Contract No:

This document was prepared in conjunction with work accomplished under Contract No. DE-AC09-08SR22470 with the U.S. Department of Energy (DOE) Office of Environmental Management (EM).

Safety Guide 100

DESIGN GUIDE FOR PACKAGING AND OFFSITE TRANSPORTATION OF NUCLEAR COMPONENTS, SPECIAL ASSEMBLIES, AND RADIOACTIVE MATERIALS ASSOCIATED WITH THE NUCLEAR EXPLOSIVES AND WEAPONS SAFETY PROGRAM

CHAPTER 2.0

STRUCTURAL ASPECTS

November 7, 1994

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ACRONYMS

ALARA	As Low as Reasonably Achievable
ASME	American Society of Mechanical Engineers
CFR	Code of Federal Regulations
DOE	Department of Energy
IAEA	International Atomic Energy Agency
LANL	Los Alanios National Laboratory
NRC	Nuclear Regulatory Commission
SARP	Safety Analysis Report for Packaging
SNL	Sandia National Laboratory
SRS	Savannah River Site

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12.0 STRUCTURAL ASPECTS

2.1 INTRODUCTION

The material is intended to aid DOE personnel and its contractors who are responsible for packages that transport radioactive materials and special weapons components. This chapter addresses the structural concerns that are encountered when designing a package. Approaches and solutions presented provide consistent and well-understood techniques for designing the structural components of a package. Other techniques can be used, but this chapter covers the strategies that are generally used. It is believed that use of the techniques described herein produces a package which conforms to the appropriate federal regulations.

The structural designer is expected to be part of a design team and to interface with specialists in the area of thermal aspects, shielding, criticality, materials, containment, quality assurance, etc. Therefore, the structural designer should be knowledgeable about other chapters in this design safety guide.

Also, this guide is to be revised periodically to include information gained from experience as well as new regulatory guidance, etc. Operating contractors, national laboratories, and DOE personnel are invited to submit recommendations for improvement in scope and content. When other methods or means are proposed to meet the intent of the federal regulations of DOE policy, those proposals should be forwarded along with justifications to the DOE certifying official for consideration.

2.1.1 Scope

This chapter covers the structural aspects of designs for drum-type containers or packages used to transport radioactive materials and special weapons components. The containers considered are assumed to be thin-walled, relatively lightweight, and do not contain liquid payloads. Emphasis is on the structural design and the information essential to the certification process.

Not all conceivable drum-type packages and combinations of package contents are covered by this guide; however, the guide does give fundamental design information applicable to most designs. The designer is free to determine and prove how a particular design satisfies federal regulations. Most importantly, the guidance in this chapter is not a substitute for good engineering judgment.

2.1.2 Approach

It is assumed that the reader (or structural designer) is experienced in the principles of structures and solid mechanics; therefore, this type of information is not included.

The approach taken in presenting structural design guidelines in this chapter is to: first, emphasize the structural requirements as dictated by federal regulations (Sect. 2.2); second, state the current structural design criteria (Sect. 2.3); third, present design guidelines for various components of the container (Sect. 2.4); and fourth, indicate methodologies for structural validation of the design (Sect. 2.5). A list of references (Sect. 2.6) and a bibliography (Sect. 2.7) are provided to aid the structural designer. Appendices A through F contain an example of a typical analysis required for structural analysis, and Appendix G is a tutorial on bolt closure.

It is emphasized throughout this chapter that DOE is responsible for the overall safety of a package designed by an applicant who is seeking a license from DOE and, therefore, all the necessary design information for verifying the adequacy of the design must be conveyed in the Safety Analysis Report for Packaging (SARP). Guidance for preparation of a SARP is presented in a separate design safety guide.

2.1.3 Design Process

The design of a package for shipping radioactive materials and special weapons components requires many disciplines as shown on the Flow Charts in Chap. 1.0. While some aspects of the design, such as shielding and criticality, define the characteristics of the package, others, such as containment and thermal aspects, describe how the package functions in various environments. The structural aspects of the package, however, both define the configuration of the package as well as describing its performance in these environments. Most of the performance requirements in 10 CFR 71 require the application of different types of structural loading to the package. The structural aspects of the design interact with all the other aspects to specify the shape, size, and performance of the package.

The first step in the design process is recognition of the need. DOE needs packages for the safe transport of radioactive material and weapon components. All package designs must minimize the risks of transporting radioactive materials to the public, workers, and the environment by maintaining any exposure as low as reasonably achievable (ALARA).

The next step in the design process involves determination of requirements. For a license, the package must demonstrate compliance with the requirements of the federal regulations in 10 CFR 71 for Type B packages. These performance-based standards are described in Sect. 2.2.

The next step in the design process is establishing design criteria to be used along with the structural requirements. To decide if a design has an appropriate response to a given loading, structural design criteria are necessary. Section 2.3 discusses how structural design criteria are used to evaluate the safety of a given design.

The synthesis and optimization of a complete package design are the next steps in the design process. The typical drum-type container has many components that require individual design considerations which involve aspects such as thermal effects, shielding, containment, and materials. Figure 2.1 shows a typical drum-type package which is often used for transporting weapons materials for DOE. The function and loading conditions for each component of the package are different; therefore, a synthesis and optimization of the complete package must occur. Section 2.4 will focus on design guidelines for the major components of the package. The design guidance provided for each component is not intended to cover all possible design conditions and package configurations. Responsibility for a particular design ultimately lies with the design team.

After a design has been synthesized and optimized, the next step in the design process is validation of the design. Section 2.5 addresses the applicable structural validation methods.

The final step in the design process is the presentation of the design. Presentation is made in a SARP which documents the package's safety and adherence to federal regulations. The appropriate design considerations and design criteria are discussed in the SARP. Using the SARP, DOE reviews the package and decides whether or not a package license will be issued. Issuance of the license is based on a package's meeting the need of safely transporting radioactive material and weapons components. Table 2.1 lists radioactive material packages that have received DOE certificates of compliance. Included in this table are references to the SARPs which describe the package.

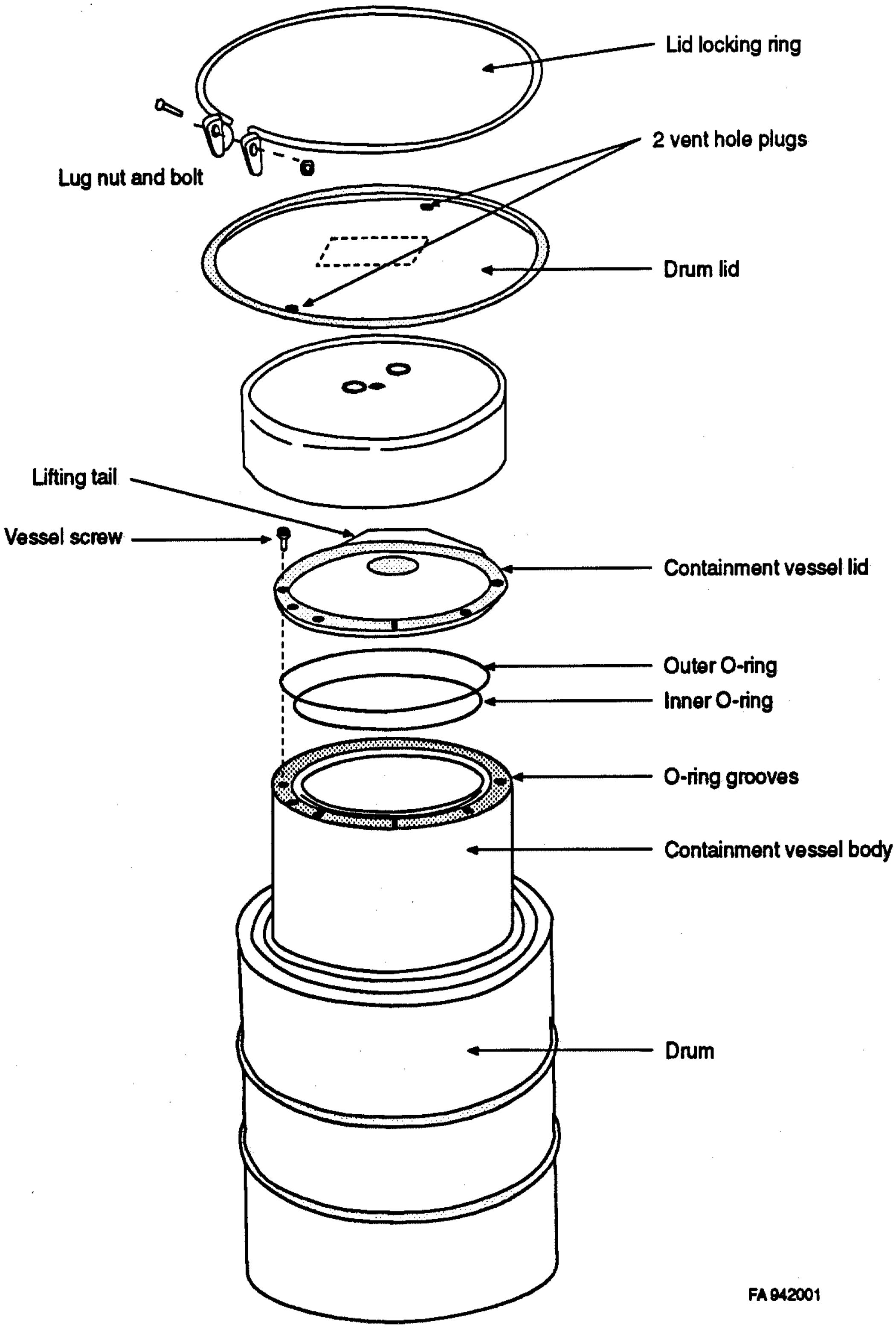


Fig. 2.1. Typical drum-type package.

Table 2.1. DOE certificates of compliance for radioactive materials packaging

CoC	User	Material	Container	Outer diameter	SARP
5320	SR	Oxides; plutonium, americium	Cask	10.75 in. and 12.75 in.	DPSPU 79-124-1, Rev. 1
5607	CH	Irradiated fuel	Cask	36 in.	T-2
5740	OR	Isotopes	TRU Cf cask	66.125 in.	ORNL-5409/R4
5797	OR	Fissile uranium	HFIR cask	25 in. & 31.5 in.	ORNL/TM-11656, Rev. 6
6387	RL	Fissile, large quantity, fuel elements, special form	HEDL model-60	6.625 in. C.V.	TC-138, Rev. 1 and Add. 1
6553	OR	Fissile uranium	UF_6 cylinder and overpack	48-in. cylinder	KY-655, Rev. 7
9099	ID	Fissile, large quantity, fuel elements	Overpack	–	EGG-ATRO-7737, Rev. 1
9132	RL	Irradiated fuel	Cask	52 in.	T-3, Rev. 6
9200	ID	Irradiated fuel	Cask	120 in.	NUPAC 125-B
9511	AL	Cesium chloride and strontium fluoride	Cask	54.25 in.	BUSS SARP, Rev. 3
9516	CH	Heat source plutonium	Cask	9.5 in.	MLM-MU-91-64-001, Rev. 5
9853	OR	Unirradiated fuel elements	Fuel containers	24.5 in.	ORNL/TM-11994
9859	OR	Tritium trap	6M drum	15 in.	ORNL/TM-8633
9932	SAN	Gases	Steel vessel	25 in.	UCRL 52424
9965	SR	Fissile, oxides	Drum - 30 gal	25 in.	DPSPU-83-124-1
9966	SR	Fissile, oxides	Drum - 30 gal	25 in.	DPSPU-83-124-1
9967	SR	Fissile, oxides	Drum - 55 gal	29.75 in.	DPSPU-83-124-1
9968	SR	Fissile, oxides	Drum - 35 gal	18.375 in.	DPSPU-83-124-1

2.2 STRUCTURAL REQUIREMENTS FOR THE TRANSPORT OF RADIOACTIVE MATERIAL

The Nuclear Regulatory Commission (NRC) regulations that govern the design of packaging for transportation of radioactive materials are found primarily in 10 CFR 71.[1] The current regulation was adopted on January 1, 1988; but there is a proposed rule for 10 CFR Part 71 which was issued by the NRC on June 8, 1988.[2] The purpose of this proposed rule is to revise the NRC regulations for the safe transportation of radioactive material to make them compatible with those of the International Atomic Energy Agency (IAEA) and thus with those of most major nuclear nations of the world. Changes in the proposed rule deal with the definitions of the package types and the contents being shipped in the packages. In addition to the package and contents specifications, there are changes in the proposed rule that will affect the structural aspects of a DOE package design. One major change is that, as part of the Hypothetical Accident Conditions specified in 10 CFR 71.73, the proposed rule has added a crush test following the free drop and before the puncture test. Another major change is a revision of the definitions used to define the hypothetical accident fire test to a definition directly tied to an actual fuel fire. The changes in the proposed rules are incorporated in this safety guide.

