead Weight	Horizontal	A	1g	1g + 18 psig Int. Pressure + 155 kips Axial Loads	18	2-D
Test Pressure	Horizontal	A	18 psig(3)			
Seal Plate Axial Load	Horizontal	A	155 kips			
Hydraulic Loads(1) (2) (Push/Pull)	Horizontal	A	80 / 60 kips	30 psig Int. Pressure + 80 kips push / 60 kips pull + Thermal (115° F)	19 / 20	2-D
Hydraulic Loads(1) (2) (Push/Pull)	Horizontal	C	80 / 80 kips	30 psig Int. Pressure + 80 kips + Thermal (115° F)	21 / 22	2-D
Hydraulic Loads(1) (2) (Push/Pull)	Horizontal	D	110 / 110 kips	30 psig Int. Pressure + 110 kips	23 / 24	2-D
Lifting	Vertical	A	6g	6g	25	Manual

Notes:
1. The hydraulic push loads are applied at the canister bottom surface within the grapple ring support.
2. The hydraulic pull loads are applied at the inner surface of the grapple ring.
3. From Chapter 4, Table 4-10, the maximum normal operating pressure is 6.4psig during transfer operation. The test pressure is 1.5 x MNOP = 1.5 x 6.4 = 9.6 psig. The canister is conservatively using 18 psig as the test pressure.

Table 3.9.1-11
Summary of Calculated Stresses for Testing Condition Loads

Load Case	Combination of Loads	Canister Orientation	Stress Limits (NB-3226) [1]	
			$P_m < 0.8\,S_y$ ($0.8\,S_y = 24{,}000$ psi)	$P_m + P_b < 1.35\,S_y$ ($1.35\,S_y = 40{,}500$ psi)
18	DW + 18 psig Int. Press. + 155 kip Axial Load	Horizontal	14,148 psi [2]	20,401 psi [3]

Notes:
1. Yield stress, $S_y = 30{,}000$ psi, is taken at test temperature of 100° F for both material SA-240 Gr.304 and SA-182 F304.
2. P_m =14,143 psi + 5 psi (Deadweight, in Load Case 18) = 14,148 psi
3. $P_m + P_b$ = 14,143 psi + 6,258 psi (Dead weight, in Load Case 18) = 20,401 psi

Table 3.9.1-12
Summary of Calculated Stress for Normal and Off-Normal Condition Transfer Loads

Load Case	Combination Of Loads	Canister Orientation	Stress Intensity Limits (NB-3220) [(1)]		
			$P_m < S_m$ (S_m = 17,500 psi)	$P_L + P_b < 1.5\,S_m$ (1.5S_m=26,250 psi)	$P_L + P_b + Q < 3\,S_m$ (3 S_m = 52,500 psi)
1	1g down + 15 psig Ext. Press. + Vac. Dry Thermal	Vertical	1,637	1,637	18,720
2	Handling 2g + 30 psig Int. Press. + Thermal (115° F)	Horizontal	1,516 + 3,332 = 4,848	13,239 + 3,332 = 16,571	13,239 + 36,219 = 49,458
3	Handling 2g's + 15 psig Ext. Press. + Thermal (-20° F)	Horizontal	1,516 + 1,666 = 3,182	13,239 + 1,666 = 14,905	13,239 + 35,001 = 48,240
19	30 psig Int. Press. + 80 kip push + Thermal (115° F)	Horizontal	7,238	7,238	34,916
20	30 psig Int. Press. + 60 kip pull + Thermal (115° F)	Horizontal	11,484	11,484	36,753
21	30 psig Int. Press. + 80 kip push + Thermal (115° F)	Horizontal	7,238	7,238	34,916
22	30 psig Int. Press. + 80 kip pull + Thermal (115° F)	Horizontal	5,249	13,790	36,931

Notes:
1. Design stress intensity, S_m is taken at 500° F of material SA-240 Gr.304 and SA-182 F304.

Table 3.9.1-13
Summary of Calculated Stresses for Accident Condition Transfer Loads (Elastic Analysis)

Load Case	Combination Of Loads	Canister Orientation	Stress Intensity Limits [1]	
			$P_m < 0.7\,S_u$ [2] (0.7 S_u = 41,440 psi)	$P_L + P_b < S_u$ [3] (S_u = 59,200 psi)
4	120 psig internal pressure under fire accident	Horizontal	13,329 [4]	13,329 [4]
5	25 psig external pressure under flood accident	Horizontal	2,777	2,777
14	Top End Drop 75g + 30 psig Int. Press.	Vertical	15,539	15,539
15	Bottom End Drop 75g + 30 psig Int. Press.	Vertical	16,492	16,492
16	Top End Drop 75g + 15 psig Ext. Press.	Vertical	18,545	18,545
17	Bottom End Drop 75g + 15 psig Ext. Press.	Vertical	19,956	19,956
23	30 psig Int. Press. + 110 kip push + Thermal (115° F)	Horizontal	9,854	9,854
24	30 psig Int. Press. + 110 kip pull + Thermal (115° F)	Horizontal	21,025	21,025

Notes:
1. See Table 3.9.1-7 for allowable stress intensities. Lesser values of allowable stresses for material SA-182 Type F304 are also used for material SA-240 Type 304.
2. 0.7 S_u is the lesser value of 2.4 S_m and 0.7 S_u, which is specified as Stress intensity limit in Reference 3.
3. S_u is the lesser value of 3.6 S_m and S_u, which is specified as Stress intensity limit in Reference 3.
4. For metal temperature of 800° F under accident fire, allowable stress intensities, $P_m < 2.4\,S_m$ (2.4×15,200 psi = 36,480 psi), $P_L+P_b < 3.6 \times S_m$ (3.6×15,200psi = 54,720 psi).

Table 3.9.1-14
Summary of Stresses for Accident Condition Transfer Loads (Elastic / Plastic Analysis)

Load Case	Combination Of Loads	Canister Orientation	Stress Intensity Limits [1]	
			$P_m < 0.7\,S_u$ [2] ($0.7\,S_u = 41{,}440$ psi)	$P_L + P_b < 0.9\,S_u$ [3] ($0.9S_u = 53{,}280$ psi)
6	Side Drop 75*g* at 0° + 30 psig Int. Press. (Top End)	Horizontal	27,990	27,990
7	Side Drop 75*g* at 0° + 30 psig Int. Press. (Bottom End)	Horizontal	19,976	19,976
8	Side Drop 75*g* at 180° + 30 psig Int. Press. (Top End)	Horizontal	28,869	28,869
9	Side Drop 75*g* at 180° + 30 psig Int. Press. (Bottom End)	Horizontal	22,666	22,666
10	Side Drop 75*g* at 0° + 15 psig Ext. Press. (Top End)	Horizontal	28,402	28,402
11	Side Drop 75*g* at 0° + 15 psig Ext. Press. (Bottom End)	Horizontal	19,381	19,381
12	Side Drop 75*g* at 180° + 15 psig Ext. Press. (Top End)	Horizontal	29,354	29,354
13	Side Drop 75*g* at 180° + 15 psig Ext. Press. (Bottom End)	Horizontal	22,650	22,650

Notes:

1. See Table 3.9.1-7 for allowable stress intensities. Lesser value of allowable stresses for material SA-182 Type F304 was also used for material SA-240 Type 304.
2. $0.7\,S_u$ is the greater value of $[S_y + 1/3\,(S_u - S_y)$ and $0.7\,S_u$, which is specified as stress intensity limit in Reference 3.
3. $0.9\,S_u$ is the value specified as Stress intensity limit in Reference 3.

Table 3.9.1-15
Summary of Calculated Stress at the End Closure Welds for Testing Condition Loads

Load Case	Combination of Loads	Canister Orientation	Stress Limits (NB-3226) (1)	
			$P_m < 0.8\, S_y$ (0.8 S_y = 24,000 psi)	$P_m + P_b < 1.35\, S_y$ (1.35 S_y = 40,500 psi)
18	DW + 18 Psig Int. Press. + 155 kip Axial Load	Horizontal	No closure welds	No closure welds

Notes:
1. Yield stress, S_y =30,000 psi, is taken at test temperature of 100° F for both material SA-240 Gr.304 and SA-182 F304.

Table 3.9.1-16
Summary of Calculated Stress at the End Closure Welds for Normal and Off-Normal Condition Transfer Loads

Load Case	Combination Of Loads	Canister Orientation	Stress Intensity Limits (NB-3220) [1]		
			$P_m < 0.8S_m$ ($0.8S_m = 14{,}000$ psi)	$P_L + P_b < 0.8(1.5\ S_m)$ ($1.2S_m$=21,000 psi)	$P_L + P_b + Q < 0.8(3\ S_m)$ ($2.4\ S_m = 42{,}000$ psi)
1	1g down + 15 psig Ext. Press. + Vac. Dry Thermal	Vertical	1,341	1,341	2,114
2	Handling 2g's + 30 psig Int. Press. + Thermal (115° F)	Horizontal	1,516 + 3,134 = 4,650	13,239 + 3,134 = 16,373	13,239 + 4,160 = 17,399
3	Handling 2g's + 15 psig Ext. Press. + Thermal (-20° F)	Horizontal	1,516 + 1,234 = 2,750	13,239 + 1,234 = 14,473	13,239 + 2,318 = 15,557
19	30 psig Int. Press.+80 kip push + Thermal (115° F)	Horizontal	3,123	3,123	3,602
20	30 psig Int. Press. + 60 kip pull + Thermal (115° F)	Horizontal	3,134	3,134	3,646
21	30 psig Int. Press.+80 kip push + Thermal (115° F)	Horizontal	3,123	3,123	3,602
22	30 psig Int. Press.+ 80 kip pull + Thermal (115° F)	Horizontal	3,134	3,134	3,646

Notes:
1. Design stress intensity, S_m is taken at 500° F of material SA-240 Gr.304 and SA-182 F304.

Table 3.9.1-17
Summary of Calculated Stresses at the End Closure Welds for Accident Condition Transfer Loads (Elastic Analysis)

Load Case	Combination Of Loads	Canister Orientation	Stress Intensity Limits [1]	
			$P_m < 0.8\,(0.7\,S_u)$ [2] $(0.8 \times 0.7\,S_u = 33{,}152$ psi)	$P_L + P_b < 0.8\,S_u$ [3] $(0.8\,S_u = 47{,}360$ psi)
4	120 psig internal pressure under fire accident	Horizontal	12,379 [4]	12,379 [4]
5	25 psig external pressure under flood accident	Horizontal	2,051	2,051
14	Top End Drop 75g + 30 psig Int. Press.	Vertical	5,641	5,641
15	Bottom End Drop 75g + 30 psig Int. Press.	Vertical	8,714	8,714
16	Top End Drop 75g + 15 psig Ext. Press.	Vertical	7,916	7,916
17	Bottom End Drop 75g + 15 psig Ext. Press.	Vertical	12,319	12,319
23	30 psig Int. Press.+110 kip push + Thermal (115° F)	Horizontal	3,132	3,132
24	30 psig Int. Press. + 110 kip pull + Thermal (115° F)	Horizontal	3,134	3,134

Notes:
1. See Table 3.9.1-7 for allowable stress intensities. Lesser values of allowable stresses for material SA-182 Type F304 was also used for material SA-240 Type 304.
2. $0.7\,S_u$ is the lesser value of $2.4\,S_m$ and $0.7\,S_u$, which is specified as Stress intensity limit in Reference 3.
3. S_u is the lesser value of $3.6\,S_m$ and S_u, which is specified as Stress intensity limit in Reference 3.
4. For metal temperature of 800° F under accident fire, allowable stress intensities, $P_m < 0.8 \times 2.4\,S_m$ $(0.8 \times 2.4 \times 15{,}200$ psi $= 29{,}184$ psi), $P_L+P_b < 0.8 \times 3.6 \times S_m$ $(0.8 \times 3.6 \times 15{,}200$psi $= 43{,}776$ psi).

Table 3.9.1-18
Summary of Calculated Stresses at End Closure Welds for Accident Condition Transfer Loads
(Elastic/Plastic Analysis)

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits (Ref. 9) (1)	
			$P_m < 0.8\ (0.7\ S_u)$ (2) $(0.8 \times 0.7\ S_u = 41{,}440$ psi)	$P_L + P_b < 0.8\ (0.9\ S_u)$ (3) $(0.8 \times 0.9 S_u = 53{,}280$ psi)
6	Side Drop 75g at 0° + 30 psig Int. Press. (Top End)	Horizontal	27,566	27,566
8	Side Drop 75g at 180° + 30 psig Int. Press. (Top End)	Horizontal	27,220	27,220
10	Side Drop 75g at 0° + 15 psig Ext. Press. (Top End)	Horizontal	27,493	27,493
12	Side Drop 75g at 180° + 15 psig Ext. Press. (Top End)	Horizontal	27,306	27,306

Note:
1. See Table 3.9.1-7 for allowable stress intensities. Lesser value of allowable stresses for material SA-182 Type F304 was also used for material SA-240 Type 304.
2. $0.7\ S_u$ is the greater value of $[S_y + 1/3\ (S_u - S_y)$ and $0.7\ S_u$, which is specified as stress intensity limit in Reference 3.
3. $0.9\ S_u$ is the value specified as Stress intensity limit in Reference 3.

Table 3.9.1-19
32PTH DSC Canister Load Combinations during Storage

Loading	Canister Orientation	Service Level	Load	Enveloped Load for Analysis	Load Combinations
Dead Weight (DW)	Horizontal(1)	A	1g Down	.65g Axial + .65g Trans. + 1.3g Vertical (See Note 2)	.65g Axial + .65g Trans. + 1.3g Vertical Down
Seismic Loads	Horizontal(1)	C (2)	0.43g Axial + 0.43g Trans. +0.20g Vertical		.65g Axial + .65g Trans. + 1.3g Vertical Down + 30 psig + Thermal (115° F)
					.65g Axial + .65g Trans. + 1.3g Vertical Down + 30 psig + Thermal (-20° F)
Internal Pressure	Horizontal(1)	A	15 psig	30 psig	Pressure
Thermal	Horizontal(1)	A	Thermal (-20° F ambient)	Thermal (-20° F ambient)	Thermal
Thermal	Horizontal(1)	A	Thermal (115° F ambient)	Thermal (115° F ambient)	Thermal
Thermal	Horizontal(1)	D (2)	Blocked Vent	Blocked Vent	1g Down + 70 psig Int. Pressure + Thermal (Blocked Vent)
Internal Pressure	Horizontal(1)	D (2)	< 67 psig due to Blocked Vent	Enveloped by 70 psig inter pressure	
Flood	Horizontal(1)	D (2)	50 ft Water (≈22 psig)	Enveloped by 30 psig external pressure design	

Notes:
1. Canister supported at HSM rails and axial restrained by the seismic restraint devices.
2. Levels C and D loads are conservatively treated as Level A loads and so evaluated.