The performance-based requirements for package design and acceptance are based on radiological effectiveness rather than structural criteria. The only specifically structural criteria in 10 CFR 71 are concerned with tie-down and lifting devices, but there are general standards in 10 CFR 71 that influence the structural design. The radiological effectiveness criteria consist of three basic safety requirements:

1. Containment: Any radioactive material release must be restricted within the limits specified in 10 CFR 71. These limits are discussed in Subsect. 2.2.1.3.

2. Subcriticality: Criticality event must not occur. This topic is discussed in Subsect. 2.2.1.4.

3. Shielding: External radiation levels must be kept within the limits specified in 10 CFR 71. These limits are listed in Subsect. 2.2.1.5.

Typical DOE container designs are closely integrated: a single component may perform more than one function, and functions are often shared by more than one component. This complexity drives the design process to a series of decisions and compromises based on more than one performance criterion. Usually, the compromises involve materials selection or design features intended to simplify operation and maintenance of the container. For example, the optimum material for shielding is not the optimum material for impact absorption or thermal protection. In DOE containers, these functions are usually shared by the same components. Thus a compromise material or materials must be selected that will perform adequately to meet all three criteria. The selection of materials is fundamental to the container design and affects the entire design, whereas operational requirements are secondary.

In 10 CFR 71, the shipping packages for radioactive materials are classified as Type A or Type B packages, depending on the maximum activity of the radioactive contents. Most DOE weapons packages are Type B.

With the exception of several Type A Fissile Class II and III packages described in 10 CFR 71.18 through 71.22, all packages must demonstrate that the radiological criteria are satisfied in tests specified in 10 CFR 71.71 (Normal Conditions of Transport) and in 10 CFR 71.73 (Hypothetical Accident Conditions).

Except for the structural requirements for lifting and tie-down devices, the regulations in 10 CFR 71 do not specify any structural requirements or standards that the shipping packages must meet. Currently, no national codes are dedicated to the design and construction of Type B packaging. While not always directly applicable to the design of DOE containers, NRC has developed regulatory guides containing design recommendations for spent fuel casks. Because many casks following these recommendations have been approved by the NRC, the regulatory guides provide good guidance for DOE packaging design.[3]

NRC has adopted the philosophy of applying strict requirements and high margins of safety to packages with high levels of radioactivity. For example, Type B, Fissile Class III packages must meet stricter requirements and require higher margins of safety than Type A, Fissile Class I packages. Regulatory Guide 7.11 defines three categories of Type B packages according to the radioactivity levels of the contents.[4] For a specific radioactive isotope, Category I includes the highest levels and requires the highest margin of safety, whereas Categories II and III include the medium and low activity levels and therefore require lower margins of safety. Figure 2.2 shows the three categories and their associated package types and radioactivity levels of contents.

Two other regulatory guides specifically address structural aspects of the package design. Regulatory Guide 7.6[5] describes the design criteria for the structural analysis of shipping cask containment vessels and explains use of the "design-by-analysis" approach for Class I components from Section III of the American Society of Mechanical Engineers (ASME) Code[6] as a design criteria for the containment vessels. Regulatory Guide 7. 8[7] elaborates on the normal and accident test conditions specified in 10 CFR 71 and recommends the loading combinations for the structural analysis of shipping casks.

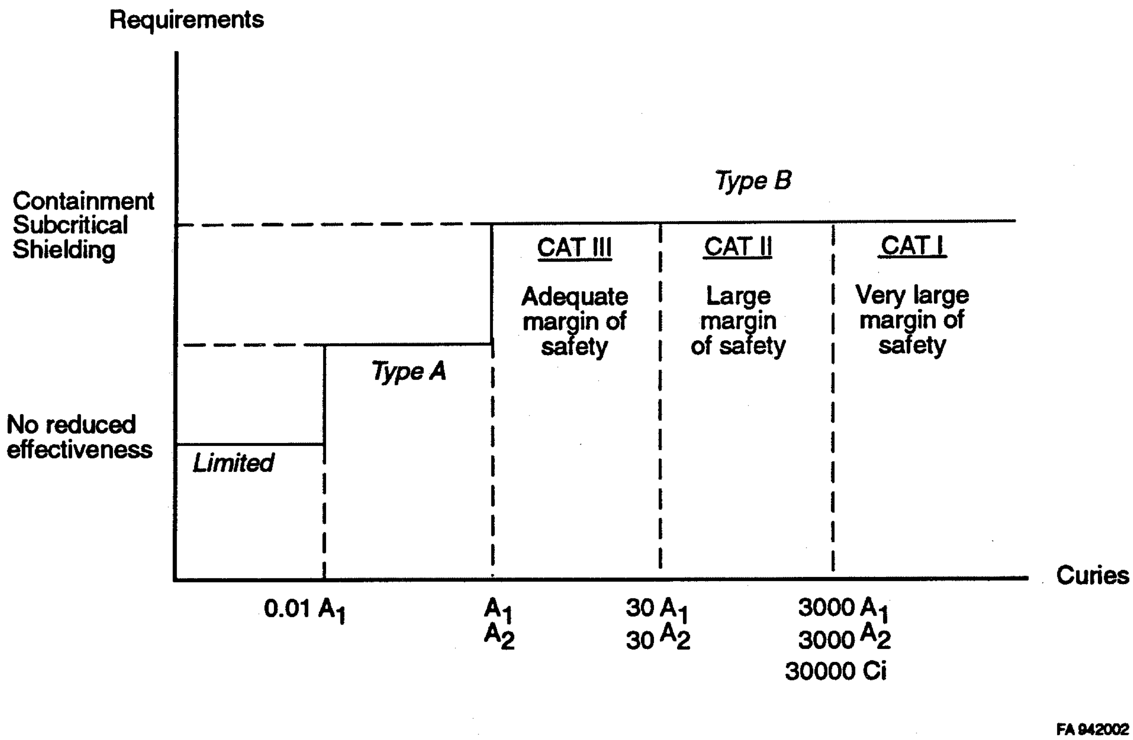


Fig. 2.2. Packaging types and activities of contents.

Plutonium air shipments are not currently being used by DOE. If such shipments are allowed in the future, the requirements of 10 CFR 71.64 concerning containment, external radiation, and criticality must be met. Plutonium air transport accident conditions are included in 10 CFR 71.74. Because these activities are no longer occurring, they are not addressed in this guide.

This guide will not specifically address special form material. This topic is addressed in 10 CFR 71.75 and 10 CFR 71.77.

2.2.1 Basic Safety Requirements

To protect public safety, public health, and the environment from the inherent risk of transporting radioactive materials, the shipping packages are required to meet, under both normal transportation and Hypothetical Accident Conditions, three basic safety requirements presented in 10 CFR 71:

- Adequate containment of radioactive materials
- Assurance of nuclear subcriticality
- Adequate shielding of the radiation emitted by the radioactive contents

Package component design and safety classification are guided by the safety requirements. The shipping package components can be divided into three safety groups according to these safety requirements. The first designated safety group, containment components, includes all components used to retain the radioactive contents in the packaging during transport. Containment components include the containment vessel, closure, seals, piping, and bolts. The second group, subcriticality components, includes all components used to control nuclear criticality during the transport of fissile materials in the packaging. The subcriticality components include neutron-absorbing materials such as boron carbide and the associated structures that retain the relative positions of the fissile and neutron absorber materials during transport. The third group, shielding and other safety components, includes all of the remaining safety-related components. In this group are gamma and neutron shielding; secondary containment seals, bolts, and closure; impact limiters; and lifting lugs and tie-down devices.

Components in each group must be evaluated for all applicable loading conditions. The structural, thermal, and radiological response of each component can affect the other components in the package. Chemical compatibility and corrosion properties of the materials must be considered. For details about the thermal, containment, subcriticality, and shielding aspects of the package design, refer to the chapters on those subjects. The following sections describe the functions of the safety component groups, the requirements specified in 10 CFR 71, and the structural aspects associated with meeting the safety requirements.

2.2.1.1 General standards for all packages

The general standards for all packages, listed in 10 CFR 71.43, contain requirements that influence the structural design of the package. The most structurally significant requirements concern containment and are discussed further in Subsect. 2.2.1.3.

Other general standards from 10 CFR 71.43 are also significant from a structural viewpoint in that they influence the closure design of the package as well as of the containment, the package thermal performance under normal operations, and the materials of the package and packaging.

2.2.1.2 Lifting and tie-down standards

The lifting and tie-down standards for all packages are listed in 10 CFR 71.45. The standards specify requirements for a minimum safety factor, failure of any lifting device under excessive load, and any other structural components of the package related to lifting the package.

The lifting and tie-down standards in 10 CFR 71.45 apply to devices that are structurally a part of the package. Most DOE Type B containers are not required to meet these requirements because they are not tie down or lifted by devices that are structurally part of the package.

The requirements of 49 CFR Part 393 Subparts 100 through 102 may be used for design of tie-down systems that are not a structural part of the package. The combination of tie-down devices shall be developed to keep the package secured and to prevent shifting under a loading equal to a static force applied to the center of gravity of the package with a vertical component equal to the weight of the package plus its contents, a horizontal component along the direction in which the vehicle travels of twice the weight of the package plus its contents, and a horizontal component in the transverse direction of twice the weight of the package plus its contents. Any hardware adapted for use with the package shall be utilized without generating stress in any material of the package in excess of its yield strength.

The requirements of RTD Standard F 8-11T[8] may also be used for the design of tie-down systems that are not a structural part of the package. Both 49 CFR Part 393 and the RTD Standard requirements are significantly less than the requirements in 10 CFR 71.45.

2.2.1.3 Containment

The containment requirements are listed in 10 CFR 71.51 and 71.71 for the Normal Conditions of Transport and in 10 CFR 71.51 and 71.73 for the Hypothetical Accident Conditions. Additional special requirements for plutonium shipment are described in 10 CFR 71.63.

Containment of radioactive material prevents contact between radioactive material and people or the environment. Typically, containment is provided by the integrity of an austenitic stainless steel containment vessel. A containment vessel usually has a bolted closure to accommodate the loading and unloading of contents. The closure contains a seal or seals which minimize leakage from the containment vessel to the environment. Penetrations of the containment which may be needed for operating purposes, such as back-filling with a tracer gas for leak testing or helium for heat transfer enhancement, are considered part of the containment system.

The function of all of the containment vessel and closure components is to maintain the containment boundary so that all the containment requirements are met under the normal transportation and accident conditions. The closures of penetrations, such as valves or sealed tubes, are also considered part of the containment system. The major loads are heat, internal pressure, and impact. Even though the regulations do not impose specific requirements on any structural components in terms of stress allowables or deformation limits, the containment boundary will be compromised if the structural components are overstressed or grossly distorted. Therefore, the structural components should be designed according to a well-established design standard such as the ASME Code as recommended in Regulatory Guide 7.6. Other codes and standards can be used as design criteria if they are as conservative as the ASME Code. All the loadings from the normal and accident conditions should be considered and combined as recommended in Regulatory Guide 7.8.

The general standards for all packages listed in 10 CFR 71.43 also contain requirements that influence the structural design of the package containment system. Several general requirements are concerned with pressure-relief valves and venting of the containment system. Usually, DOE packages do not require pressure-relief devices; however, the containment must be designed to handle any pressure excursions resulting from not having a relief system during either normal transport or accident conditions. Containment must be maintained if there is a pressure or temperature increase due to chemical reactions resulting from water inleakage, irradiation, or thermal effects on the package or content during either normal transport or accident conditions. Since the containment cannot be continuously vented, it must be designed to retain an internal pressure, even when intermittent pressure relief is provided. When valves or other relief systems are provided, they must be structurally protected against accident conditions or inadvertent operation in normal use.

The special requirements in 10 CFR 71.63 for plutonium shipment include a requirement that, under certain conditions, the material must be packed in a separate inner container placed within the outer packaging. This arrangement is usually referred to as double containment. The normal transportation and Hypothetical Accident Conditions are specified, and filters and mechanical cooling systems are not permitted in either condition.

Note that 10 CFR 71.63 exempts reactor fuel elements, metal or metal alloy, and other plutonium-bearing solids. In DOE containers the need for double containment has been determined on a case-by-case basis. Since the plutonium in DOE containers is usually in metallic form and often is contained entirely in a sealed subassembly, the plutonium is routinely shipped without a separate inner container. In designing a plutonium container, the decision to use double containment is one of the first decisions that must be made.

2.2.1.4 Subcriticality

Most DOE containers in the Weapons Safety Program are used for transporting fissile materials. The accident with the worst possible consequence for a fissile material is one which results in a criticality. Protection for criticality can be achieved by physical limitation on the amount of fissile materials in the package, adjusting and maintaining the geometry of the fissile contents being shipped, or providing neutron poisons to absorb sufficient neutrons to assure subcriticality.

Subcriticality design and performance requirements are described in 10 CFR parts 71.55 through 71.61. Criteria that influence the structural design of the package, include the following:

- First, the container with its contents must be designed so that it will remain subcritical if water leaks into the containment system or liquid contents leak out of the containment system.

- Second, an exception may be approved if appropriate measures are taken before each shipment to ensure that the containment system does not leak. In practice, this means that there must be double containment; breach of a single containment vessel, if undetected before shipping, can lead to a criticality accident. If double containment is used, there must be undetected breaches in both levels of the containment before a similar accident can occur.

- Third, under Normal Conditions of Transport the package must be designed and constructed so that it will remain subcritical and its geometric form will not be substantially altered.

- Fourth, under Hypothetical Accident Conditions the package must be designed and constructed and its contents limited so that the package will remain subcritical.

Exemptions to the requirements of 10 CFR 71.55 through 71.61 included in 10 CFR 71.53 are based on the actual composition, both the physical form and materials, of the shipment. Before beginning the design process, these exemptions should be investigated thoroughly. In most cases, however, they do not apply to DOE shipments.

Subcriticality safety components also include the structures that maintain safe geometry inside the container and the neutron absorber materials such as boron carbide. The primary structural concern is structural deformation that alters the configuration required for subcriticality. An important aspect of this concern is the interaction of the package internals and content with the containment system. If the containment system is breached from the inside by impact of the package internals during an accident, a criticality may occur due to subsequent flooding, even if the impact limiters eliminate damage by external forces. In addition, thermal effects on subcriticality structural component geometry and material properties and distribution as a result of the hypothetical accident are a major concern. If the critically safe geometry is lost or if neutron absorbing materials are degraded or displaced a criticality can result.

2.2.1.5 Shielding

A radiation shield is a barrier that absorbs ionizing energy or subatomic particles emanating from a radioactive source. Shielding against both gamma and neutron radiation may be needed in a package design. Shielding against the highly penetrating gamma radiation is achieved by using heavy, high-atomic-number materials such as lead, steel, or even neutron-free depleted uranium. These materials typically surround the containment vessel and are in turn enclosed within an outer steel shell. The function of the neutron shield is to attenuate the neutron dose from fissile materials. Neutron shield materials, usually hydrogenous materials such as water and polyethylene, typically surround the packaging on its exterior surfaces.

Shielding performance requirements are listed in 10 CFR 71.47 and 71.51. The regulations require that the package surface dose rate shall not exceed the specified value under Normal Conditions of Transport, as defined in 10 CFR 71.71; the transport index, which is defined in 10 CFR 71.4, shall not exceed 10. Under Hypothetical Accident Conditions, the package surface dose rate shall not exceed the specified value at the package surface.

The shielding safety components include the gamma and neutron shields. Frequently, gamma and neutron shields are used as both thermal insulation and impact absorbers. Therefore, the mechanical or thermal energy-absorption characteristics of these materials are important design parameters. The main structural concerns are the permanent deformation of the shields and the distortion of other structural components causing a reduction of thickness of the shield material or gaps to form in the shield material, which allows radiation streaming to occur. In addition, thermal effects on shielding during the hypothetical accident are of major concern. Often, the impact absorbers contain hydrogenous material which is helpful for shielding. The concern is that hydrogen is often lost by combustion during the hypothetical accident fire. Lead shielding is subject to slump or cold flow even under normal conditions, and this problem is aggravated at the higher temperatures encountered during the hypothetical fire. If slumping or other changes in the lead configuration occur, shielding integrity may be lost.

2.2.2 Performance Standards

Package approval standards in 10 CFR 71.41 state that to show compliance to the safety requirements of containment, subcriticality, and shielding, the effects on a package of the tests specified in 10 CFR 71.71(Normal Conditions of Transport) and the tests specified in 10 CFR 71.73 (Hypothetical Accident Conditions) must be evaluated by testing a sample package or scale model. Though the regulations specify the normal and accident conditions as test conditions and procedures, it is acceptable, for analytical evaluations, to translate the test conditions into loading conditions. In fact, some conditions are better suited to show compliance by testing, whereas others are more cost effective by analysis. See Sect. 2.5 for additional discussion of testing and analytical methods.

2.2.2.1 Normal Conditions of Transport

10 CFR 71.71 states that package designs for Normal Conditions of Transport may use separate test specimens for a free drop test, a compression test, and a penetration test; and each of these test specimens must be subjected to a water spray test before any other test. The code also specifies the initial conditions and the test conditions for heat, cold, pressure, and vibration. The test condition specification for each of the above tests (water spray, free drop, corner drop, compression, and penetration) are also included in the code. The corner drop test specification (10 CFR 71.71(c)(8)) is not anticipated to be applicable for DOE shipments.

2.2.2.2 Hypothetical Accident Conditions

10 CFR 71.73 states that tests for free drop, puncture, and thermal exposure must be conducted in the order specified for evaluation of Hypothetical Accident Conditions. In addition, an undamaged specimen must be used for a water immersion test, and a crush test must be included after the free drop test, as indicated by the proposed rule change.[2] This test is specified as follows:

> "... (2) Crush. Subjection of the specimen to a dynamic crush test by positioning the specimen on a flat, essentially unyielding, horizontal surface so as to suffer maximum damage by the drop of a 500 kg (1,100 lb) mass from 9 m (29.5 ft) onto the specimen. The mass must consist of a solid mild steel plate 1 m (3.28 ft) by 1 m and must fall in

a horizontal attitude. The crush test is required only when the specimen has a mass not greater than 500 kg (1,100 lb), an overall density not greater than 1,000 kg/m^3 (62.4 lb/ft^3) based on external dimensions, and radioactive contents greater than 1,000 A_2 not as special form radioactive material."

2.3 STRUCTURAL DESIGN CRITERIA

The regulations in 10 CFR 71 do not specify any structural requirements for shipping packages other than those for lifting and tie-down devices. The NRC has developed regulatory guides for the design of spent fuel casks. These guides provide good design guidance for weapons components and special assembly package design. The following subsections contain discussions of the design guidance provided in the regulatory guides, the current design criteria for weapons packages that have been developed from national codes and standards referenced by these regulatory guides, and proposed changes to the current design criteria for weapons packages.

2.3.1 Structural Criteria from Regulatory Guides

Shipping casks of radioactive materials are designed and used by the nuclear power industry. One major use is transporting spent fuel assemblies from the nuclear power plants of U.S. utilities. In the absence of a general set of design criteria for shipping casks of radioactive materials, the NRC has developed numerous regulatory guides to provide design recommendations for spent fuel casks. Because the utility companies are familiar with the design criteria for nuclear components in Section III of the ASME Code[6] and because the safety concerns in dealing with radioactive materials are similar in shipping packages as those in Section III, the regulatory guides have adopted portions of the ASME Code to form a set of structural design criteria for shipping casks. Using the same philosophy of component safety classifications as those in the ASME Code, the NRC applies stricter requirements and higher margins of safety to packages with higher levels of radioactivity. Regulatory Guides 7.6, 7.8, and 7.11 specifically address structural aspects of the package design.

Regulatory Guide 7.6

Regulatory Guide 7.6[5] describes the design criteria for the structural analysis of shipping cask containment vessels and explains the "design by analysis" approach for Class I components from Section III, Subsection NB of the ASME Code. Design criteria for Level A service limits and Level D service limits from the ASME Code are adopted in Regulatory Guide 7.6 for normal and accident conditions, respectively. Regulatory Guide 7.6 adopts the ASME Code concepts concerning stress categories and assigning different stress intensity limits according to the significance of stress categories. The guide uses linear elastic analysis for design and allows the principle of superposition to be used for load combinations. In recommending linear elastic analysis, Regulatory Guide 7.6 does not preclude an appropriate nonlinear treatment of other cask components, such as lead shielding and impact limiters. Figure 2.3 outlines a procedure for identifying and combining linear elastic loads, classifying stresses, and comparing the stress results with the acceptance criteria specified in Regulatory Guide 7.6.

Regulatory Guide 7.8

Regulatory Guide 7.8[7] identifies the normal transport and hypothetical accident test conditions specified in 10 CFR 71 and recommends the loading combinations for the structural analysis of shipping casks. Table 1 of Regulatory Guide 7.8, Summary of Load Combinations for Normal and Hypothetical Accident Conditions of Transport, is duplicated in Table 2.2.

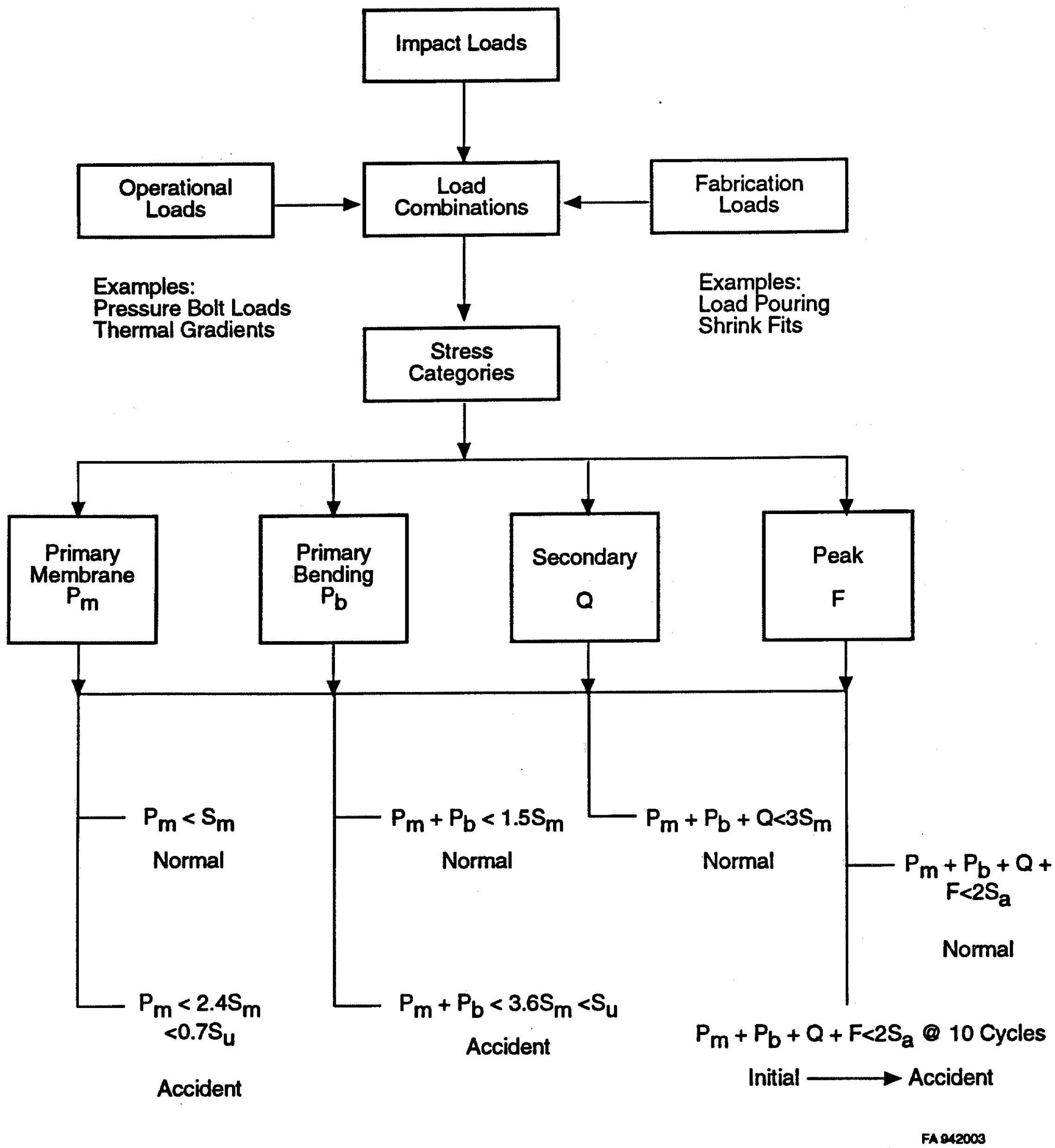


Figure 2.3 Linear elastic load combinations and stress intensity limits.