Table 3.9.1-20
Summary of Calculated Stresses for Normal and Accident Condition Loads (1)
(Canister in horizontal storage position)

Load Case	Applied Loads	Stress Intensity Limits			
		$P_m < S_m$ (S_m = 18.1 ksi)	$P_L + P_b < 1.5S_m$ ($1.5S_m$ = 27.15 ksi)	Q	$P_L + P_b + Q < 3\,S_m$ ($3\,S_m$ = 54.3 ksi)
1	Deadweight (1g down) (2)	0.4	4.86	---	---
2	30 psig Internal Pressure	3.33	3.33	---	---
3	Seismic (.65g Axial + .65g Trans. + 1.3g Vertical Down) (3)	1.52	11.03	---	---
4	Thermal (-20° F)	---	---	20.60	---
5	Thermal (115° F)	---	---	18.48	---
6	Thermal (Blocked Vent)	---	---	13.84	---
7	Accident 70 psig Internal Pressure	7.77	7.77	---	---
8	Accident Flood (Enveloped by ext. pressure of 30 psig)	3.33	3.33	---	---
3 + 4	30 psig Internal Pressure + Seismic + Thermal (-20° F)	3.33 + 1.52 = 4.85	3.33 + 11.03 = 14.36	20.60	3.33 + 11.03 + 20.60 = 34.96
3 + 5	30 psig Internal Pressure + Seismic + Thermal (115° F)	3.33 + 1.52 = 4.85	3.33 + 11.03 = 14.36	18.48	3.33 + 11.03 + 18.48 = 32.84
1 + 6 + 7	Deadweight + 70 psig Int. Pressure + Thermal (Blocked Vent) (4)	0.4 + 7.77 = 8.17	4.86 + 7.77 = 12.63	13.84	4.86 + 7.77 + 13.84 = 26.47
1 + 8	Deadweight + Flood (30 psig ext. pressure)	0.4 + 3.33 = 3.73	4.86 + 3.33 = 8.19	---	---

Notes:
1. Accident loads are conservatively treated as Normal loads since the allowable stress intensities for accident loads are higher than those for normal loads as indicated in Table 3.9.1-8.
2. The maximum stress intensities are obtained from Table 3.9.1-5.
3. Seismic load includes 1g down Deadweight. The maximum stress intensities are obtained from Table 3.9.1-5.
4. Seismic event is assumed not to occur with accident event of blocked vent.

Table 3.9.1-21
Summary of Calculated Stresses under Normal and Accident Conditions of Storage Loads (At End Closure Welds) (1)

Load Case	Combination of Loads	Stress Intensity Limits $P_m < 0.8S_m$ [$0.8S_m$ = 14.48 ksi]	$P_L + P_b$ < 0.8 ($1.5S_m$) [0.8 ($1.5S_m$) = 21.72 ksi]	Q	$P_L + P_b + Q$ < 0.8($3S_m$) [0.8 ($3S_m$) = 43.44 ksi]
1	Deadweight (1g down)(2)	0.4	4.86	---	---
2	30 psig Internal Pressure	3.33	3.33	---	---
3	Seismic (.65g Axial + .65g Trans. + 1.3g Vertical Down) (3)	1.52	11.03	---	---
4	Thermal (-20° F)	---	---	20.60	---
5	Thermal (115° F)	---	---	18.48	---
6	Thermal (Blocked Vent)	---	---	13.84	---
7	Accident 70 psig Internal Pressure	7.77	7.77	---	---
8	Accident Flood (Enveloped by ext. pressure of 30 psig)	3.33	3.33	---	---
3 + 4	30 psig Internal Pressure + Seismic + Thermal (-20° F)	3.33 + 1.52 = 4.85	3.33 + 11.03 = 14.36	20.60	3.33 + 11.03 + 20.60 = 34.96
3 + 5	30 psig Internal Pressure + Seismic + Thermal (115° F)	3.33 + 1.52 = 4.85	3.33 + 11.03 = 14.36	18.48	3.33 + 11.03 + 18.48 = 32.84
1 + 6 + 7	Deadweight + 70 psig Internal Pressure + Thermal (Blocked Vent) (4)	0.4 + 7.77 = 8.17	4.86 + 7.77 = 12.63	13.84	4.86 + 7.77 + 13.84 = 26.47
1 + 8	Deadweight + Flood (30 ext. pressure)	0.4 + 3.33 = 3.73	4.86 + 3.33 = 8.19	---	---

Notes:
1. Accident loads are conservatively treated as normal condition loads since the allowable stress intensities for accident loads are higher than those for normal loads as indicated in Table 3.9.1-8.
2. The maximum stress intensities are obtained from Table 3.9.1-5.
3. Seismic load includes 1g down Deadweight. The maximum stress intensities are obtained from Table 3.9.1-5.
4. Seismic event is assumed not to occur with accident event of blocked vent

Table 3.9.1-22
ASME Code Allowable Stresses for the Alternate Canister Bottom Assembly Design (Subsection NF)

Loading Condition	Stress Category	Stress Limit	Material	Allowable Stress (ksi.)
Normal Conditions (1)	Membrane Stress Intensity, P_m	S_m	SA-240 Type 304	17.5
			SA-182 Type F304	17.5
			SA-36	19.3
	Membrane + Bending Stress Intensity, $P_m + P_b$	1.5 S_m	SA-240 Type 304	26.25
			SA-182 Type F304	26.25
			SA-36	28.95
	Primary + Secondary Stress Intensity, $P_m + P_b + Q$	2 S_y	SA-182 Type F304	38.8
			SA-182 Type F304	38.8
			SA-36	58.6
Off-Normal Condition (2)	Membrane Stress Intensity, P_m	1.5 S_m	SA-240 Type 304	26.25
			SA-182 Type F304	26.25
			SA-36	28.95
	Membrane + Bending Stress Intensity, $P_m + P_b$	2.25 S_m	SA-240 Type 304	39.38
			SA-182 Type F304	39.38
			SA-36	43.43
Accident Conditions (3)	Membrane Stress Intensity, P_m	The largest of 1.2 S_y, 1.5 S_m, and 0.7 S_u	SA-240 Type 304	44.38
			SA-182 Type F304	41.44
			SA-36	40.6
	Membrane + Bending Stress Intensity, $P_m + P_b$	The largest of 1.8 S_y, 2.25 S_m, and 1.0 S_u	SA-240 Type 304	63.4
			SA-182 Type F304	59.2
			SA-36	58

Notes:
1. Service Level A limits per Paragraph NF-3221.2 of Reference 5.
2. Service Level C limits per Paragraph NF-3221.2 of Reference 5.
3. Service Level D limits per Paragraph F-1332 of Reference 3.

Table 3.9.1-23
ASME Code Allowable Stresses for the Weld between the Canister Shell and the Bottom Outer Cover as per Subsection NF

Loading Condition	Stress Category	Stress Limit	Material	Allowable Stress (ksi.)
Normal Conditions (1)	Membrane Stress Intensity, P_m	S_m	SA-240 Type 304	17.5
	Membrane + Bending Stress Intensity, $P_m + P_b$	1.5 S_m	SA-240 Type 304	26.25
	Tensile stress normal to the effective weld throat	0.3 S_u	SA-240 Type 304	19.02
Off-Normal Condition (2)	Membrane Stress Intensity, P_m	1.5 S_m	SA-240 Type 304	26.25
	Membrane + Bending Stress Intensity, $P_m + P_b$	2.25 S_m	SA-240 Type 304	39.375
	Tensile stress normal to the effective weld throat	1.5 × 0.3 S_u	SA-240 Type 304	28.53
Accident Conditions (3)	Membrane Stress Intensity, P_m	The largest of 1.2 S_y, 1.5 S_m, and 0.7 S_u	SA-240 Type 304	44.38
	Membrane + Bending Stress Intensity, $P_m + P_b$	The largest of 1.8 S_y, 2.25 S_m, and 1.0 S_u	SA-240 Type 304	63.4
	Tensile stress normal to the effective weld throat	2 × 0.3 S_u (4)	SA-240 Type 304	38.04

Notes:
1. Service Level A limits per Paragraph NF-3226.2 of Reference 5.
2. Service Level C limits per Paragraph NF-3226.2 of Reference 5.
3. Service Level D limits per Paragraph F-1332 of Reference 3.
4. Service Level D limits per Paragraph F-1334 of Reference 3.

Table 3.9.1-24
Summary of Calculated Stresses in the Alternate Canister Design for Normal Condition Loads (Subsection NB components)

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits (NB-3222) [1]		
			$P_m < S_m$ (S_m = 17,500 psi)	$P_L+P_b < 1.5\ S_m$ ($1.5S_m$= 26,250 psi)	$P_L+P_b+Q < 3\ S_m$ ($3\ S_m$ = 52,500 psi)
1	1g down + 15 psig Ext. Press. + Vac. Dry Thermal	Vertical	1,657	1,657	14,668
2	Handling 2g + 30 psig Int. Press. + Thermal (115° F)	Horizontal	1,516 + 3,831 = 5,347 [2]	13,239 + 3,831 = 17,070 [2]	13,239 + 35,266 = 48,505 [2]
3	Handling 2g + 15 psig Ext. Press. + Thermal (-20° F)	Horizontal	1,516 + 1,772 = 3,288 [2]	13,239 + 1,772 = 15,011 [2]	13,239 + 27,619 = 40,858 [2]
19	30 psig Int. Press. + 80 kip push + Thermal (115° F)	Horizontal	7,238	7,238	33,189
20	30 psig Int. Press. + 60 kip pull + Thermal (115° F)	Horizontal	4,115	4,115	35,029

Notes:
1. Design stress intensity, S_m is taken at 500° F of material SA-240 Gr.304 and SA-182 F304; see Table 3.9.1-97 for allowable stresses.
2. Maximum stress intensity in the canister due to pressure and thermal loads from this evaluation are conservatively added to the maximum stress intensity due to the 2g transfer load from Table 3.9.1-3.

Table 3.9.1-25
Summary of Calculated Stresses in the Alternate Canister Design for Normal Condition Loads (Subsection NF components)

Load Case	Combination Of Loads	Canister Orientation	Stress Intensity Limits (NF-3221.2) [1]		
			$P_m < S_m$ (S_m = 17,500 psi)	$P_m + P_b < 1.5\ S_m$ ($1.5S_m$=26,250 psi)	$P_m + P_b + Q < 2\ S_y$ ($2\ S_y$ =38,800 psi)
1	1g down + 15 psig Ext. Press. + Vac. Dry Thermal	Vertical	331	331	6,369
2	Handling 2g's + 30 psig Int. Press. + Thermal (115 °F)	Horizontal	1,516 + 4,067 = 5,583 [2]	13,239 + 4,067 = 17,306 [2]	13,239 + 4,729 = 17,968 [2]
3	Handling 2g's + 15 psig Ext. Press. + Thermal (-20 °F)	Horizontal	1,516 + 540 = 2,056 [2]	13,239 + 540 = 13,779 [2]	13,239 + 12,448 = 25,687 [2]
19	30 psig Int. Press. + 80 kip push + Thermal (115 °F)	Horizontal	2,499	2,499	6,394
20	30 psig Int. Press. + 60 kip pull + Thermal (115 °F)	Horizontal	10,410 [3]	24,790 [3]	30,340

Notes:
1. Design stress intensity (S_m) and yield stress (S_y) are taken at 500° F for all materials, see Table 3.9.1-22 for allowable stresses.
2. Maximum stress intensity in the canister due to pressure and thermal loads from this evaluation are conservatively added to the maximum stress intensity due to the 2*g* transfer load from Table 3.9.1-3.
3. Linearized stress intensities through the thickness of the component plates.

Table 3.9.1-26
Summary of Calculated Stresses in the Alternate Canister Design for Normal Condition Loads (Subsection NF welds)

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits [1] (NF-3226.2) $S_x < 0.3\,S_u$ ($0.3\,S_u$ = 19,020 psi)
1	1g down + 15 psig Ext. Press. + Vac. Dry Thermal	Vertical	0
2	Handling 2g + 30 psig Int. Press. + Thermal (115° F)	Horizontal	13,239 + 1,502 = 14,741 [2]
3	Handling 2g's + 15 psig Ext. Press. + Thermal (-20° F)	Horizontal	13,239 + 23 = 13,262 [2]
19	30 psig Int. Press. + 80 kip push + Thermal (115° F)	Horizontal	152
20	30 psig Int. Press. + 60 kip pull + Thermal (115° F)	Horizontal	12,498

Notes:
1. Tensile strength, S_u, is taken at 500° F for material SA-240 Gr.304, see Table 3.9.1-23 for allowable stresses.
2. Maximum stress intensity in the canister due to pressure and thermal loads from this evaluation are conservatively added to the maximum stress intensity due to the 2g transfer load from Table 3.9.1-3.

Table 3.9.1-27
Summary of Calculated Stresses in the Alternate Canister Design at the End Closure Welds for Normal Condition Loads

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits (1) (NB-3222)		
			$P_m < 0.8S_m$ ($0.8S_m$ = 14,000 psi)	$P_L + P_b < 0.8(1.5\ S_m)$ ($1.2S_m$=21,000 psi)	$P_L+P_b+Q< 0.8(3\ S_m)$ ($2.4\ S_m$ = 42,000 psi)
1	1g down + 15 psig Ext. Press. + Vac. Dry Thermal	Vertical	1,341	1,341	2,112
2	Handling 2*g* + 30 psig Int. Press. + Thermal (115° F)	Horizontal	1,516 + 3,134 = 4,650 (2)	13,239 + 3,134 = 16,373 (2)	13,239 + 3,646 = 16,885 (2)
3	Handling 2*g* + 15 psig Ext. Press. + Thermal (-20° F)	Horizontal	1,516 + 1,232 = 2,748 (2)	13,239 + 1,232 = 14,471 (2)	13,239 + 2,247 = 15,486 (2)
19	30 psig Int. Press.+80 kip push + Thermal (115° F)	Horizontal	3,123	3,123	3,602
20	30 psig Int. Press. + 80 kip pull + Thermal (115° F)	Horizontal	3,110	3,110	3,660

Notes:
1. Design stress intensity, S_m is taken at 500° F for material SA-240 Gr.304; see Table 3.9.1-7 for allowable stresses.
2. Maximum stress intensity in the canister due to pressure and thermal loads from this evaluation are conservatively added to the maximum stress intensity due to the 2*g* transfer load from Table 3.9.1-3.

Table 3.9.1-28
Summary of Calculated Stresses in the Alternate Canister Design for Off-Normal Condition Loads (Subsection NB components)

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits [1] (NB-3222)		
			$P_m < S_m$ (S_m = 17,500 psi)	$P_L + P_b < 1.5\ S_m$ ($1.5S_m$ = 26,250 psi)	$P_L+P_b+Q<3S_m$ ($3\ S_m$ = 52,500 psi)
21	30 psig Int. Press. + 80 kip push + Thermal (115° F)	Horizontal	7,238	7,238	33,819
22	30 psig Int. Press. + 80 kip pull + Thermal (115° F)	Horizontal	4,223	4,223	34,905

Notes:
1. Design stress intensity, S_m is taken at 500° F for material SA-240 Gr.304, see Table 3.9.1-7 for allowable stresses.

Table 3.9.1-29
Summary of Calculated Stresses in the Alternate Canister Design for Off-Normal Condition Loads (Subsection NF components)

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits [1] (NF-3221.2)	
			$P_m < 1.5\,S_m$ ($1.5S_m$ =26,250 psi)	$P_m + P_b < 2.25\,S_m$ ($2.25S_m$= 39,380 psi)
21	30 psig Int. Press. + 80 kip push + Thermal (115° F)	Horizontal	2,499	2,499
22	30 psig Int. Press. + 80 kip pull + Thermal (115° F)	Horizontal	13,860 [2]	33,010 [2]

Notes:
1. Design stress intensity, S_m is taken at 500° F for material SA-240 Gr.304, see Table 3.9.1-22 for allowable stresses.
2. Linearized stress intensities through the thickness of the component plates.

Table 3.9.1-30
Summary of Calculated Stresses in the Alternate Canister Design for Off- Normal Condition Loads (Subsection NF weld)

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits [1] (NF-3226.2) $S_x < 1.5 \times 0.3\ S_u$ ($0.45\ S_u = 28{,}530$ psi)
21	30 psig Int. Press. + 80 kip push + Thermal (115° F)	Horizontal	152
22	30 psig Int. Press. + 80 kip pull + Thermal (115° F)	Horizontal	16,407

Note:
1. Tensile strength, S_u, is taken at 500° F for material SA-240 Gr.304; see Table 3.9-23 for allowable stresses.