Table 2.2. Summary of load combinations for normal and hypothetical accident conditions of transport

Normal or Accident Condition	Applicable initial condition								
	Ambient Temperature		Insolation		Decay heat		Internal pressure		Fabrication Stresses
	100°F	-20°F	Max	0	Max	0	Max	Min	
NORMAL CONDITIONS (analyze separately)									
Hot environment-100°F ambient temperature			X		X		X		X
Cold environment--40°F ambient temperature				X		X		X	X
Increased external pressure - 20 psia		X		X		X		X	X
Minimum external pressure - 3.5 psia	X		X		X		X		X
Vibration and shock: Normally incident to the mode of transport	X		X		X		X		X
		X		X		X		X	X
Free drop: 1-ft drop	X		X		X		X		X
		X		X		X		X	X
ACCIDENT CONDITIONS (apply sequentially)									
Free drop: 30-ft drop	X		X		X		X		X
		X		X		X		X	X
Puncture: drop onto bar	X		X		X		X		X
		X		X		X		X	X
Thermal: fire accident	X		X		X		X		X

Regulatory Guide 7.11

Regulatory Guide 7.11[4] defines three categories of Type B packages according to their levels of content. The description of these categories and component safety groups is presented in Sect. 2.2, and the three categories and their associated package types and levels of contents are shown in Fig. 2.2. A set of standards for the design, manufacture, use, and maintenance for the three component safety groups in each category is discussed in Regulatory Guide 7.11. For a specific radioisotope, Category I includes the highest levels of activity to be transported and requires the highest margins of safety, while Categories II and III include the medium and low activity levels and therefore require lower margins of safety.

2.3.2 Structural Design Criteria for Weapons Components

The ASME Code, Section III[6] provides rules for nuclear power plant components in areas of design, fabrication, operation, maintenance, and quality assurance. Section III is in two divisions. The rules in Division 1 are for metallic containment structures and those in Division 2 are for the concrete reactor vessel and containment. Division 1 is appropriate for DOE packages since most of them are metallic. Section III identifies nuclear components as Class 1, Class 2, and Class 3 components according to decreasing order of importance to safety. Stricter material specifications and more detailed fatigue analyses are required for Class 1 components. Consequently, Section III provides separate design criteria for the different classes of components.

Using the terminology found in regulatory guides, Category I transport packages are equivalent to ASME Class 1 components, and Category II packages are equivalent to ASME Class 3 components. Similarly, Service Level A and Service Level D are equivalent to the Normal Conditions of Transport

Table 2.3. Structural design criteria (based on ASME Code)

Component safety group	Container contents		
	Category I	Category II	Category III
Containment	Section III Subsection NB	Section III Subsection ND	Section VIII Division 1
Subcriticality	Section III, Subsection NG		
Shielding and other	Section VIII, Division 1 or Section III, Subsection NF		

and Hypothetical Accident Conditions for packages, respectively. Table 2.3 presents the applicable ASME structural code design criteria, by category, for containment, subcriticality and shielding. Category III uses ASME Code, Section VIII for containment and all other categories use ASME Code, Section III. To aid in understanding the design philosophy of ASME Code, Section III, the basis for the design-by-analysis approach is presented below.

ASME Code, Section III

Section III permits use of different design approaches, "design by formula," and "design by analysis." The design-by-formula approach is the "cookbook" method. General formulas are provided for vessel, pump, valve and piping designs. The designs are made according to step-by-step rules, and the allowables for the design-by-formula approach are necessarily conservative. By contrast, the design-by-analysis approach requires detailed analyses; therefore, the allowables can be set higher. Some designs that are not qualified using the design-by-formula approach may qualify with the design-by-analysis approach. Once the design approach is decided, all the applicable rules should be followed and the design must be consistent within the approach.

With advances in analytical and experimental techniques, the design-by-analysis approach is more attractive than the design-by-formula approach in yielding a well-balanced design for critical safety components. Additionally, it is possible to determine local stresses in a structure in detail. It is therefore unreasonable to retain the same allowables throughout the structure because high local stresses do not constitute a global structural failure. The rationale of assigning different allowables for different types of stress is, "A calculated value of stress means little until it is associated with its location and distribution in the structure and with the type of loading which produced it. Different types of stress have different degrees of significance and must, therefore, be assigned different allowable values. For example, the average hoop stress through the thickness of the wall of a vessel due to internal pressure must be held at a lower value than the stress at the root of a notch in the wall. "[6] Likewise, thermal stress allowables can be higher than those due to dead weight or pressure. Therefore, the design-by-analysis approach requires dividing stresses into categories and assigning different allowable values to different groups of categories.

Section III divides stresses into three groups: primary stress (P), secondary stress (Q), and peak stress (F). “Primary stress is a stress developed by the imposed loading which is necessary to satisfy the laws of equilibrium between external and internal forces and moments. The basic characteristic of a primary stress is that is not self-limiting. If a primary stress exceeds the yield strength of the material through the entire thickness, the prevention of failure is entirely dependent on the strain-hardening properties of the material."[6] The primary stress can be further divided into three types of stresses according to spatial distributions: general primary membrane stress (P_m), local primary membrane stress (P_1), and primary bending stress (P_b). Examples of primary stress are stresses due to impact loads, internal pressure, and bolt loads. The stress state caused by these loads is divided into membrane and bending components.

"Secondary stress is a stress developed by the self-constraint of a structure. It must satisfy an imposed strain pattern rather than being in equilibrium with an external load. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the discontinuity conditions or thermal expansions which cause the stress to occur."[6] "Peak stress is the highest stress in the region under consideration. The basic characteristic of a peak stress is that it causes no significant distortion and is objectionable mostly as a possible of fatigue failure."[6]

The philosophy of the design-by-analysis approach is based on linear elastic analysis. Failure by the maximum-shear-stress theory of failure is used as the basis for stress allowables. Stress categories are calculated as "stress intensity" before they are compared with the allowables which are called "stress intensity limits" in the code. Stress intensity is defined as twice the maximum shear stress at a point. If the principal stresses are S_1, S_2 ,and S_3 while $S_1 > S_2 > S_3$ (algebraically), the maximum shear stress is 1/2 x (S_1 - S_3) and it must be less than or equal to the stress intensity limit for a given or combined loading condition. Stress allowables, that is stress intensity limits, are not expressed in terms of the yield strength of the material but rather as multiples of S_m. S_m is the stress intensity limit for general primary membrane stress, and values of it for various metal alloys are tabulated in Section II, Part D of the code.

Yield strength is not a sufficient criterion in determining the allowable stress because of a wide range of ductility and strain-hardening properties in materials. To prevent unsafe designs in materials with low ductility and in materials with high yield-to-tensile ratios, the ASME Code requires the stress allowable to be equal to or less than the smaller of two-thirds of the yield strength or one-third of the ultimate tensile strength. Table 2.4 summarizes the basic stress intensity limits and the multiples of yield strength and ultimate strength that these limits do not exceed for four stress categories: general primary membrane, local primary membrane, primary membrane plus primary bending, and primary plus secondary.

The stress limit for each stress category is related to the potential failure mode. The primary stress limits aim to prevent plastic deformation and to give a nominal factor of safety on the ductile burst pressure, while the primary plus secondary stress limits are intended to prevent excessive plastic deformation and collapse. Finally, the peak stress limit is intended to prevent fatigue failure as a result of cyclic loadings. The stress limits for P_m are more conservative than those for for $P_1 + P_b$. Since most stress states in a package component are a combination of membrane and bending stresses, these stresses

Table 2.4. Basic stress intensity limits

Stress category	Stress limit	Allowable stress intensity: Based on yield S	Allowable stress intensity: Based on tensile S
General primary membrane (P_m)	S_m	2/3 S_y	1/3 S_u
Local primary membrane (P_l)	1.5 S_m	S_y	1/2 S_u
Primary membrane plus primary bending ($P_l + P_b$)	1.5 S_m	S_y	1/2 S_u
Primary plus secondary ($P_l + P_b + Q$)	3.0 S_m	S_y	S_u

must be separated into different parts. In general, the maximum membrane stress occurs at the neutral axis where the bending stress is zero and the maximum membrane plus bending stress occurs at the extreme fibers. A conservative approach for evaluating stresses is to use the P_m limits for all stress states without separating them into membrane and bending components. In some cases this approach may not be appropriate and the $P_1 + P_b + Q$ limits must be used.

In the code, all subsections in Section III and Section VIII, *Rules for Construction of Pressure Vessels*, Division 1, have provisions for the design by formula approach. The design-by-formula approach requires less rigorous analysis than does the design-by-analysis approach. For less critical safety components, the design-by-formula approach may be preferred because of its simplified procedures for the design. One advantage in using design by formula is that direct stresses, not stress intensities, are used to compare with the stress allowables. Stress intensity calculations and determination of stress categories are omitted completely in the design by formula approach. The stress allowables (S), which are different from stress intensity allowables (S_m), for various metal alloys are also tabulated in Section II, Part D of the code.

ASME Code, Section VIII

ASME Code, Section VIII, Division 1 has subsections devoted to the requirements for the design criteria based on the method used for fabricating the pressure vessel and on methods used for classes of construction material for the pressure vessel. The example analyses presented in Appendices A through F show the use of Section VIII of the ASME Code for the design of a Category III pressure vessel.

2.3.3 Proposed Changes to the Design Criteria for Weapons Components

Many shipping casks have been designed according to the recommendations in the regulatory guides and have been approved by the NRC. However, design criteria from other design codes may also be used as long as they can be justified to be as conservative as the ASME Code. Not all the design criteria in the ASME Code for nuclear components are directly applicable to shipping-package design. For example, pressure loads are the primary design loads in nuclear components, but impact loads are the primary design drivers for the shipping packages. High operating temperatures and cyclic loadings, while major design concerns for nuclear components, are low design factors for transport packagings. For these reasons, a special working group, NUPACK, was formed to develop applicable rules for shipping packages of radioactive materials[9]. Until applicable codes and standards are developed, the regulatory guides and the applicable ASME Code structural design criteria identified in Table 2.3 (or equivalent) can serve as a set of guidelines and design criteria for the DOE packaging design.

2.4 STRUCTURAL DESIGN GUIDELINES

The main package components of a drum-type container are package internals, containment system, shielding, impact limiters/thermal insulation, and tie-down and lifting devices. The integration of these components into a functional design requires a significant effort. Containment performance and radiological protection are the main drivers in the design of a DOE container. Very few components of a DOE container serve only one function, and the design is inevitably a series of compromises.

This section emphasizes, for each package component, the function, loading conditions, design details, and validation methods normally utilized for the design. The intent is to provide consistent and well-understood techniques for designing the structural aspects of a package.

Section 2.4.1 describes the functions of the major structural components and discusses their integration in the design process. Section 2.4.2 discusses the primary function of each major component as related to the regulatory requirements of 10 CFR 71. Sections 2.4.3, 2.4.4, and 2.4.5 discuss the preliminary design, detailed design, and design validation.