Table 3.9.1-31
Summary of Calculated Stresses in the Alternate Canister Design at the End Closure Welds for Off-Normal Condition Loads

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits [1] (NB-3222)		
			$P_m < 0.8S_m$ ($0.8S_m$ = 14,000 psi)	$P_L+P_b < 0.8(1.5\ S_m)$ ($1.2S_m$=21,000 psi)	$P_L+P_b+Q < 0.8(3\ S_m)$ ($2.4\ S_m$ = 42,000 psi)
21	30 psig Int. Press. + 80 kip push + Thermal (115° F)	Horizontal	3,123	3,123	3,602
22	30 psig Int. Press. + 80 kip pull + Thermal (115° F)	Horizontal	3,110	3,110	3,660

Notes:
1. Design stress intensity, S_m is taken at 500° F for material SA-240 Gr.304, see Table 3.9.1-7 for allowable stresses.

Table 3.9.1-32
Summary of Calculated Stresses in the Alternate Canister Design
for Accident Condition Loads (Subsection NB components)

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits (1)	
			$P_m < 0.7\ S_u$ ($0.7\ S_u = 41{,}440$ psi)	$P_L + P_b < S_u$ ($S_u = 59{,}200$ psi)
4	120 psig internal pressure under fire accident	Horizontal	15,389 (2)	15,389 (2)
5	25 psig external pressure under flood accident	Horizontal	2,953	2,953
14	Top End Drop 75g + 30 psig Int. Press.	Vertical	15,645	15,645
15	Bottom End Drop 75g + 30 psig Int. Press.	Vertical	16,584	16,584
16	Top End Drop 75g + 15 psig Ext. Press.	Vertical	18,685	18,685
17	Bottom End Drop 75g + 15 psig Ext. Press.	Vertical	20,081	20,081
23	30 psig Int. Press. + 110 kip push + Thermal (115° F)	Horizontal	9,854	9,854
24	30 psig Int. Press. + 110 kip pull + Thermal (115° F)	Horizontal	4,393	4,393

Notes:
1. Lesser values of allowable stresses for material SA-182 Type F304 is also used for material SA-240 Type 304. See Table 3.9.1-7 for allowable stress intensities.
2. For metal temperature of 800° F under accident fire, allowable stress intensities, $P_m < 2.4\ S_m$ ($2.4\times15{,}200$ psi = 36,480 psi), $P_L + P_b < 3.6\times S_m$ ($3.6\times15{,}200$psi = 54,720 psi), where S_m = 15,200 psi for both SA240-304 and SA182-F304 at 800 °F.

Table 3.9.1-33
Summary of Calculated Stresses in the Alternate Canister Design for Accident Condition Loads (Subsection NF components)

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits [1]	
			$P_m < 0.7\,S_u$ ($0.7\,S_u$ = 40,600 psi)	$P_m + P_b < S_u$ (S_u = 58,000 psi)
4	120 psig internal pressure under fire accident	Horizontal	16,570 [2]	16,570 [2]
5	25 psig external pressure under flood accident	Horizontal	900	900
14	Top End Drop 75g + 30 psig Int. Press.	Vertical	6,337	6,337
15	Bottom End Drop 75g + 30 psig Int. Press.	Vertical	9,024	9,024
16	Top End Drop 75g + 15 psig Ext. Press.	Vertical	6,992	6,992
17	Bottom End Drop 75g + 15 psig Ext. Press.	Vertical	9,121	9,121
23	30 psig Int. Press. + 110 kip push + Thermal (115° F)	Horizontal	3,565	3,565
24	30 psig Int. Press. + 110 kip pull + Thermal (115° F)	Horizontal	19,090 [3]	45,480 [3]

Notes:
1. Least values of allowable stresses for materials SA-240 Type 304, SA-182 Type F304, and SA-36 are used for stress evaluations. See Table 3.9.1- 22 for allowable stress intensities.
2. For metal temperature of 800° F under accident fire, allowable stress intensities, $P_m < 0.7\ S_u$ ($0.7 \times 53{,}300$ psi = 37,310 psi) and $P_m + P_b < S_u$ ($1 \times 53{,}300$ psi = 53,300 psi), where S_u = 53,300 psi of SA-36 at 800 °F (the least tensile strength value at 800 °F of SA-240 Type 304, SA-182 Type F304, and SA-36)
3. Linearized stress intensities through the thickness of the component plates.

Table 3.9.1-34
Summary of Calculated Tensile Stresses Normal to the NF Weld Throat In the Alternate Canister Design for Accident Condition Loads (Subsection NF welds)

Load Case	Combination of Loads	Canister Orientation	Tensile Stress Limits [(1)] $S_x < 2 \times 0.3\ S_u$ ($0.6\ S_u$ = 38,040 psi)
4	120 psig internal pressure under fire accident	Horizontal	6,249 [(2)]
5	25 psig external pressure under flood accident	Horizontal	39
14	Top End Drop 75*g* + 30 psig Int. Press.	Vertical	483
15	Bottom End Drop 75*g* + 30 psig Int. Press.	Vertical	596
16	Top End Drop 75*g* + 15 psig Ext. Press.	Vertical	598
17	Bottom End Drop 75*g* + 15 psig Ext. Press.	Vertical	432
23	30 psig Int. Press. + 110 kip push + Thermal (115° F)	Horizontal	135
24	30 psig Int. Press. + 110 kip pull + Thermal (115° F)	Horizontal	21,928

Notes:
1. See Table 3.9.1-23 for allowable tensile stresses.
2. For metal temperature of 800° F under accident fire, allowable stress intensities, $S_x < 2 \times 0.3\ S_u$ (= 0.6 × 62,800 psi = 37,680 psi), where S_u = 62,800 psi at 800 °F for SA-240 Type 304.

Table 3.9.1-35
Summary of Calculated Stresses in the Alternate Canister Design for Accident Condition Loads (Top End Closure Welds)

Load Case	Combination of Loads	Canister Orientation	Stress Intensity Limits (1)	
			$P_m < 0.8 \times 0.7\ S_u$ (0.56 S_u=33,152 psi)	$P_L + P_b < 0.8 \times S_u$ (0.8 S_u=47,360 psi)
4	120 psig internal pressure under fire accident	Horizontal	12,502 (2)	12,502 (2)
5	25 psig external pressure under flood accident	Horizontal	2,054	2,054
14	Top End Drop 75*g* + 30 psig Int. Press.	Vertical	5,578	5,578
15	Bottom End Drop 75*g* + 30 psig Int. Press.	Vertical	8,582	8,582
16	Top End Drop 75*g* + 15 psig Ext. Press.	Vertical	7,808	7,808
17	Bottom End Drop 75*g* + 15 psig Ext. Press.	Vertical	12,153	12,153
23	30 psig Int. Press. + 110 kip push + Thermal (115° F)	Horizontal	3,132	3,132
24	30 psig Int. Press. + 110 kip pull + Thermal (115° F)	Horizontal	3,132	3,132

Notes:
1. Lesser values of allowable stresses for material SA-182 Type F304 is also used for material SA-240 Type 304. See Table 3.9.1-7 for allowable stress intensities.
2. For maximum metal temperature at 800° F under accident fire condition, allowable stress intensities are $P_m < 0.8 \times 2.4\ S_m$ ($0.8 \times 2.4 \times 15{,}200$ psi = 29,184 psi) and $P_L + P_b < 0.8 \times 3.6 \times S_m$ ($0.8 \times 3.6 \times 15{,}200$psi = 43,776 psi).

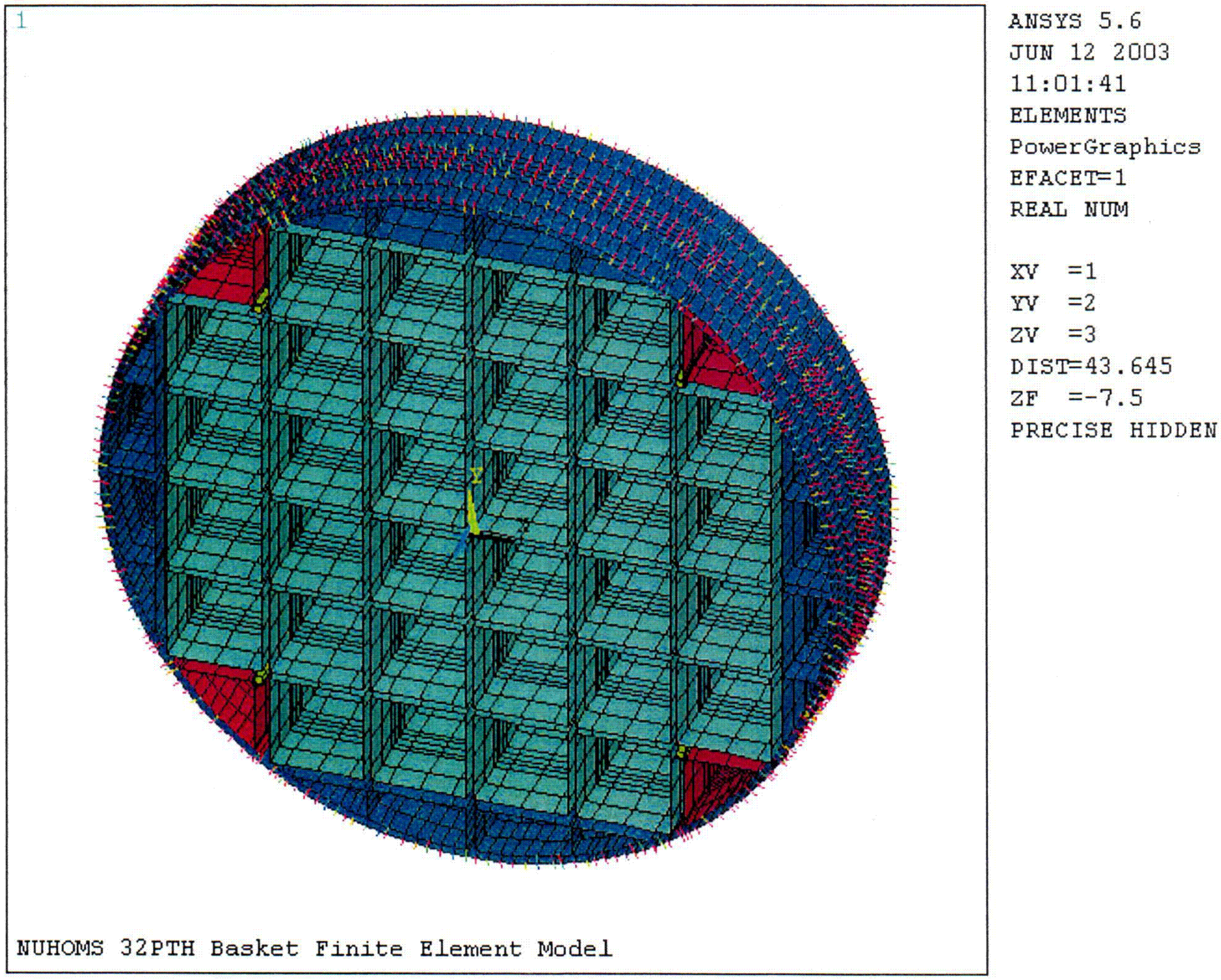


Figure 3.9.1-1

32PTH DSC Basket – 3D Cross Section Finite Element Model

C01

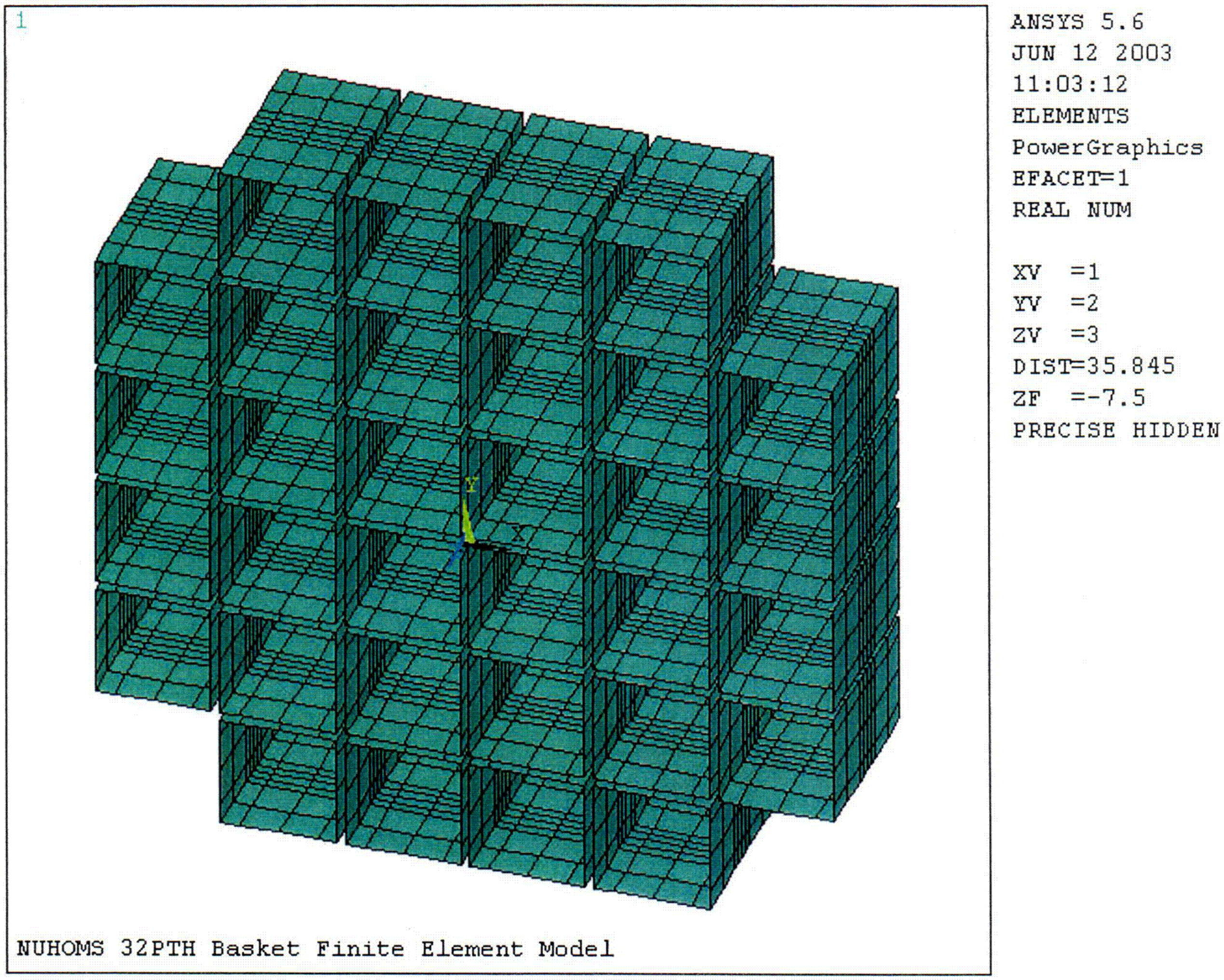


Figure 3.9.1-2
32PTH DSC Basket Cross Section Finite Element Model – Fuel Compartments