Section 2.4.6 briefly addresses quality assurance issues in the design and validation process. Section 2.4.7, Structural Design Examples, shows specific examples of existing drum-type containers, their unique design features, and some lessons learned during development. These examples may be valuable in developing new container designs.

This chapter addresses packaging structural design. Chapter 10, Materials and Fabrication addresses the materials and fabrication techniques commonly used in DOE containers.

2.4.1 Principal Structural Components

The principal structural components of a drum-type container are the containment system, the impact limiter/thermal insulation, shielding, packaging internals, and tie-down and lifting devices. The components interact and most have more than one function. The functions of each component are described in the following subsections.

2.4.1.1 Containment system

Containment system refers to all items whose primary function is to provide and maintain the contamment boundary around the contents being transported and around other package internals. Chapter 4, Containment, of this design safety guide contains additional information. The containment system maintains the necessary geometric configuration to assure that the contents remain subcritical, protects the contents and package internals from damage, keeps the temperature of the contents and the package internals low enough to prevent damages due to heat or pressure, and assures proper shielding geometry. The containment system also provides contamination protection for personnel working with the packaging and for the packaging itself and other convenience packaging features used during preparation and receiving.

2.4.1.2 Impact limiters/thermal insulation

For drum-type containers, impact limiters or cushioning materials are often utilized for thermal insulation as well. If the limiter is also utilized for thermal insulation, close coordination between structural designer and the thermal designer/analyst is necessary to select final materials and thicknesses needed to meet the combined requirements. Chapter 3, Thermal Aspects, of this design safety guide contains information on thermal performance issues.

Impact limiters also may play a key role in preventing criticality accidents, if the limiter size is a key determinant in the array spacing of containers. If this is the case, the impact-limiter design must ensure that deformation during normal or Hypothetical Accident Conditions is limited to an amount which ensures subcriticality at all times. In addition, hydrogenous or other materials in the impact limiter may also be involved in preventing criticality accidents. (See discussion on hydrogenous materials in Subsect. 2.2.1.5.) If hydrogenous or other materials in the impact limiter are required, the impact-limiter design must ensure that a sufficient amount of the material remains during and after normal or Hypothetical Accident Conditions.

Similarly, impact limiters may also function in radiation shielding. If this is the case, the impact limiter design must ensure that deformation and material loss during normal or Hypothetical Accident Conditions does not result in a loss of shielding.

Since most DOE containers are drum-type, this design safety guide concentrates on designs in which the impact-limiter material is confined to the inside of the drum. Containers with external impact limiters attached to the extremities of the package are not discussed.

2.4.1.3 Shielding

The purpose of shielding is to attenuate the gamma and neutron radiations emitted by radioactive decay of the contents. In DOE containers, the structural parts of the containment are almost always sufficient to provide shielding of short-ranged radiation, such as alpha and beta, and they are usually sufficient to reduce gamma and neutron radiation to acceptable levels. Occasionally, additional high-density material, such as lead or depleted uranium, must be added to reduce gamma or neutron doses. In these cases the design may be dictated by radiation attenuation requirements rather than structural considerations. Nevertheless, the shielded containment system must preserve the shielding capability under both normal and accident conditions. Chapter 5, Radiation Shielding, of this design safety guide contains information on shielding performance issues for DOE containers.

2.4.1.4 Package internals

Package internals refers to items inside the containment boundary other than the actual contents that are being transported. When double containment is used, for purposes of this design safety guide, the package internals is expanded to include all items inside the outer containment boundary other than the actual contents that are being transported. Included are the inner containment boundary and items inside the inner containment boundary other than the actual contents that are being transported, as well as any material between the inner and outer containment boundaries.

The package internals have, as a minimum, the four following essential functions: 1) they must maintain the necessary geometric configuration for the contents of the containment vessel to assure that the contents remain subcritical under all loading conditions; 2) they must protect the containment boundary from damage by the contents and the package internals themselves under all loading conditions; 3) maintain the temperature of the contents, the containment boundary, and the package internals to a low enough level to prevent damage by heat or pressure; and 4) assure proper shielding geometry. Although shielding is generally external to the containment boundary in DOE packages, it is not constrained from being part of the package internals.

Other supplementary functions of the package internals include physical and thermal protection of the contents during shipping, contamination protection for personnel working with the packaging and for the packaging itself, and other convenience packaging features used during preparation and receiving.

2.4.1.5 Tie-down and lifting devices

The tie-down system is the arrangement of tie-down hardware that secures the package onto the vehicle. The requirements in 10 CFR 71.45 are interpreted as applicable only for the tie-down devices that are a structural part of the package. The requirements in 49 CFR Part 393 and RTD Standard F 8-11T may apply to tie-down systems that are not a structural part of the package.

Lifting devices are those structural elements that are permanently attached to the container and serve as the interface between the container and lifting mechanism, such as a crane or forklift, during loading and unloading from the transportation vehicle. Most drum-type packages weigh less than 454 kg (1000 lb) and usually do not have permanent lifting devices. Some guidance for the design of lifting devices for drum-type packages is provided by Smallwood.[10]

2.4.2 Design Requirements

The following information is intended to guide the designer of a container to the appropriate regulatory requirements. Section 2.2 discusses the safety requirements information contained in 10 CFR 71.

There are some general standards in 10 CFR 71.43 that influence the structural design of all package components. They are discussed more fully in Sect. 2.2.

2.4.2.1 Containment system

Containment requirements are discussed in Subsect. 2.2.1.3 and are cited in 10 CFR 71.51 and 71.71 for the Normal Conditions of Transport and in 10 CFR 71.51 and 71.73 for the Hypothetical Accident Conditions. Additional special requirements for plutonium shipment are described in 10 CFR 71.63.

Subcriticality design and performance requirements, described in 10 CFR parts 71.55 through 71.61, are summarized in Sect. 2.2.1.4. Subcriticality is required in all conditions, including water leakage into the containment system or leakage of a package’s liquid contents from the containment system. An exception is provided in 10 CFR 71.55(b) and 10 CFR 71.55(c), which states the condition for approval of a package which does not meet this requirement. In practice, this condition means that there must be double containment, since breach of a single containment vessel, if undetected before shipping, can lead to a criticality accident. If double containment is used, there must be undetected breaches in both levels of the containment before a similar accident can occur.

The general requirements 10 CFR 71.43 include several that apply specifically to the containment system. The containment system must be securely closed by a positive fastening device that cannot be opened unintentionally or by pressure changes within the package. Failure of package valves or other devices that allow radioactive contents to escape must be protected against unauthorized operation and, except for a pressure relief device, must be provided with an enclosure to prevent any leakage. A package must not incorporate a feature which allows continuous venting during transport.

Several special requirements in 10 CFR 71.63 for plutonium shipment are outlined in Subsect. 2.2.1.3. In some cases the plutonium must be packed in a separate inner container and, placed within the containment system that meets all of the requirements of 10 CFR 71, Subparts E and F for packaging of material in normal form. In DOE containers, the need for double containment has been determined by working with DOE on a case-by-case basis. Since the plutonium in DOE containers is usually in metallic form and often is contained entirely in a sealed subassembly, the plutonium is routinely shipped without a separate inner container. In designing a plutonium container, the decision on use of double containment is obviously one of the first decisions that must be made.

2.4.2.2 Impact limiters/thermal insulation

The design of the impact limiter/thermal insulation is largely dictated by the requirements of 10 CFR 71.71 and 10 CFR 71.73 for resistance to damage during normal transport and Hypothetical Accident Conditions. These requirements are discussed in Subsects. 2.2.2.1 and 2.2.2.2, respectively.

2.4.2.3 Shielding

Shielding performance requirements defined in 10 CFR 71.47 and 10 CFR 71.51 are discussed in Subsect. 2.2.1.5. There are also requirements for shielding during and after Hypothetical Accident Conditions in 10 CFR 71.73.

2.4.2.4 Packaging internals

The design requirements for packaging internals are similar to those discussed in Subsect. 2.4.2.1 for the containment system.

2.4.2.5 Tie-down and lifting devices

The requirements for lifting and tie-down devices are discussed in Subsect. 2.2.1.2.

2.4.3 Preliminary Design

The initial step in a package design is the determination of the following information:

- Size, shape, weight, materials, and mass properties of the content of the package
- Criticality requirements of the content of the package
- Shielding requirements of the content of the package
- Requirement (if any) for double containment
- Heat generation of the content of the package
- Maximum allowable temperature for the content of the package

Based on the information gathered, decisions must be made on the configuration of package. After the development of a set of alternative designs, the alternatives should be compared and the most promising candidate selected. The selection of the best concept should be based on the following criteria:

- Performance against the regulatory requirements.

- The degree of technological risk involved in the design. How much new technology will be developed and used in the design and is there any experience in the use of the new technology in previous container designs?

- The usability of the design. Is it difficult or time consuming to assemble correctly, or unnecessarily complex with many small parts to lose?

- The producibility of the design. This includes cost, schedule, and technological challenges involved in the production of the container.

- The basis for qualification of the design components. Identification of those components which can be qualified by analysis only, those which can be qualified only by testing, and those requiring a combination of analysis and testing.

The following sections include guidance on factors involved in the preliminary design process.

2.4.3.1 Containment system

An important step early in the development of the design containment system is the selection of the materials. The materials used must perform adequately under both Normal and Hypothetical Accident Conditions. From a thermal standpoint, this means that the materials must be mechanically sound at normal transport temperatures from -40^0C (-40^0F) to hypothetical accident condition temperatures of up to several hundred degrees. Because of hypothetical accident impact loads, which can occur at very low temperatures, brittle fracture of the containment materials including the vessel and the bolts is a major concern.

Hydrocarbon materials used in seals should also be carefully selected for performance at temperature extremes. One consideration is thermal decomposition at high temperatures, both during normal transport and Hypothetical Accident Conditions, and the other consideration is seal performance due to loss of resilience at low temperature. While not normally a problem encountered in DOE packages, radiolysis of hydrocarbons is a concern at higher radiation levels and may also affect seals.

Selection of seal materials is a very complex problem. In addition to the thermal and mechanical assaults on the seals, several other questions arise. Are the seals reusable? How often are the seals to be reused and what are the replacement criteria? Are there any special chemical compatibility requirements? If helium or halogen leak checking is to be used, how permeable are the seals to helium or halogen tracer gases? If the container is used to ship tritium, how permeable are the seals to tritium?

The materials selected for the containment system must be compatible under all conditions of transport. There must be no significant chemical, galvanic, or other reaction among the package components and contents. Decomposition of materials in the package internals can cause pressurization of the containment boundary if gases are generated. If the mechanical properties or a physical configuration of the package internals change, they may fail to protect the integrity of the containment boundary.

The containment boundary performs a shielding function against radiation, but this is not normally a primary function. The selection or thickness of material can, in rare instances, be influenced by shielding requirements. The materials selected must be compatible with the design code chosen as the basis for the structural design of the containment system. For example, only materials listed in the ASME Code are acceptable if the AMSE Code is the basis for the containment design.

In evaluating the mechanical requirements for the containment boundary, the primary loads to be considered are inertial loads due to Normal and Hypothetical Accident Conditions, vibration, and differential thermal expansion. The load paths to the containment boundary from externally applied forces and from the package contents through and including the package internals must be evaluated very carefully.

When the contents of the package generate a significant amount of heat, it may be necessary to devise a way to remove heat in order to prevent high-temperature damage to the containment, the package internals, or even to the contents. At the same time it is necessary to prevent external heat sources from transferring heat into the package during accidents.

2.4.3.2 Impact limiters/thermal insulation

To protect the containment and the internal payload from excessive stresses during both normal transport and a hypothetical accident free fall of 9 m (30 ft) onto a horizontal, essentially unyielding surface, impact limiters or cushioning materials are normally provided between the outside shell of the container and the containment vessel. Additional cushioning material may also be used inside the containment to reduce imposed impact induced stresses. This additional material is discussed in Subsect. 2.4.3.4, Package Internals.

Impact protection should also be designed to maintain other package safety features. In an accident, the impact limiter deforms and absorbs energy. It is particularly important to protect containment vessel closures from deformation.