C02

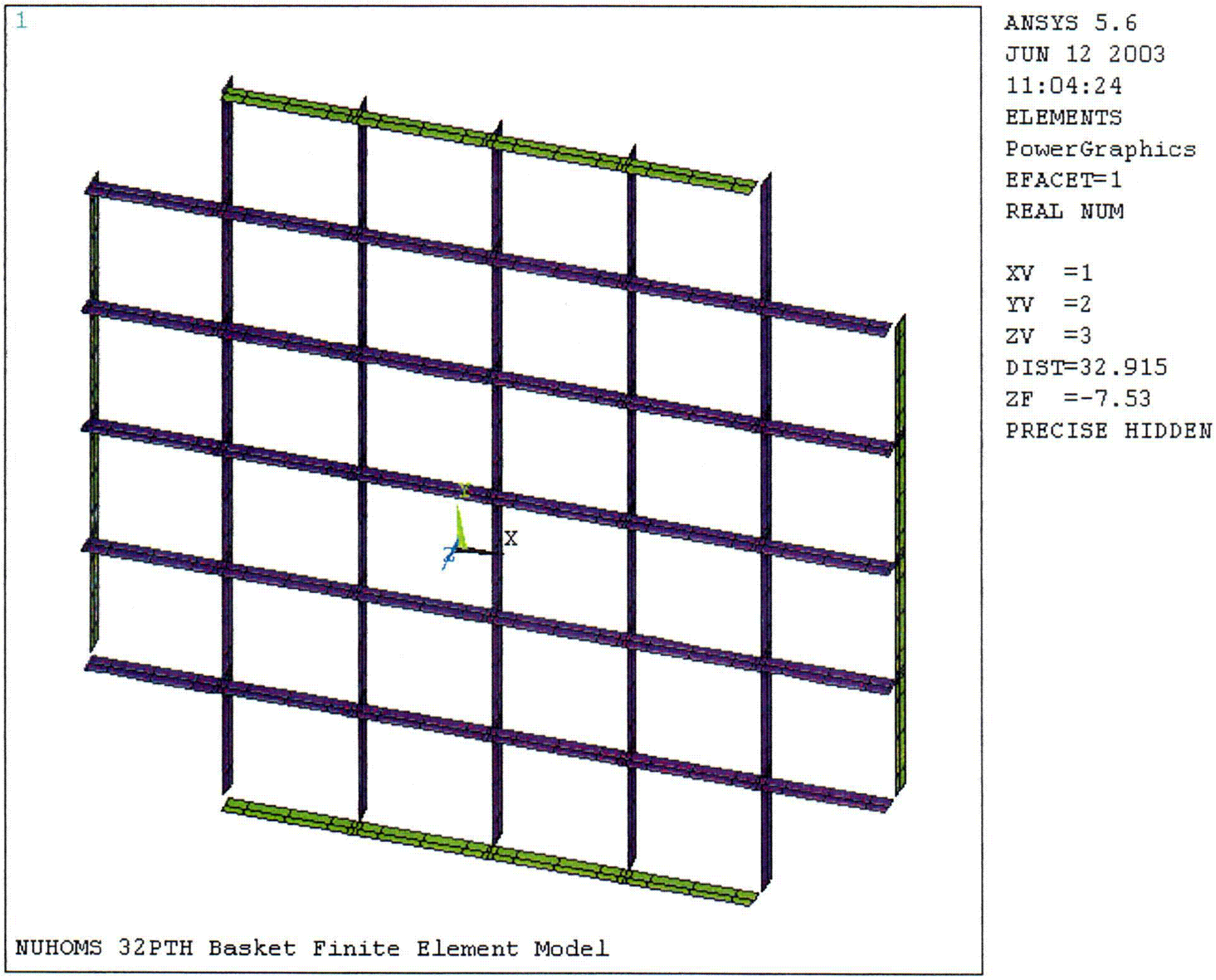


Figure 3.9.1-3
32PTH DSC Basket Cross Section Finite Element Model – Center and Outer Plates

C03

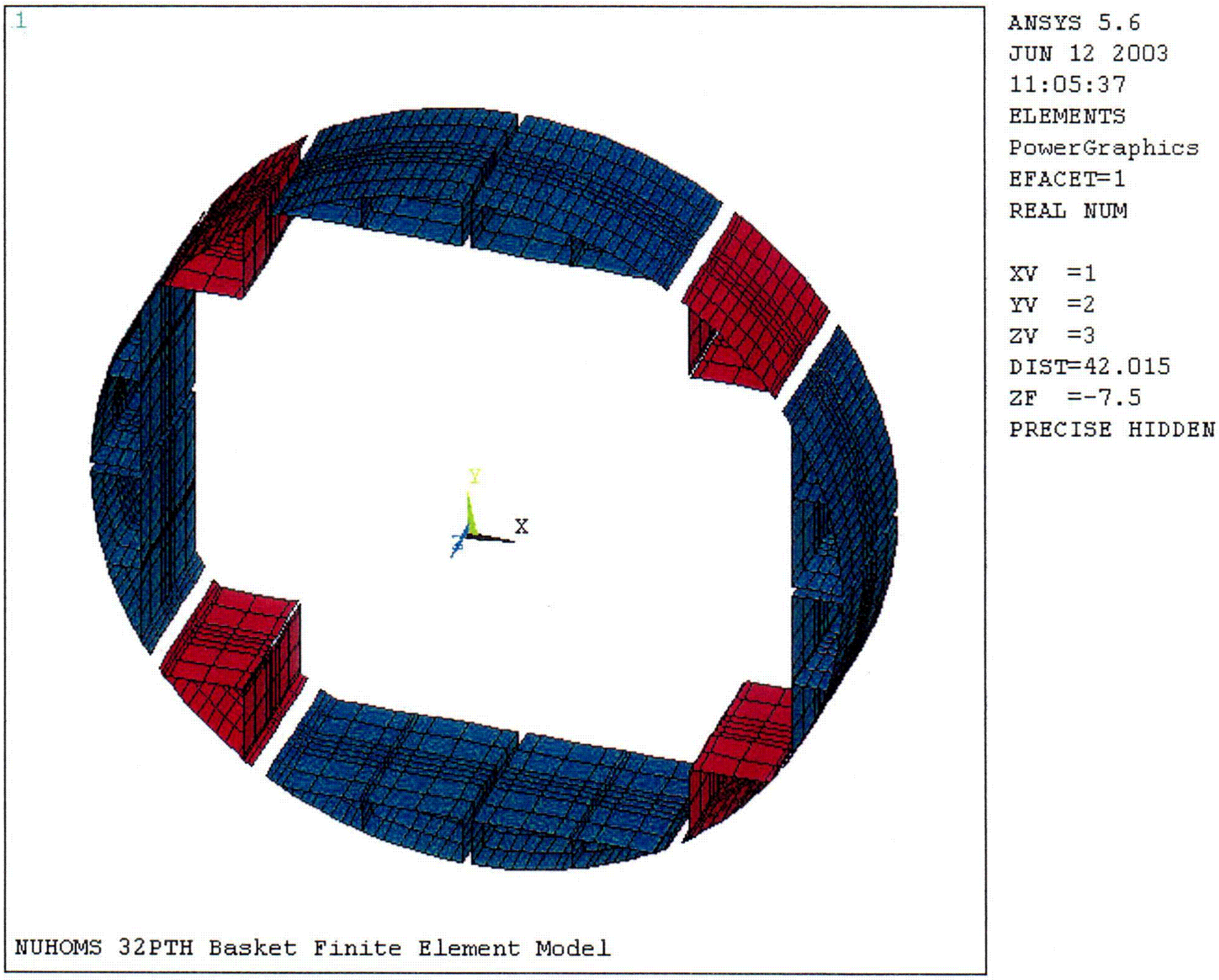


Figure 3.9.1-4
32PTH DSC Basket Cross Section Finite Element Model – Support Rails

C04

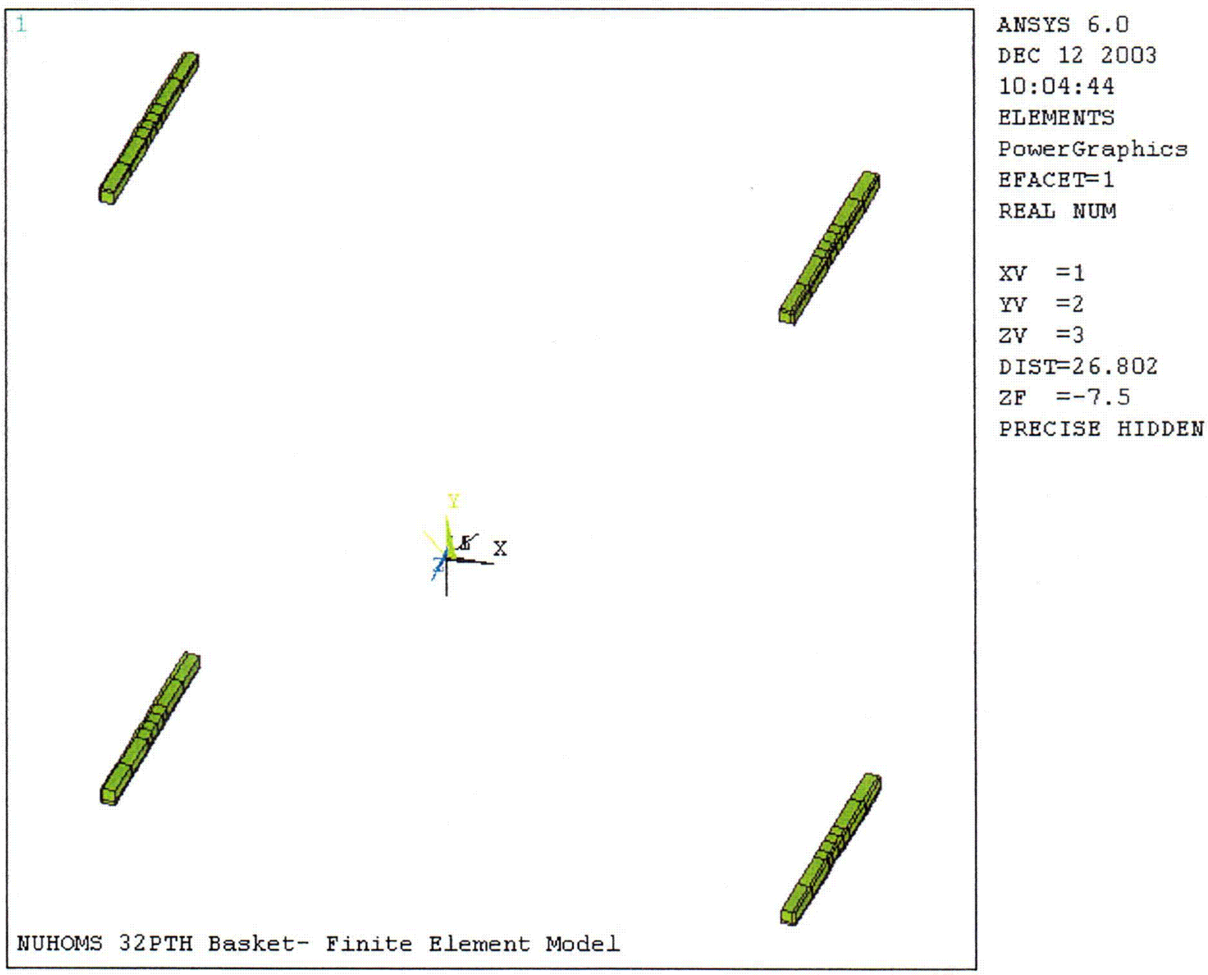


Figure 3.9.1-5
32PTH DSC Basket Cross Section Finite Element Model – 7/8 inch Square Bars

C 05

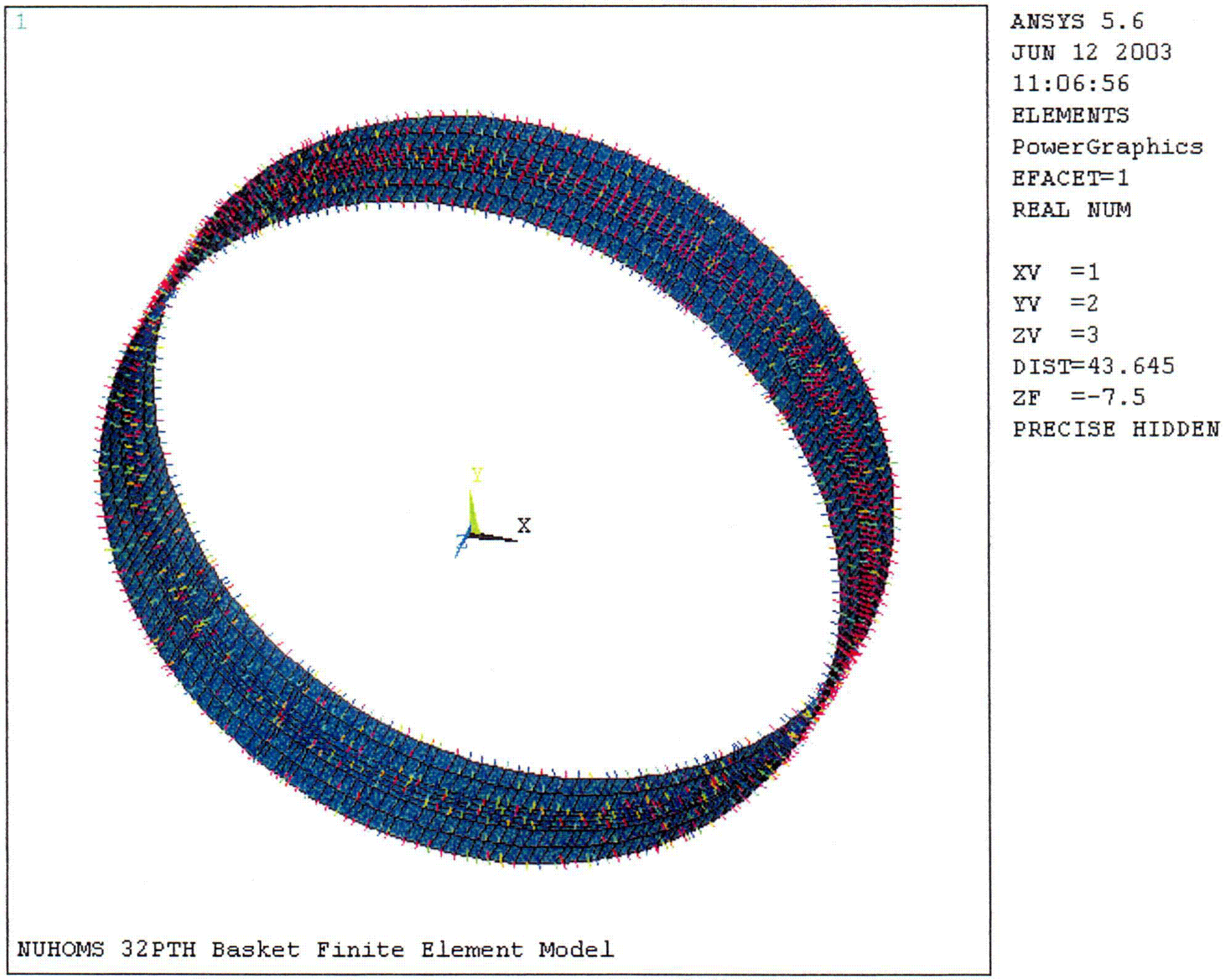


Figure 3.9.1-6
32PTH DSC Basket Cross Section Finite Element Model – Canister and Gap Elements

C06

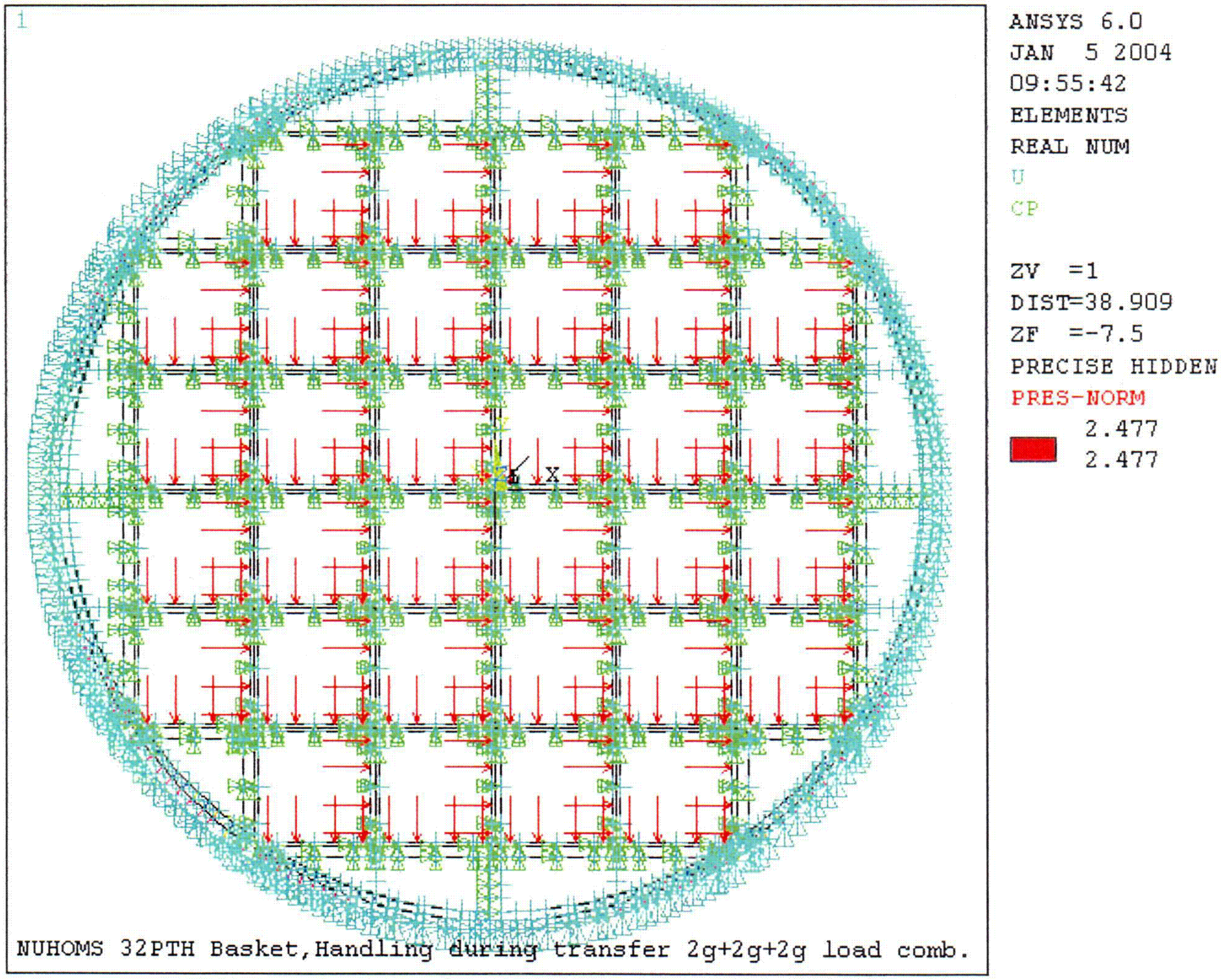


Figure 3.9.1-7
32PTH DSC Basket FEM – Transfer Handling Loads Boundary Conditions

C07

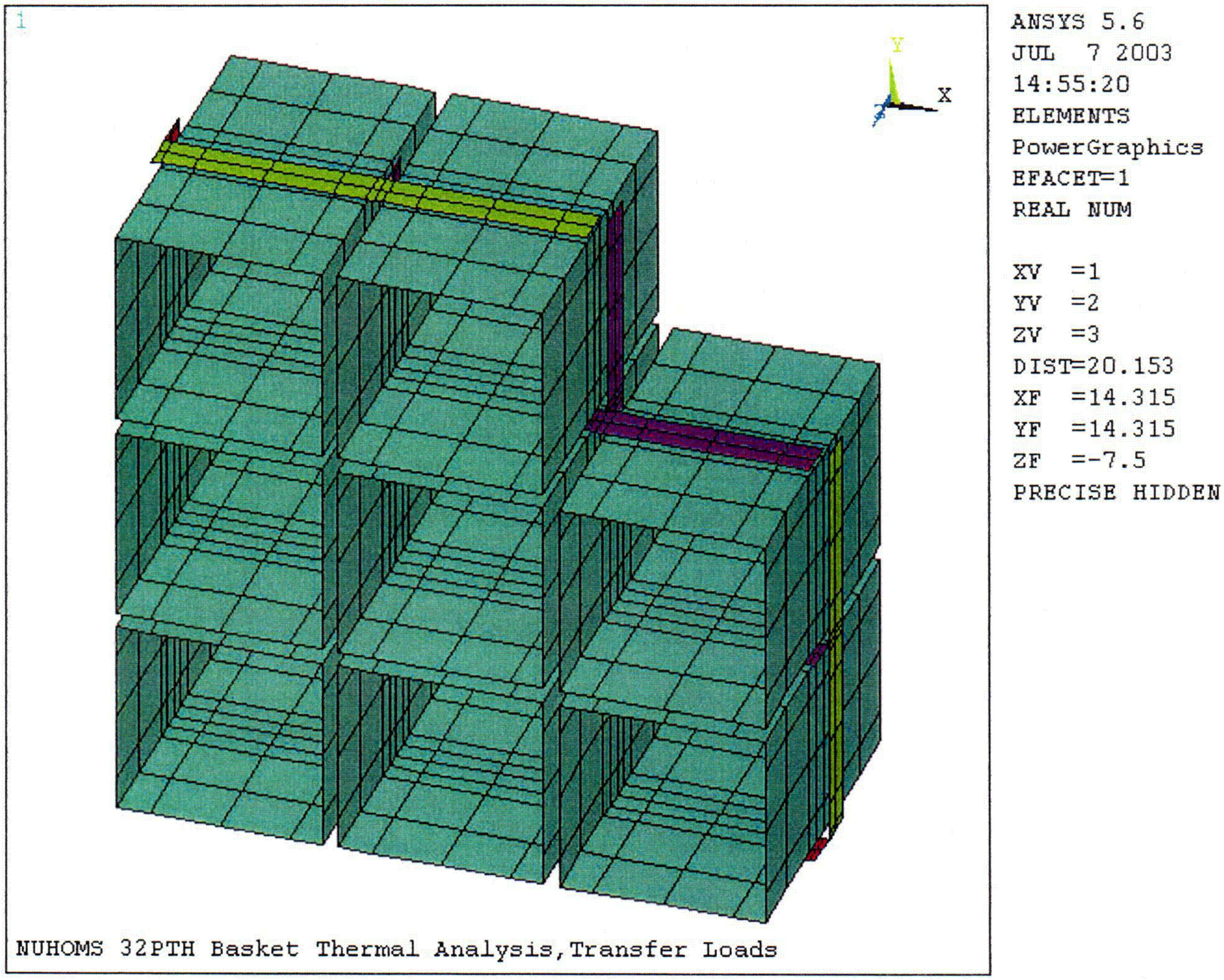


Figure 3.9.1-8
Fuel Compartments Finite Element Model – Thermal Analysis

C08

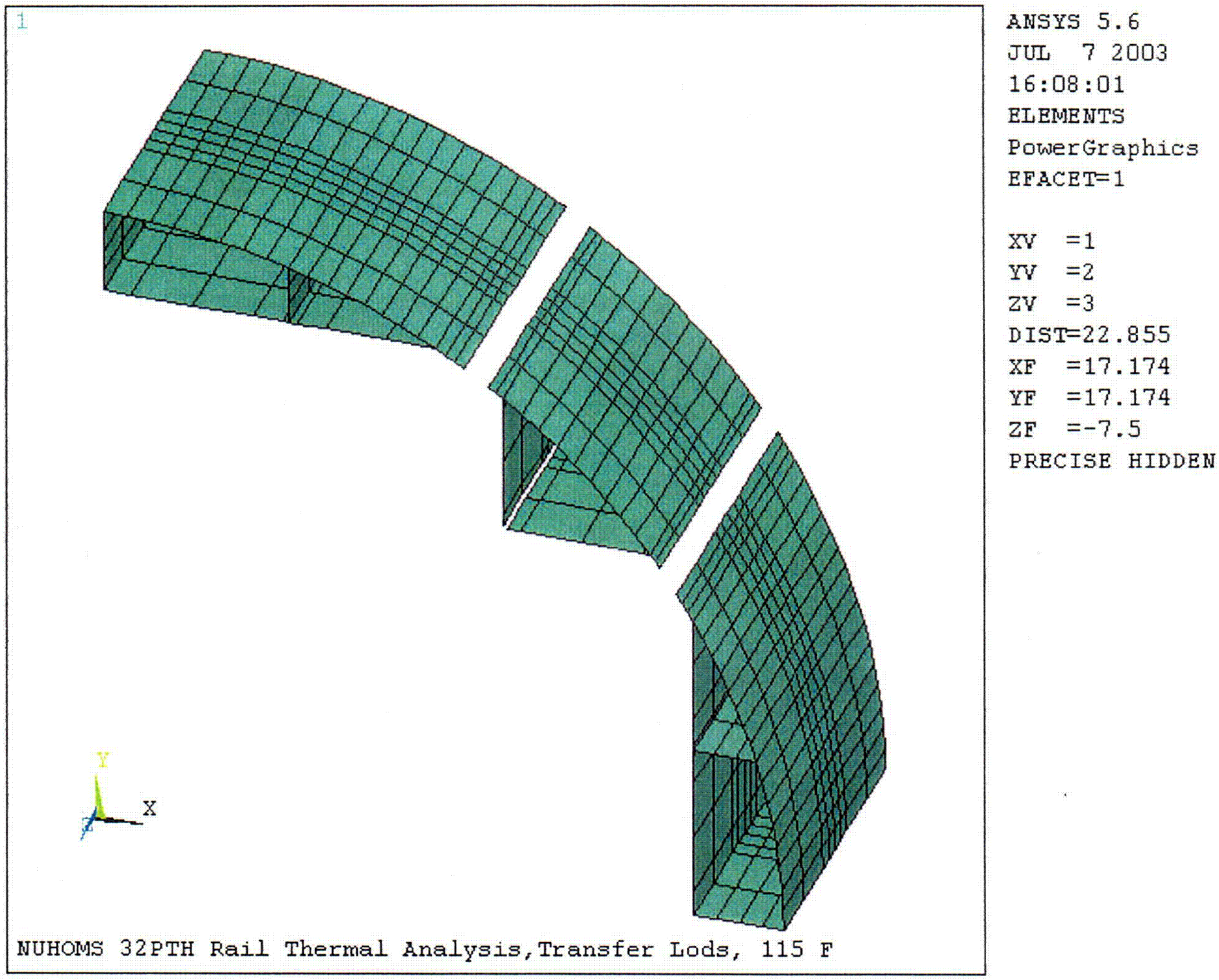


Figure 3.9.1-9
Support Rail Finite Element Model – Thermal Analysis

C09

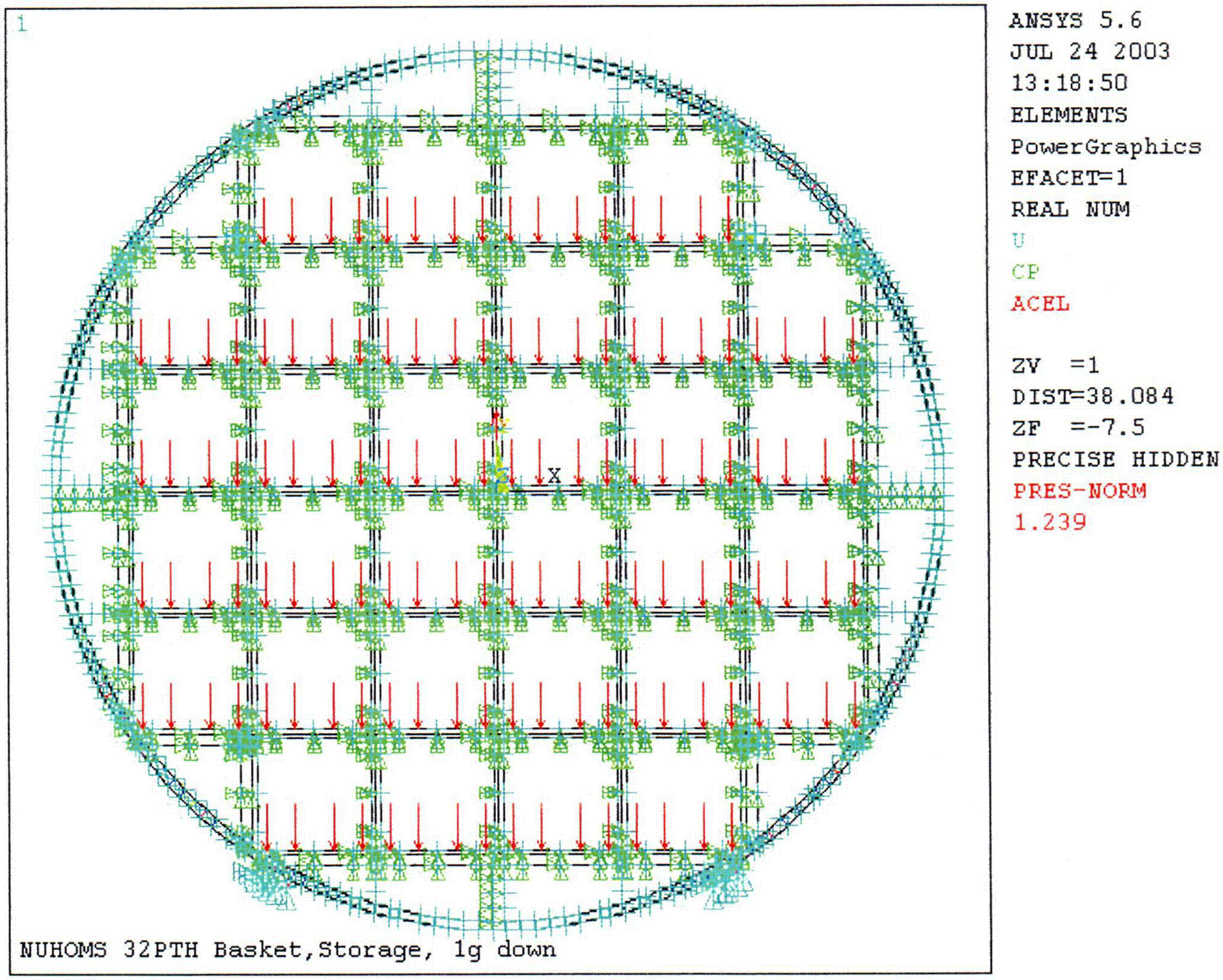


Figure 3.9.1-10
Finite Element Model Boundary Conditions for Dead Weight Storage Load