The energy-absorbing device may be temporarily attached to the package for shipment only, or it may be an integral part of the package design. The device may be made of wood, foam, crushable fibrous material, crushable metal configured in a honeycomb matrix, or metal fins which might be designed to bend at a particular force.

Typically, the impact limiters/thermal insulation of a DOE container consists of an outer metal drum filled with an impact-absorbing material. The containment vessel is placed in a cavity in this material. The metal drum is typically a standard heavy duty drum adapted for this use. In the past, Department of Transportation 17H drums were frequently used. These drums are now being phased out in favor of performance-based designs such as lAl drums described in 49 CFR Part 178.504. In the past, a forged bolt type lock ring was used with removable head drums, but recent experience testing indicates that a bolt-on lid should be used on drums weighing more than 227 kg (500 lb) and should be considered at even lower weights. Keg-type drum designs are also used.

In DOE drum-type containers, the impact-absorbing material also functions as thermal insulation for the package during both normal transport and the hazardous accident fire test. Thus, superior high-temperature performance is required and the structural analyst must work very closely with the thermal analyst. Wood and Celotex™ have been widely used for this application, and have worked satisfactorily. They do have the drawback of being combustible materials which function by absorbing heat energy during pyrolysis of the hydrocarbons in the absence of oxygen and by the boiling off of their considerable moisture content. Obviously, they must be protected from exposure to air during the fire, while at the same time gases that are generated must be vented off. This is often accomplished by drilling vent holes in the outer drum. These holes, covered with plastic tape or plugs to prevent water from entering during normal transport, melt in a fire to provide venting. Celotex™ in particular must be protected from water damage, because it is made of sugar cane fibers in a composite matrix with a water soluble glue and can lose its mechanical properties rapidly when wet. This composite nature of Celotex™ is advantageous in other ways; however, the material has very good crushing properties under impact and is not as stiff or anisotropic as are other materials. Fiber insulating materials, foams, and honeycomb are also used in impact limiters/thermal insulation.

The specific loading conditions for which the impact limiter must be designed are presented in Sect. 2.2. The limiter must also be designed so that other tests specified in 10 CFR 71 do not impair the ability of the limiter to function in possible future impact conditions. For example, the water spray test must not cause degradation of the limiter material. For some materials (such as Celotex™) this means that the limiter must be enclosed in a covering material which will not be degraded by water. The impact limiter must also be designed to tolerate free drops of up to 1 m (depending on container weight), without consequential damage. Experience indicates that only minor provisions are required to meet normal operating conditions.

In selecting limiter material, environmental conditions under both normal and accident conditions must be considered. If the limiter will be exposed to sunlight, heat, or water, then the possible degradation due to these effects must be considered. Material selection must also consider size restrictions (a small limiter must be stiffer than a large limiter in order to afford the same energy absorption capacity), weight restrictions, cost, and safety.

Appropriate design data and design technique are closely related. The data required for design are contingent on the selection of a design technique. A conservative sizing calculation does not require as complete a set of material property data as does a dynamic, nonlinear, finite-element model. Conversely, if dynamic material properties are not available, it is not practical to use a sophisticated dynamic analysis.

Preliminary design information for a number of common materials are discussed in the following paragraphs.

Foam impact limiter materials

Foams have characteristics that can make them attractive choices for impact limiter applications. Foams can be manufactured to be essentially isotropic and thus do not require concerns about questions concerning the direction in which the impact load is applied. In an accident the orientation of the container is unknown. Consequently, the energy absorbing structure used must be able to withstand the application of an impact load from any direction. The isotropy of foam materials generally meets that requirement.

To perform optimally, an impact limiter must absorb a maximum amount of energy while transferring a minimum peak force to the shipping container. The energy absorbed during impact is equal to the work done by the impact force as it crushes the foam material. The work done by the impact force is determined by computing the area under the plot of impact force versus displacement or by integrating the specific energy absorption in the foam over the volume of material which is active during the impact process. The optimum impact limiter maximizes the absorbed energy while minimizing the magnitude of the impact force. This requirement suggests that the ideal form for the force versus deflection curve is rectangular, thus indicating that the force maintains a constant minimum throughout the impact process. Most foams, when loaded in compression, display a stress-strain curve with small initial elastic region up to about 5% strain terminated by yielding. Yielding is followed by a region of relatively constant stress with increasing strain. This region extends to strains of 50 to 70% depending on density after which the foam "locks up," and the stress rises rapidly with increasing strain. A typical stress-strain curve for a foam is shown in Fig. 2.4. The compressive strength of foam usually has its maximum at low temperatures and decreases continuously for higher temperatures.

Properties of foams are determined by the various materials from which they are made. Manufacturers' data usually provide complete information for the analysis of rigid polyurethane foams, including temperature effects and variability of material properties.

Design optimization using foam energy absorbers requires the following considerations. For a given weight to be protected, a given drop height, and a given thickness of foam, some foams are too

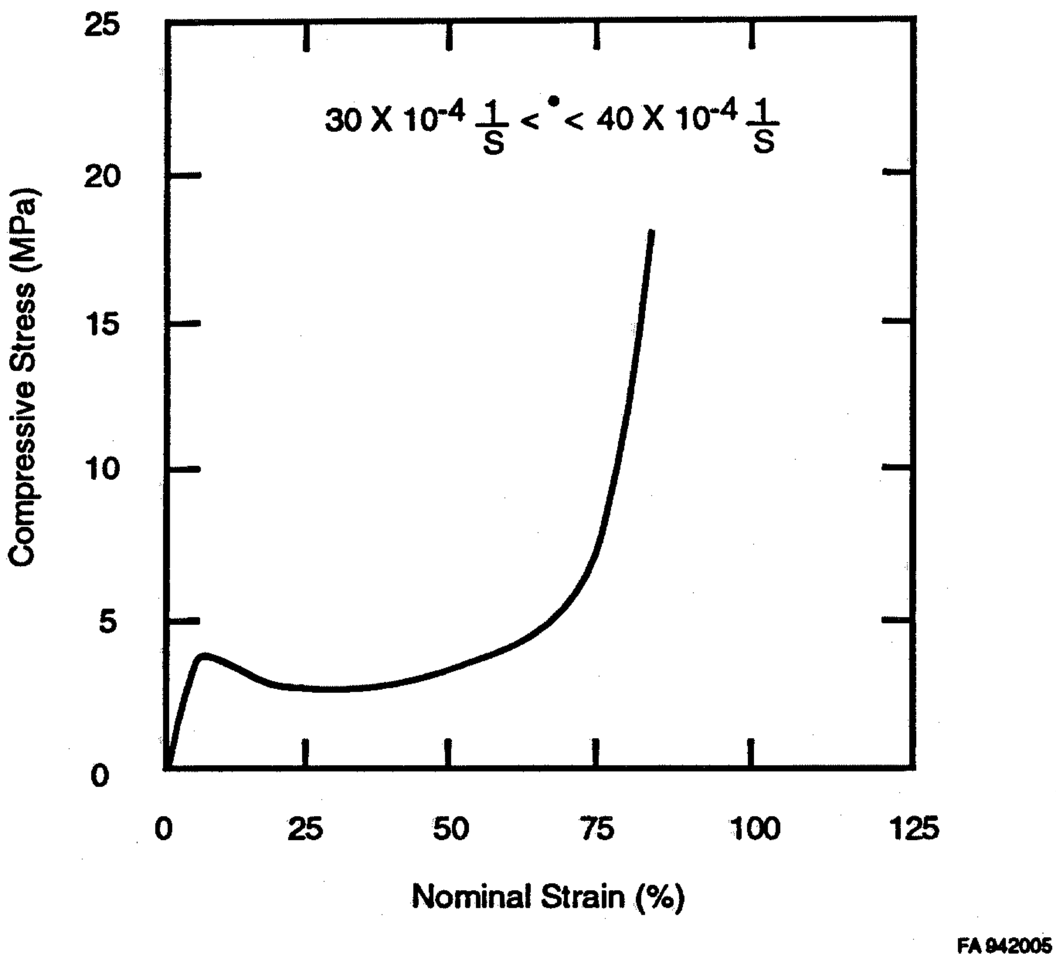


Fig. 2.4. Typical foam stress/strain curve.

stiff and stop the weight before the ultimate strain of the foam is attained; other foams are so soft that the weight does not stop until the foam reaches lock-up. Between these two extremes is the optimum design in which maximum strain in the foam is achieved without penetrating the lock-up region. This design maximizes the use of the strain range for which the stress remains relatively constant, consistent with minimum acceleration. From a design perspective, this requires specifying the contact area and thickness of the foam as well as choosing the foam yield stress to achieve the optimum design. The steps in the design procedure are:

1. Specification of the weight of the container to be protected by cushioning

2. Specification of the height of free drop to be experienced (30 ft for Hypothetical Accident Conditions)

3. Choosing a reasonable thickness for the foam

4. Selecting a foam resulting in the acceptable peak acceleration during impact

Wood impact limiter materials

In spite of being orthotopic, wood ranks among the highest of all materials in specific energy absorption, which makes it popular for protecting container internal components. Wood has a wider flat portion of the stress/strain curve than most impact-absorbing materials. All material properties of wood, however, vary depending on the angle from the direction of grain in the wood. The grain is not always parallel to the same direction throughout an entire piece of wood but may vary about a mean direction. Wood with a variation in grain angle, relative to the principal grain direction, greater than 7% should be avoided to prevent a reduction in compression strength. Celotex™ is a "wood" product made of sugar cane fibers in a composite matrix with a water-soluble glue. Thus Celotex™ has the advantage of very good crushing properties under impact and is not as stiff or as anisotropic as other wood materials.

Mechanical properties are also affected by the moisture content and temperature of the wood. Drier wood is usually stronger, and impact absorbers are made of dried wood enclosed in a protective cover. Celotex™ in particular must be protected from water damage, because it is made of sugar cane fibers in a composite matrix with a water-soluble glue and can lose its mechanical properties rapidly when wet.

Like most materials, the material properties of wood generally decrease as temperature is increased and improve as temperature is reduced. This is also true for Celotex™.

Experimental stress/strain curves indicate that balsa and redwood, loaded parallel to the grain, have nearly ideal properties for use as impact absorbers; they have nearly flat response up to high strains. Balsa is capable of strains over 80% before lock-up, and some redwoods have measured strains up to 73% before lock-up. A static load deflection curve for balsa is shown in Fig. 2.5 and Table 2.5 gives the mechanical properties of balsa for a range of material density.

Material properties for balsa and redwood that are useful for uniform crush analysis of those woods loaded parallel to the wood grain are given in several reports[11],[12],[13],[14],and[15]. The effectiveness of wood impacted off of the grain axis has not been fully explored in the dynamic regime.

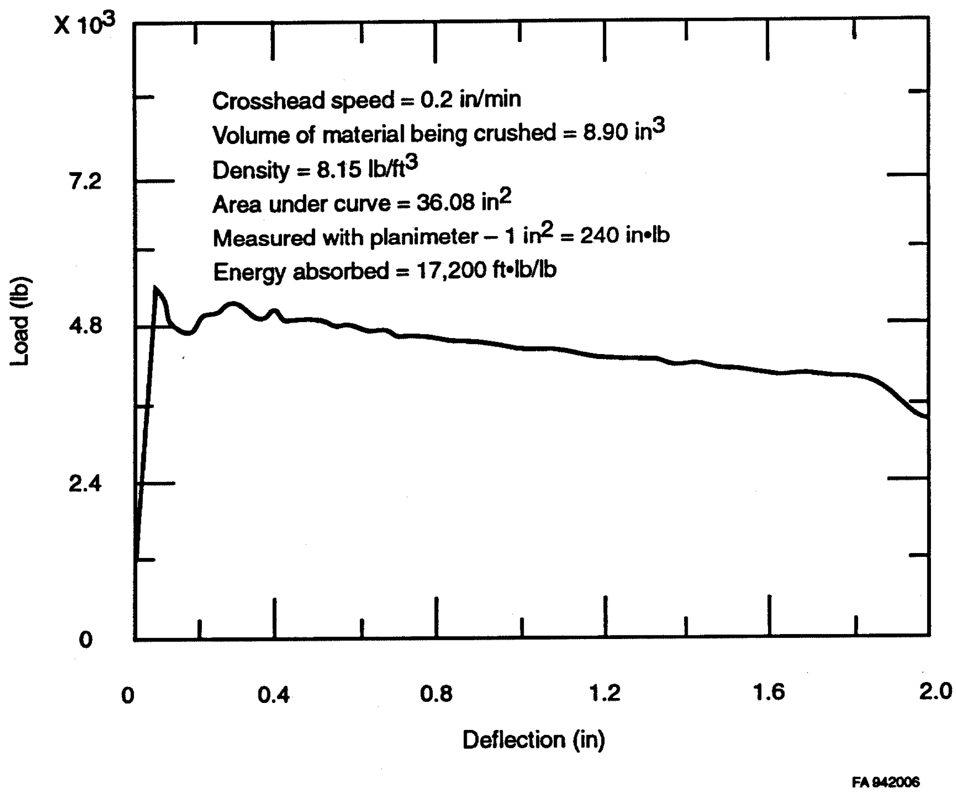


Fig. 2.5. Static load deflection curve for balsa wood.