C10

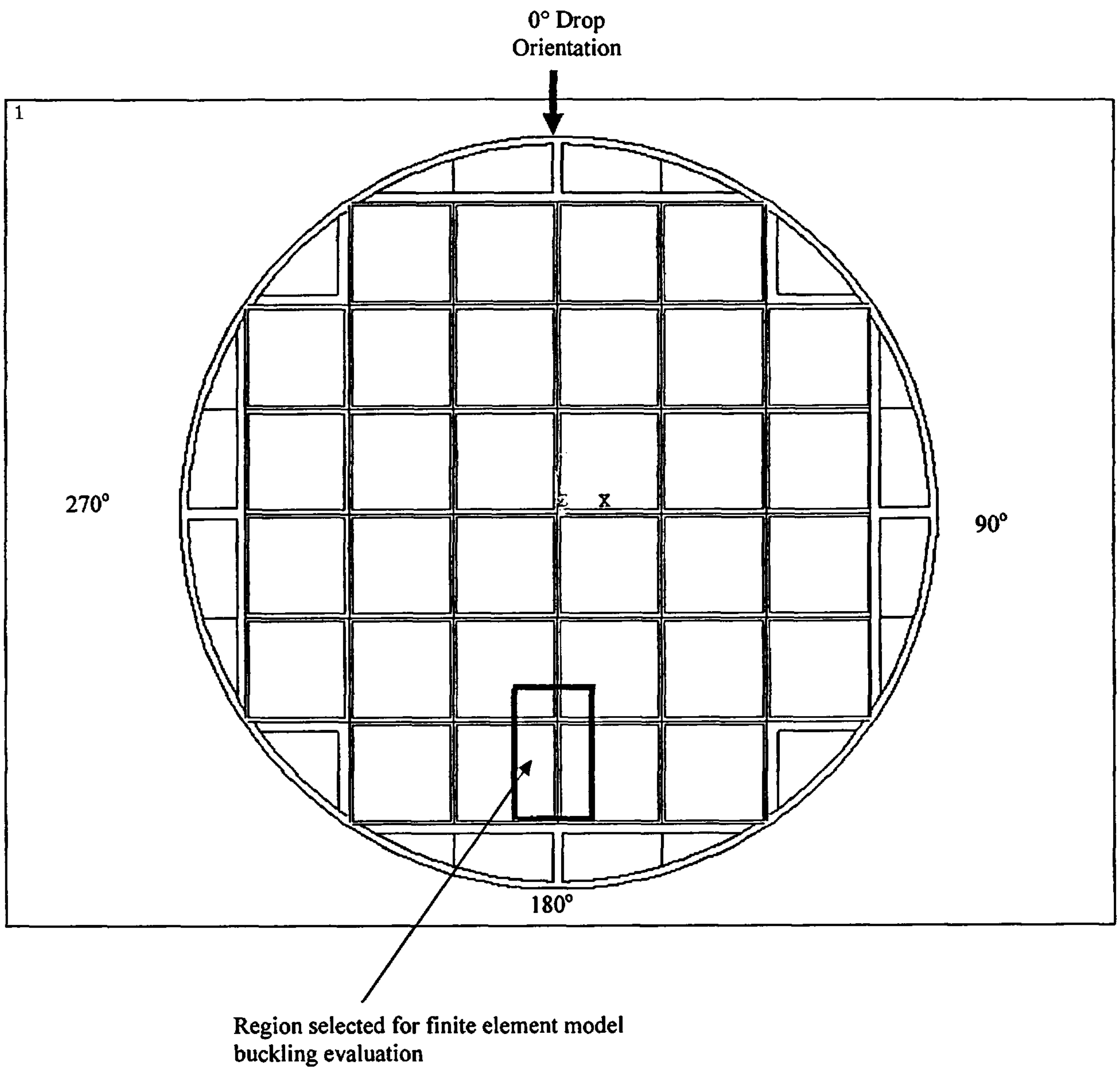


Figure 3.9.1-11
32PTH DSC Basket Geometry for Buckling Evaluation

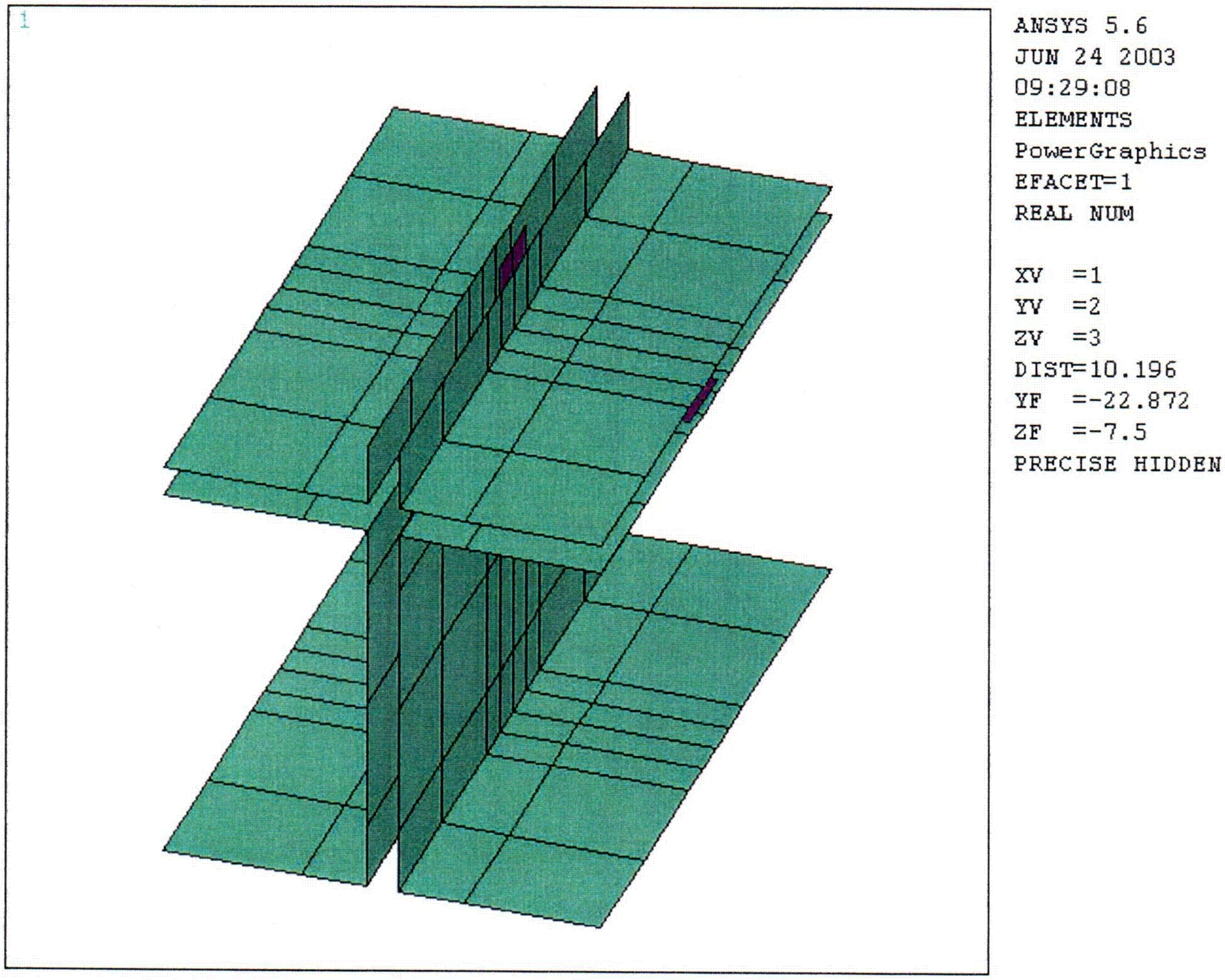


Figure 3.9.1-12
Finite Element Model used for Fuel Basket Grid Buckling Evaluation

C11

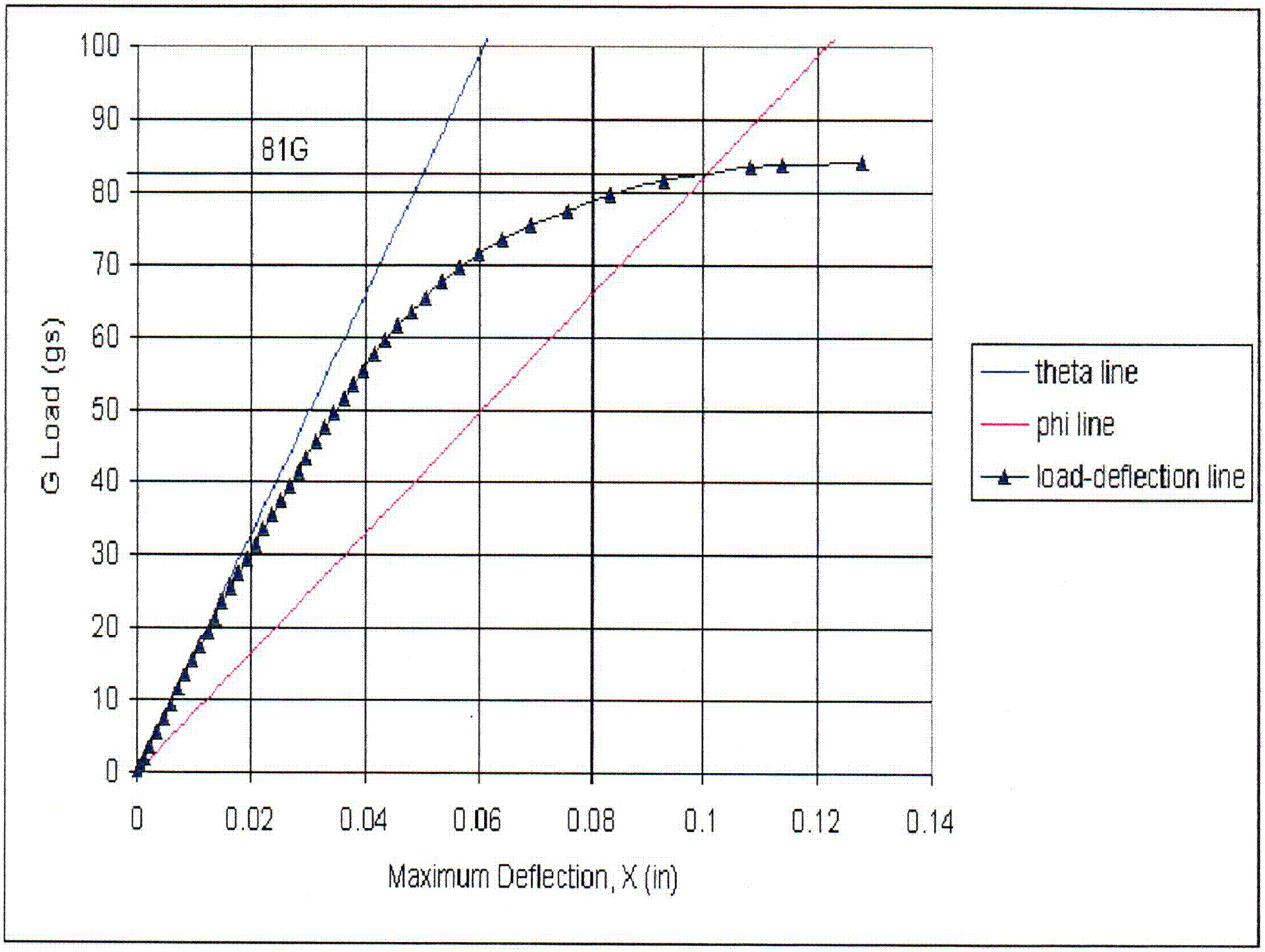


Figure 3.9.1-13

Basket Fuel Compartment - Collapse Load Computation for the 45° Orientation

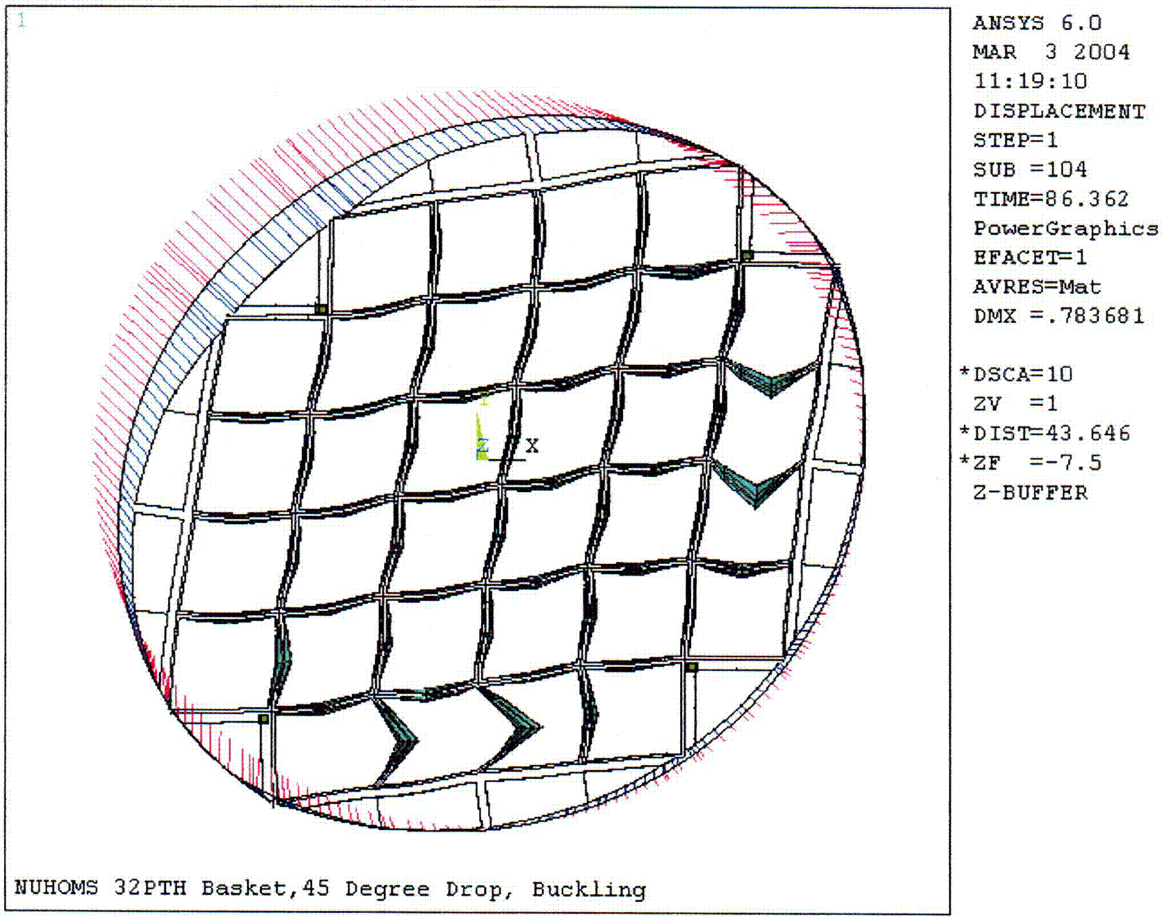


Figure 3.9.1-14
Full Size Basket Buckling Analysis – Deformed Geometry for 45° Orientation Drop at 86.362g