Table 2.5. Averages of peak and mean crushing strengths for various densities of balsa wood

Average of density range (lb/ft^3)	Average of peakcrushing strength (psi)	Peak strength, mean crushing strength (psi)	Mean strength %	Number of tests
6.5 - 7.4	1689	1232	137.0	9
7.5 - 8.4	2080	1452	143.2	18
8.5 - 9.4	1460 [a]	965 [a]	153.3	3
9.5 - 10.4	2750	1624	169.3	8
10.5 - 11.4	3070	1780	172.5	17
11.5 - 12.4	3055 [a]	1715 [a]	178.2	2

[a] Low values believed due to insufficient testing.

Generally, the techniques for designing foam impact limiters can be used for wood limiters as long as the wood is loaded parallel to the grain. Experiments indicate that wood has low shear strength relative to compressive strength; for end impact, the wood effectiveness is that in the cylindrical volume projected down from the contact surface between the limiter and the containment. For corner drops, the effective cylinder of wood is projected from the containment-limiter interface parallel to the grain of the wood.

Honeycomb impact limiter materials

Honeycomb impact absorbers are constructed by bonding corrugated strips of thin material together to form a relatively low-density solid full of cylindrical voids. Honeycombs constructed of paper, fiber-reinforced plastic, aluminum, or stainless steel all have nearly ideal load deformation curves for impact limiters; that is, flat up to about 75% strain in compression. New material shows a sharp rise of load at impact above the plateau of load. However, this short duration peak load can be completely eliminated by slightly buckling the material, i.e., by specifying precrushing of the honeycomb during the manufacturing process. The cylindrical holes in the material are all parallel in common honeycomb, and, consequently, the material properties are orthotopic. Orthotopic impact absorbers are positioned so that the force is parallel to the holes. The variation of static crush strength of corrugated honeycomb with density can generally be found in vendor data[16].

Manufacturers' literature suggests that strain rate effects increase the crush strength of honeycomb by up to 30% in a dynamic test over the static value. Also, the influence of temperature is included in manufacturers' literature; the temperature dependence of the yield strength of the material from which it is made can be used to characterize the honeycomb.

Honeycomb materials are ideal candidates for the uniform crush analysis design procedure, but the lack of material property data makes finite-element analysis difficult without an experimental program.

Steel impact limiter designs

Since steel is as stiff as most containment materials, an impact limiter of steel must be shaped and positioned to allow the limiter to dissipate energy by distorting plastically. Steel limiters have been designed in the shape of frames made of welded tubing, shells of revolution, and fins. Generally, steel impact limiters are not used for drum-type containers. However, some inner containment designs must absorb energy from Normal and/or Hypothetical Accident Conditions. Some manual calculations can be used to design/analyze steel impact limiters, but in general finite-element computer programs are used to predict behavior.

Because of their relatively high ultimate strain, mild steel and stainless steel are common materials for the construction of impact limiters. Individual sources of a complete set of material data necessary for nonlinear dynamic analysis may be difficult to find. However, Rack and Knorovsky[17] identify sources for temperature and strain rate dependent data for stainless steels used for shipping containers[17] while Nicholas includes strain rate dependence of other steels.[18]

Design of steel impact limiters or containers to resist the regulatory drop requirements are best accomplished with use of nonlinear dynamic impact computer codes like HONDO[19] and DYNA.[20] Actual drop testing is usually necessary to verify the design.

2.4.3.3 Shielding

This section discusses structural issues in the design of the shielding. Chapter 5, Radiation Shielding, is the major source of information on shielding performance issues for DOE containers.

In DOE containers, the structural parts of the containment almost always are sufficient to provide shielding of short-range radiation such as alpha and beta, and usually are sufficient to reduce gamma and neutron radiation to acceptable levels. In cases in which this is not true, shielding against gamma radiation is achieved by using materials such as lead, steel, or even neutron-free depleted uranium. Sometimes materials such as polyethylene or boron carbide are added to provide neutron shielding. In all cases, the design must assure that the mechanical configuration of any shielding materials is maintained. In addition, when lead is used as a shield material, care must be taken that the lead does not slump or cold flow and lose its configuration. This can happen at relatively low temperatures. To prevent decomposition it is important to protect any plastic or other hydrogenous neutron shielding from high temperatures or from intense gamma radiation.

Loads that can result in rupture or severe distortion of the containment vessel also pose a threat to the shielding system. Hypothetical accident stresses and concentrated loads that may cause punctures can reduce the efficiency of the shielding even if the containment retains its ability to prevent excessive radioactive release. Vibration and differential thermal expansion may cause problems if the mechanisms holding the shielding components in place are subject to fatigue.

The design must also be evaluated to assure that the materials selected for the shielding are compatible under all conditions of transport. The materials selected must be such that there will be no significant chemical, galvanic, or other reaction among the package components and contents. The primary concern is thermal degradation or decomposition, both during normal transport and Hypothetical Accident Conditions, particularly when plastic or other low-temperature materials are used. While not normally a problem encountered in DOE packages, radiolysis of hydrocarbons is a concern at higher radiation levels. Decomposition can affect the shielding properties of the material and can also cause pressurization of the containment boundary if gases are generated.

The most significant concern is a case in which decomposition of materials in the shielding results in a loss of mechanical properties or a physical configuration change that may lead to a failure to adequately shield personnel from the contents.

2.4.3.4 Packaging internals

The first step in the design of packaging internals is to decide the level of protection required to safeguard the containment vessel and the package content from mechanical or thermal damage during all conditions of transport and to develop an initial conceptual design. These conceptual designs must be evaluated to assure that the materials selected for the package internals are compatible under all conditions of transport. The materials selected must be such that there will be no significant chemical, galvanic, or other reaction among the package components and contents. The primary concern is thermal decomposition, both during normal transport and Hypothetical Accident Conditions, particularly when plastic foam or other low temperature materials are used. While not normally a problem encountered in DOE packages, radiolysis of hydrocarbons is a concern at higher radiation levels. Decomposition can affect the mechanical and thermal properties of the material and can also cause pressurization of the containment boundary if gases are generated.

The most significant concern is a case in which decomposition of materials in the package internals results in a loss of mechanical properties or a physical configuration change that may lead to a criticality or to a failure to protect the integrity of the containment boundary. This affects all structures that maintain safe geometry inside the container as well as neutron absorber materials such as polyethylene or boron carbide.

Preliminary design of shielding that is inside the containment boundary should be done using these guidelines and those of Subsect. 2.4.3.3.

In evaluating the mechanical requirements for the package internals, the primary loads to be considered are inertial loads due to Normal and Hypothetical Accident Conditions, vibration, and differential thermal expansion. The load paths from the package contents through and including the package internals to the containment boundary must be very carefully evaluated.

Since the package internals, by definition, are within the pressure boundary, pressures external to the containment have little if any effect. However, if a closed cell plastic foam or a similar material is used to support the package contents or to protect the containment boundary, its mechanical performance might be adversely effected by high pressures generated by temperature increases or material decomposition inside the containment boundary.

If the contents of the package generate a significant amount of heat, additional difficulties arise. It may be necessary to devise a way to carry heat away from the contents in order to prevent high-temperature damage to the containment, the package internals, or even to the content itself. The design problem arises from the necessity to prevent external heat sources during normal transport or Hypothetical Accident Conditions from using the same means of heat transfer to carry heat in the opposite direction.

2.4.3.5 Tie-down and lifting devices

The tie-down system is the arrangement of tie-down hardware, such as shackles and wire rope, that secures the package onto the vehicle. Design concepts for tie-down and lifting devices concepts should include operational necessities and conveniences as well as mechanical strength considerations.

In evaluating the mechanical requirements for the tie-down or lifting devices, the primary loads are defined by the weight and configuration of the package and loadings defined in 10 CFR 71.45. The load paths to the package must be very carefully evaluated. Because of the requirement that failure of the tie-downs or lifting devices must not adversely effect the performance of the package, these features must essentially be designed to fail under the specified loads and in specified ways.

A tie-down manual for type B containers has been developed by Smallwood.[10] This manual tested a DOE type B container to the requirements of 10 CFR 71.45. Although the outer container yielded at the maximum load, the inner container was not affected.

2.4.4 Detailed Design

After a design concept has been selected, the detailed design process can begin. This process includes final selection of materials, detailed calculations, and preparation of final design specifications and drawings in preparation for prototype testing and other certification activities. The following sections include guidance on factors involved in the detailed design process.

2.4.4.1 Containment system

This chapter discusses structural design of the containment system. Chapter 4, Containment, is the major source of information on containment. The containment boundary in a DOE container is usually an austenitic stainless steel containment vessel with a bolted closure to accommodate the loading and unloading of contents. The closure contains a seal or seals to minimize leakage from the containment vessel to the environment. Penetrations of the containment may be needed for operating purposes; for example, backfilling with a tracer gas for leak testing or helium for heat-transfer enhancement. The closures of penetrations, such as valves or sealed tubes are also considered part of the containment system.

The containment vessel is usually a welded cylindrical vessel with a flange at one end to allow for loading and unloading the content. The flange design is largely dictated by the design of the closure, which will be discussed below. It is important to note that in DOE containers, which are normally relatively lightweight and which use soft impact limiters, top flanges may be somewhat exposed to impact damage during hypothetical accidents, particularly those that impact at the top corner. Since the impact limiters are also often used as thermal insulation, there is also a possibility that additional thermal loads will be encountered after such an accident. To afford more protection the closure flange is often moved to the center or at least some distance away from the corner.

To simplify construction of the containment vessel and eliminate potential structural and leakage problems at welds, containment boundary designers should consider the use of single-piece forged or drawn vessels.

Usually, DOE packages will not require pressure-relief devices; however, the containment must be designed to handle any pressure excursions resulting from not having a relief system during either normal transport or accident conditions. Containment must be maintained if there is a pressure or temperature increase due to chemical reactions resulting from water inleakage, irradiation, or thermal effects on the package or content during either normal transport or accident conditions. Since the containment cannot be continuously vented, it must be designed to retain an internal pressure, even when intermittent pressure relief is provided.

The major loads are heat, internal and external pressure, vibration, and impact. Even though the regulations do not impose any specific requirements on any structural components in terms of stress allowables or deformation limits, the containment boundary will be compromised if the structural components are overstressed or grossly distorted. Therefore, the structural components should be designed according to a well-established design standard such as the ASME Code, as recommended in Regulatory Guide 7.6. Other codes and standards can be used as design criteria provided they are as conservative as the ASME Code. All the loadings from the normal and accident conditions should be considered and combined, as recommended in Regulatory Guide 7.8.

As set out in Sect. 2.2 and in Regulatory Guide 7.11, the NRC has adopted the philosophy of applying stricter requirements and higher margins of safety to Type B packages with higher levels of radioactivity.[4] Category I containers are used with contents of the highest level of radioactivity, Category II containers ship contents with moderate levels of radioactivity, and Category III containers ship contents with still lower radioactivity. The three categories and the associated radiation levels are shown in Fig. 2.2.

Section 2.3 contains an explanation of Structural Design Criteria for containers. These criteria should be used for the design of all safety related structures in the package internals.