C13

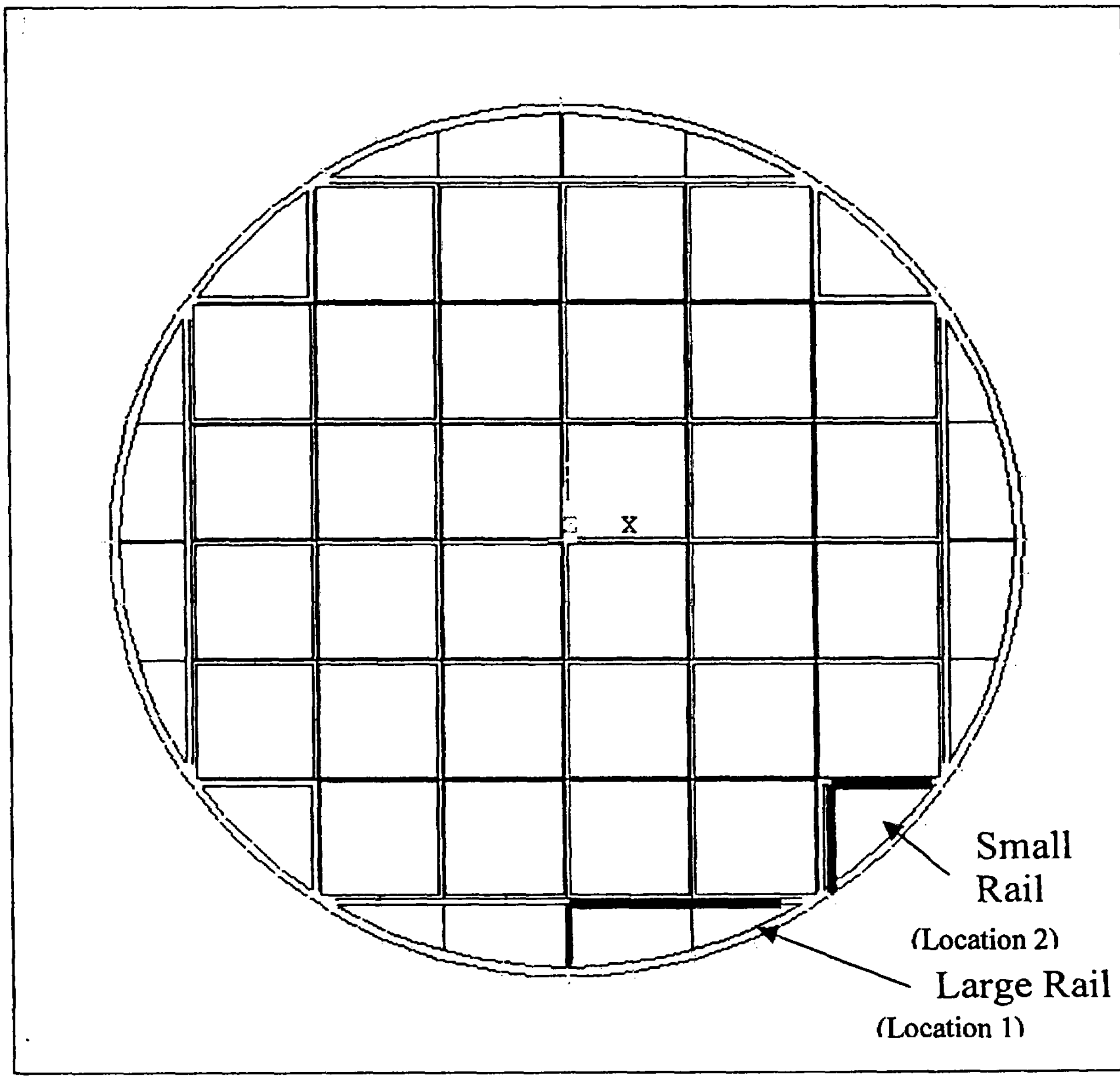


Figure 3.9.1-15
Fuel Basket Support Rail Cross Section Geometry

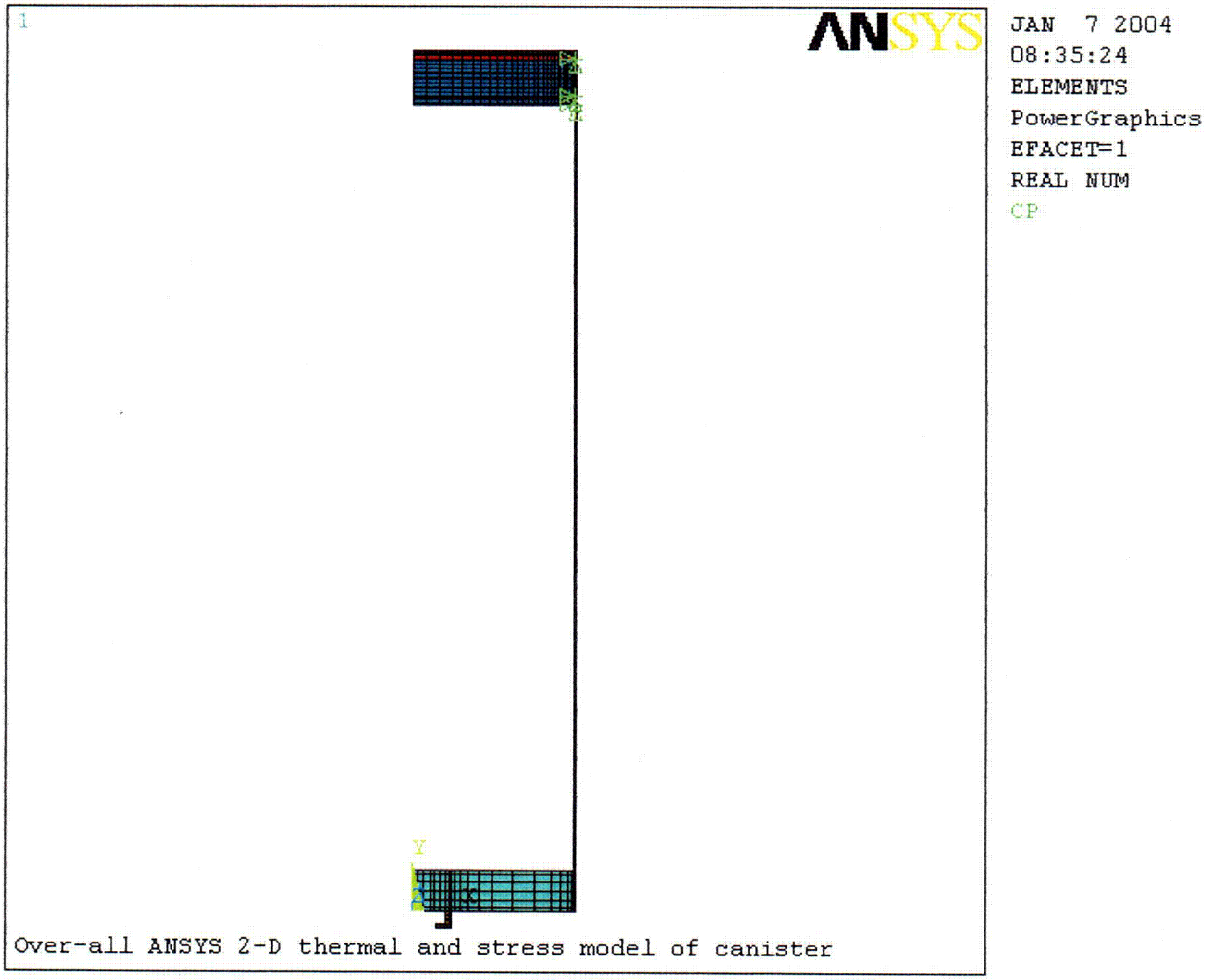


Figure 3.9.1-16
2-D Canister Axisymmetrical Thermal and Stress Finite Element Model

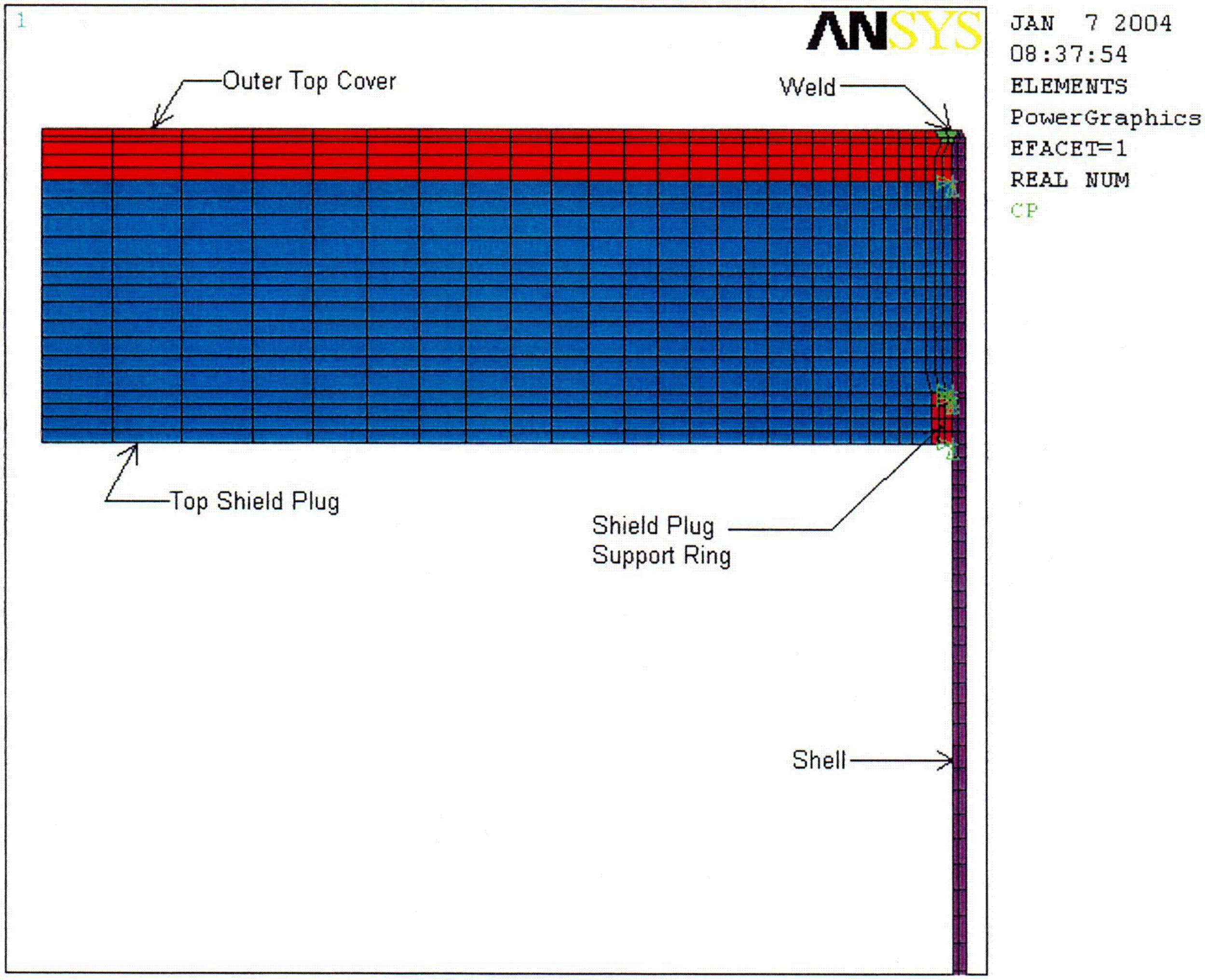


Figure 3.9.1-17
Top End of the 2-D Axisymmetrical Canister Model

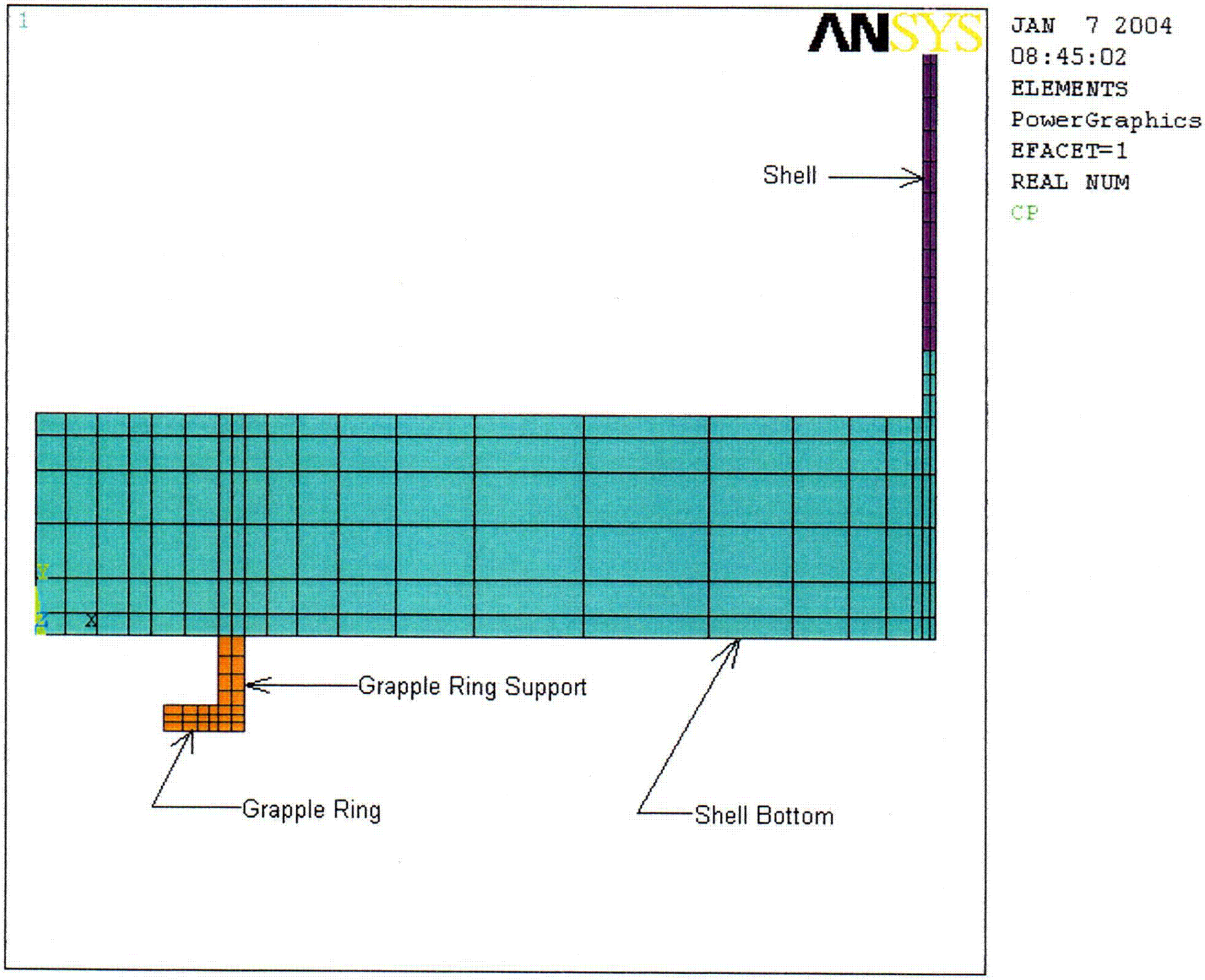


Figure 3.9.1-18
Bottom End of the 2-D Axisymmetrical Canister Model

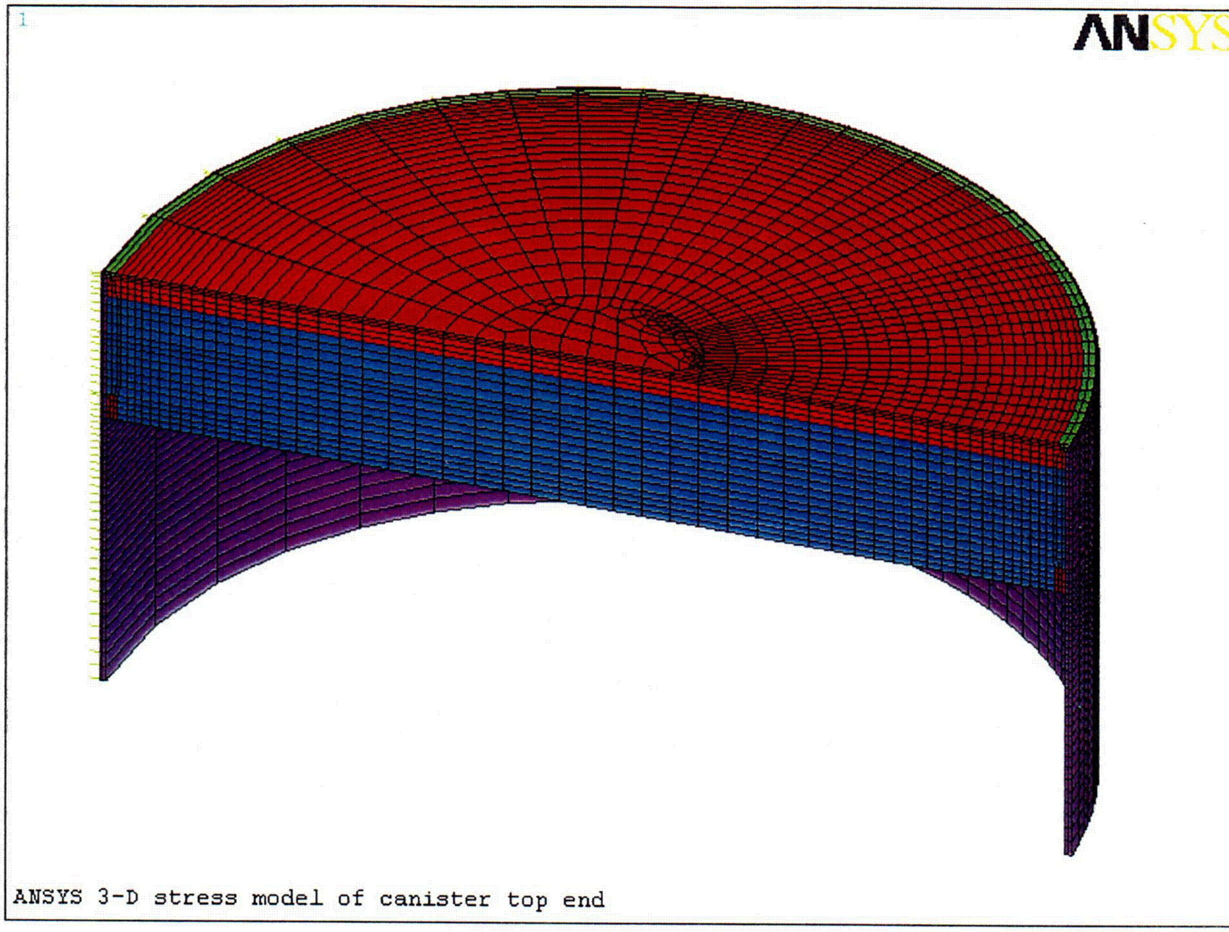


Figure 3.9.1-19
3-D DSC Canister Top End Assembly Finite Element Model

C17

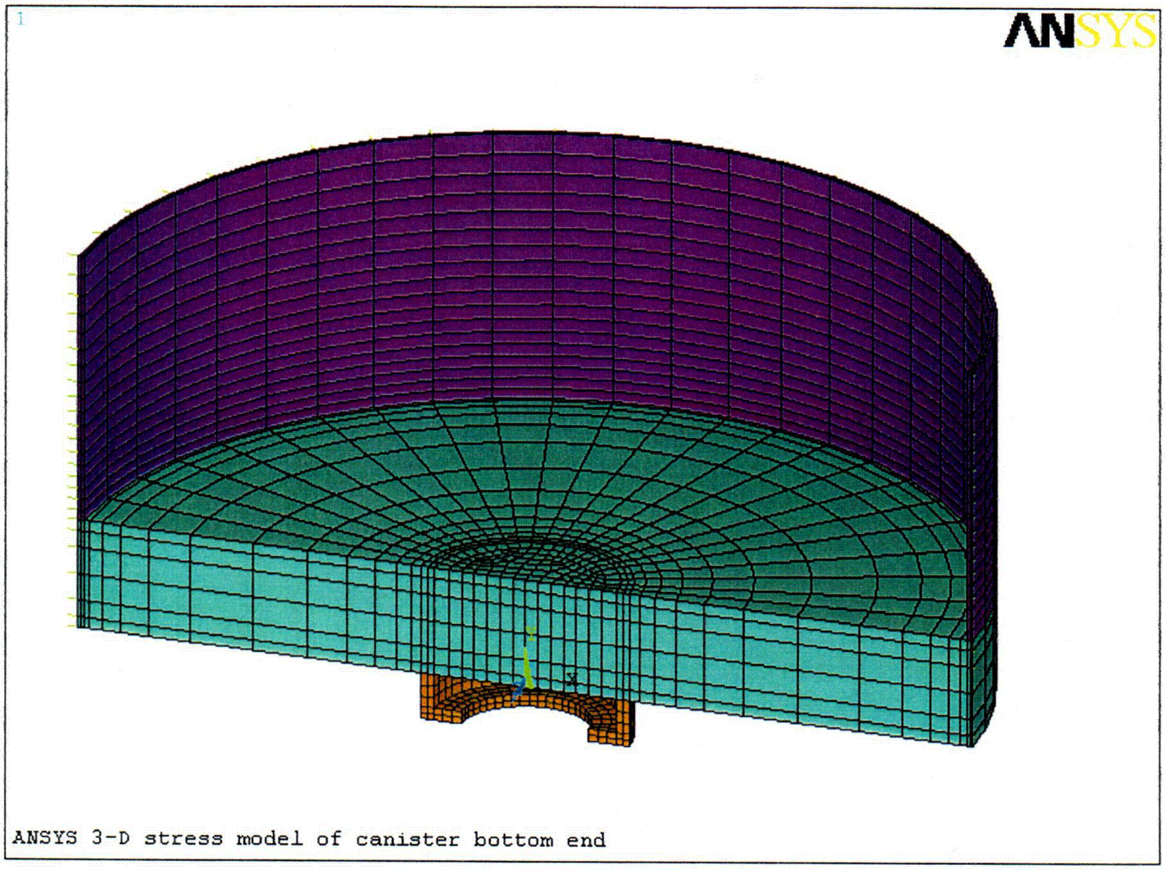


Figure 3.9.1-20
3-D DSC Canister Bottom End Assembly Finite Element Model

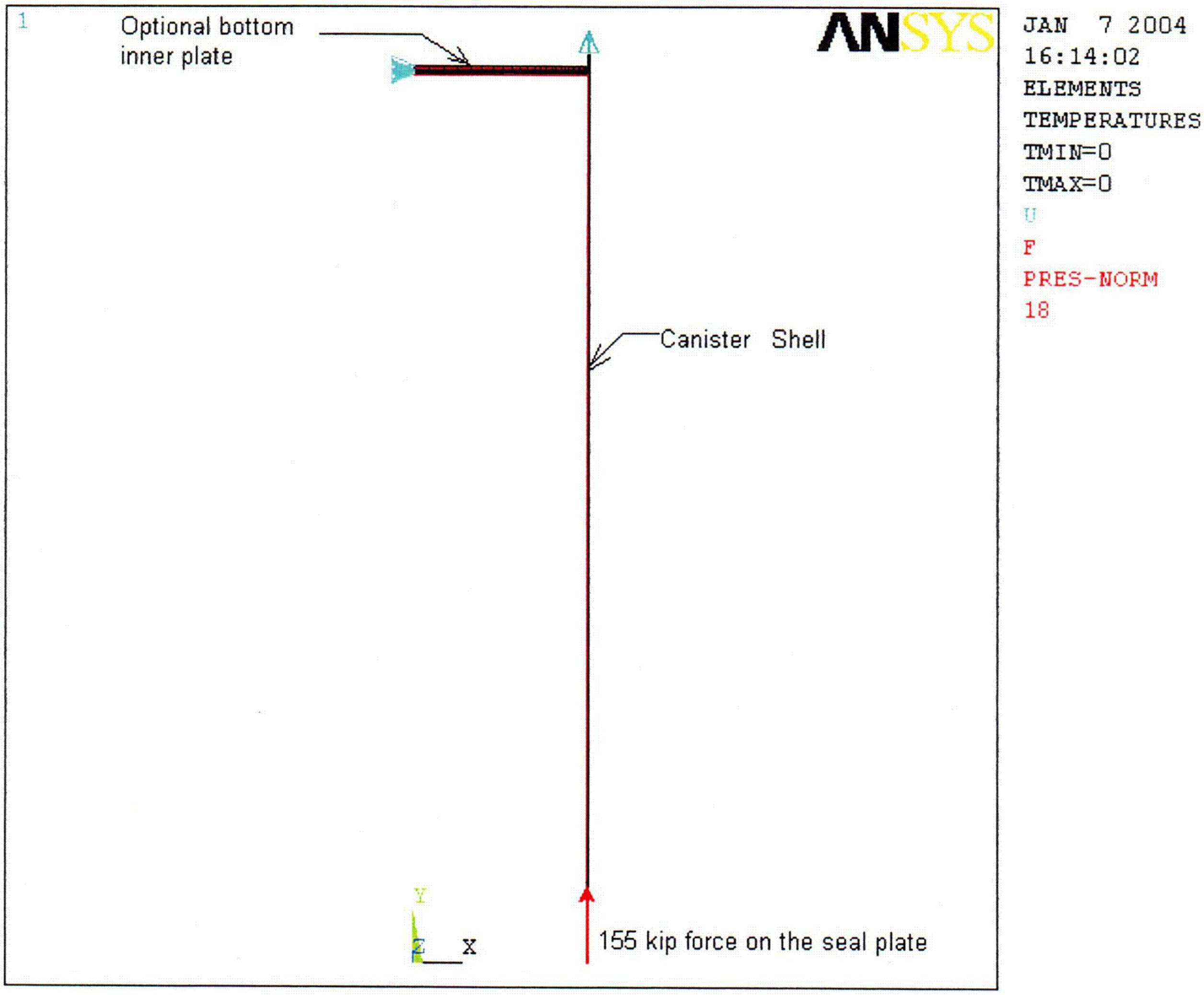


Figure 3.9.1-21
32PTH DSC Canister Finite Element Model used for Pressure Test Analysis

C19

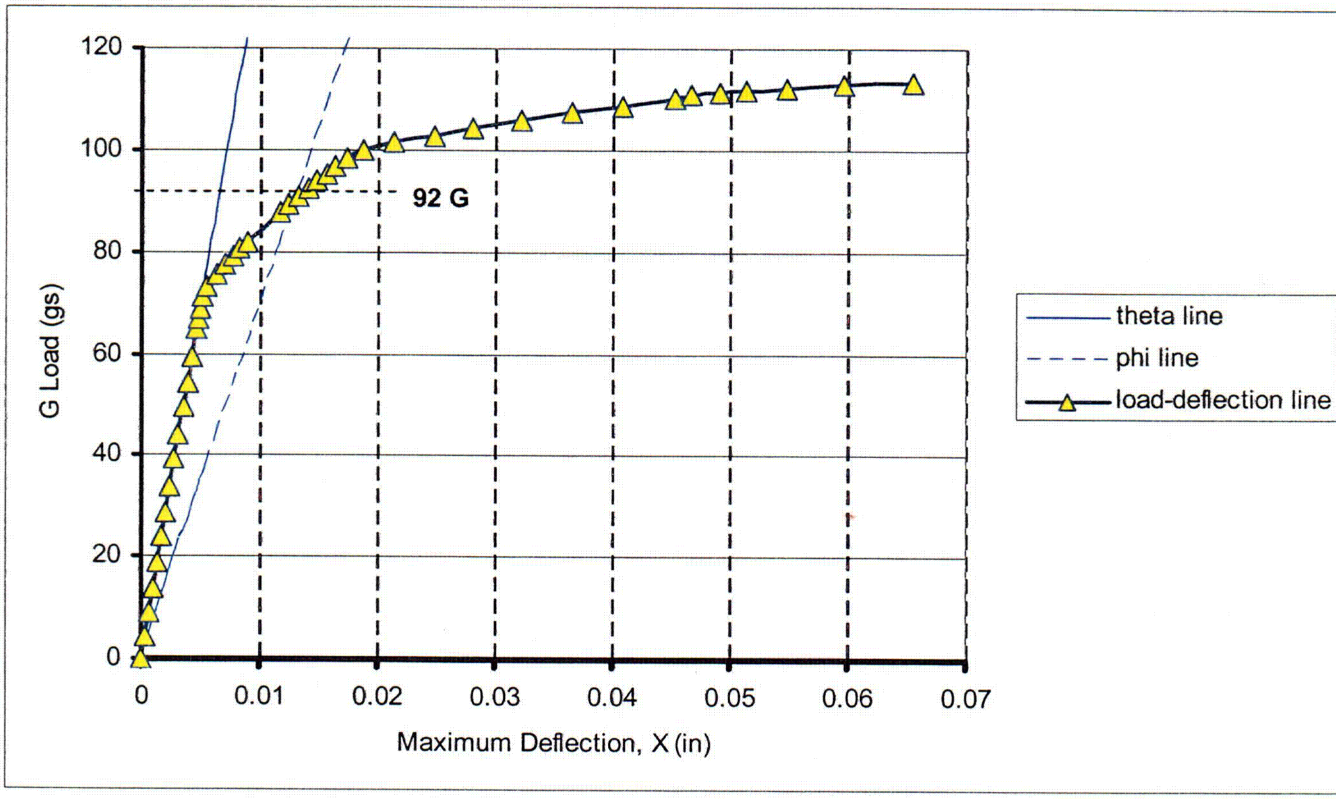


Figure 3.9.1-22

32PTH DSC Canister End Drop – Collapse Load Computations

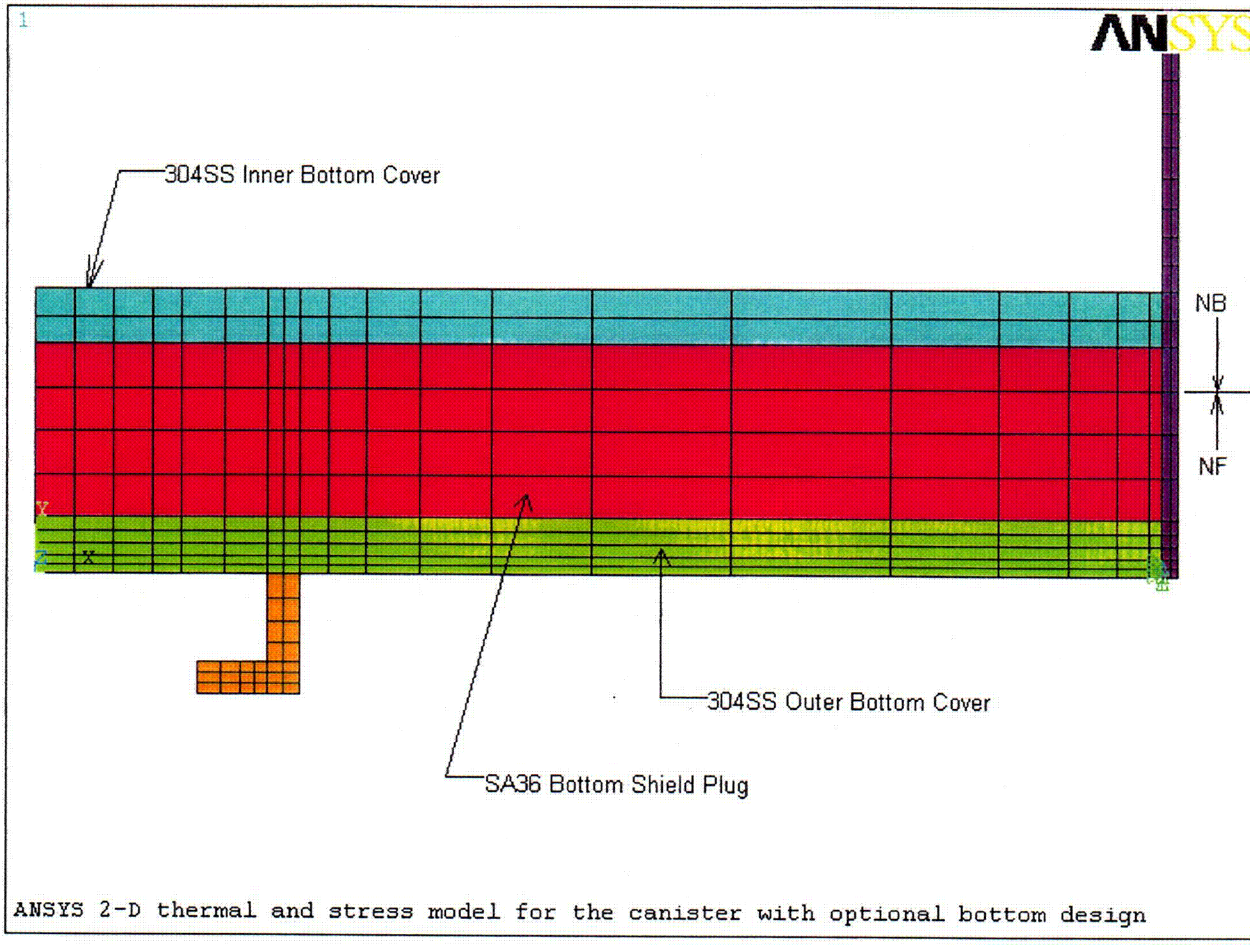


Figure 3.9.1-23
2-D Axisymmetric Finite Element Model of the Alternate Canister Bottom Assembly

C21