Regulatory Guide 7.6[5] describes the design criteria for the structural analysis of shipping cask containment vessels and explains the use of the design-by-analysis approach for Class I components from Section III of the ASME Code[6] as design criteria for the containment vessels. Regulatory Guide 7.8 elaborates on the normal and accident tests conditions specified in 10 CFR 71 and recommends the loading combinations for the structural analysis of shipping casks.[7]

The design of the closure system for a containment boundary is of utmost importance. Most containment systems for DOE containers have bolted lids with elastomer O-rings. The design of the lid, in addition to satisfying the structural code used in the design, should consider the load paths of external loads, particularly in accident conditions. To minimize shear loading on bolts, designs that include a protected closure are encouraged. Protected means that no transverse force components from the impact limiter can be delivered to the closure lid during any hypothetical accident. The lid is recessed into a counterbore that protects the edges of the lid. This is not a regulatory requirement, but is a good design practice and is encouraged. In the past, many containers have been certified that do not use this feature; if a protected closure is not used on a design, the bolting system and seals must be designed to accommodate the large transverse loads.

Some containment systems actually use a double lid arrangement. This system is not normally used on drum-type containers, but it does offer some advantages when the closure is not adequately protected from either impact loads or thermal loads. By using two lids, the inner lid has additional protection. This design may also provide an additional facility for leak testing. Although not widely used in DOE containers at this time, this system may be used effectively when the additional expense and complication are justified.

As previously mentioned, the seal design of containment systems for DOE containers normally consists of elastomer O-rings or gaskets. These are usually installed in a face seal arrangement. In this design the integrity of the seal depends on the maintenance of preload on the closure bolts to energize the seals. This means that the bolting systems must be strong enough to withstand all of the externally applied loads without loss of preload. The ASME code defines the allowable stresses for bolts and should be applied as advised in Regulatory Guide 7.6[5], Regulatory Guide7.8[7] and Regulatory Guide 7.1 l[4] depending on the radioactive content of the package. The design of the bolting should account for all pressure loads, external forces and vibration loads, and thermal expansion and contraction.

The bolting system must perform at normal transport temperatures from -40^{0}C (-40^{0}F) to Hypothetical Accident Condition temperatures of up to several hundred degrees. Because of hypothetical accident impact loads which can occur at very low temperatures, brittle fracture of the bolts is a major concern. Bolting materials should be selected with this in mind. Another bolt material selection criterion is the possibility of galling the bolt threads. This can be a real concern if the bolts are used in threaded holes in a stainless steel containment vessel, instead of with nuts. The galling problem may be avoided by proper selection of materials for the bolts and vessel or by using thread inserts.

Seal design has been mentioned previously, but additional attention is warranted. The elastomer O-rings or gaskets on DOE containers are usually installed in a face seal arrangement. Other materials and arrangements are used. In particular, metal seals may be used where high temperatures cannot be avoided or where seal reuse is not required. Some containment boundaries are actually welded closed. These “metal only” seal solutions offer additional security in some situations, particularly when the container may be used for long-term storage. Face seals have been widely used, but radial and other self energizing arrangements which might be considered have the advantage of not relying on bolt preloads to maintain the seal.

Normally, two seals are used. There is often a penetration into the annulus between the two seals. In this arrangement, the outer seal is normally used only as a means to leak-check the inner seal. The inner seal is a single seal, since if it leaks the content may be released into the annulus, which has a penetration into it. This annulus is normally plugged, but the plug is not normally leak-checked. If the plug is leak-checked after installation, a double seal may be claimed. Note that this is not the same as double containment, since the vessel may leak in places other than the seals.

2.4.4.2 Impact limiters/thermal insulation

The type of impact limiter chosen dictates the range of possible analysis methods and the complexity of analysis. When the shipping package can be separated into a containment part and an impact limiter part and the two parts are structurally dissimilar; for example, the impact material is less stiff and less dense than the containment, many analytical techniques are valid, and this variety can be exploited to control the cost of analysis. In the case of an impact limiter that is structurally similar to the containment, a designer may be limited to modeling the combined structure or experimentally determining response.

Complexity of analysis also varies with the spacial dimensions necessary to describe the geometry of the impact limiter as it undergoes deformation. The simplest deformation state is uniform crush over a volume, which corresponds to a zero-dimension state. A one-dimensional solution occurs when the cross-sectional area of the impact limiter varies in a predictable way as it crushes. Two-dimensional solutions are necessary for cases in which a two-dimensional stress state is important, such as when a shear stress limits the impact limiter material involved in crushing. A material with a nontrivial shear strength, such as steel, impacting in a skew orientation requires a full three-dimensional analysis.

A drum-type container has impact limiter/cushioning material inside the drum and inside the inner containment and thus, the material is confined or restrained by the containment or outer shell. Therefore, one-dimensional analysis of the limiter material may not be accurate, but may be used for initial or preliminary designs.

Uniform crush analysis

Some impact limiter materials can be accurately analyzed by assuming uniform crush of the material. In this analysis, the impact limiter materials are assumed to be ideally massless and insensitive to strain rate. Materials such as honeycomb, balsa, Celotex™, and woods exhibit these characteristics. In these materials, an adjustment for inertial forces and dynamic behavior can be incorporated and a nominally static analysis can be performed. For this type of analysis, the stress in the limiter material must be uniform or representable as uniform at every instant of time. This may hold for a drum-type container impacting flat on its end or side. Materials with very low shear strength relative to compressive strength, such as balsa and some rigid foams, crush as though only the volume of material under the area of contact is effective. The impact limiter should be designed to ensure that the deformation of the material is not large enough to cause the material to "lock up" before sufficient kinetic energy has been absorbed. See Subsect. 2.4.3.2 for more information on "lock up" of foam and wood impact limiter materials.

A typical analysis for the case of uniform crush is presented below for an impact limiter material with the strictly increasing stress/strain curve shown in Fig. 2.6. This shape curve is typical for limiters and cushioning material used in drum-type containers (e.g., wood, Celotex™, foam).

At the beginning of a drop the kinetic energy (KE) of the container is

$$KE = 1/2 \; W/g \; V^2 ,$$

where: W = container weight, lb,

g = acceleration of gravity, 980.6 cm/s^2 (386.4 in./s^2),

V = container velocity, in./s.

The center of mass of the container moves downward by a distance h (in inches). The work done by gravity is,

$$Wg = Wh .$$

With a constant area, A (in square inches), the force developed in the impact limiter retarding the container is

$$F = \sigma(\in)A, \text{ lb} ,$$

with stress σ, expressed as a function of strain $\in$, assuming that strain rate is not important. The strain can also be written,

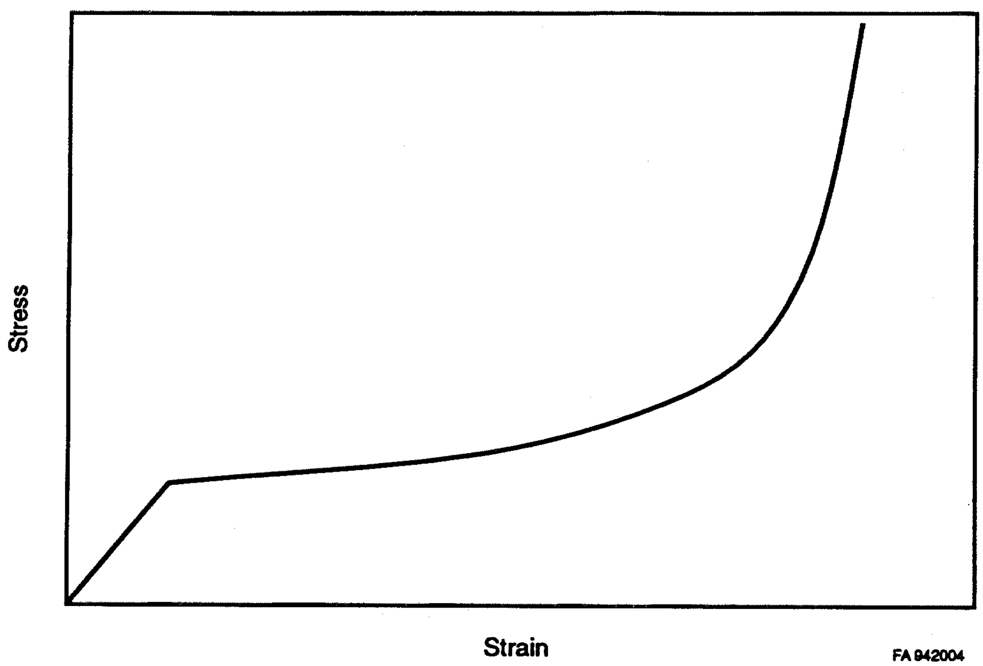


Figure 2.6. Uniform crush stress/strain curve.

$$\epsilon = h/L ,$$

where L is the original length of the impact limiter.

During this time the work done is crushing the impact limiter is,

$$W_c = - \int F(h)dh ,$$

which can be written as

$$W_c = - AL \int\sigma(\epsilon)d\epsilon.$$

The kinetic energy then is written,

$$1/2\ W/g\ V_2^2 = 1/2\ W/g\ V_1^2 + Wh - AL \int\sigma(\epsilon)d\epsilon.$$

If $\sigma(\epsilon)$ is analytically integrable, the equation can be solved for the crush hour to stop the container with $V_2 = 0$. Some readily integrable stress/strain functions to deal with are straight line segments, polynomials, or combinations of the two.

Linear deformation analysis

A variation in cross-sectional area of the impact limiter along the direction of crush produces a condition of non-uniform stress as well as non-uniform strain that can be analyzed by breaking the limiter into elements of constant area. The number of elements used depends on the rate of change of area.

Many divisions are needed where area changes rapidly, but only one element is needed for a length of constant area.

For this type analysis the axial force is constant through the impact limiter, but the stress varies. Working with a strain-rate-independent material, the strain can be read from the stress/strain curve for each element of material. Manual calculation of total crush of the limiter and internal energy dissipated becomes difficult because of the large number of elements and internals to be evaluated. However, computer programs can be developed to carry out all the needed calculations.

This calculation method is valid for the analysis of corner impacts of a container provided that the material is essentially isotropic and has a low shear strength relative to compressive strength. A discussion of modeling a cylindrical impact limiter during a side and corner drop is presented in Hill and Joseph.[11]

Two-dimensional analysis

Complications in the design can invalidate one-dimensional solutions; however, two-dimensional, dynamic, structural analysis codes are available to provide solutions. Typical computer codes suitable for dynamic analysis of a nonlinear material with a thinner cladding (like a thin walled outer container) over foams, woods, or honeycomb impact limiters are ABAQUS/EXPLICIT,[21] LSDYNA,[20] ANSYS,[22] HONDO,[19] and PRONTO.[23] Some of these codes contain a material subroutine to model crushable foam behavior as well as elastic-plastic behavior. These codes are representative of codes that are capable of analyzing ductile material behavior.

Two-dimensional treatment of shipping container impact is appropriate for the end drop, which is axisymmetric. However, side drops and corner drops require three-dimensional analysis.

The material properties required for each code or material subroutine differ, but in general they require an elastic modulus, a yield stress, a plastic modulus, Poisson's ratio, a hardening parameter, and density. The special foam model uses bulk pressure versus dilatation, bulk unloading modulus, three yield function constants, elastic shear modulus, and the pressure cutoff for tensile fracture. Although the bulk data are rare in the literature, bulk properties can be derived from uniaxial test results.

Three-dimensional analysis

Most of the computer codes listed in the previous section on two-dimensional analysis can model three-dimensional behavior is stress and strain. The elastic-plastic material subroutines in these codes are identical for three- or two-dimensional analysis and the same material properties are required. A corner drop is one condition that requires a three-dimensional analysis code for solution.

2.4.4.3 Shielding

This section discusses structural design of the shielding. Chapter 5, Radiation Shielding, is the major source of information on shielding performance issues for DOE containers.

As set out in Sect. 2.2 and in Regulatory Guide 7.11[4], NRC has recommended that the structural design of all shielding structures be in accordance with criteria contained in the ASME Boiler and Pressure Vessel Code, Section VIII Division I or in Section III, Subsection NF.

Section 2.3 contains a detailed explanation of Structural Design Criteria for containers. These criteria should be used for the design of all safety related structures in the package.

Regulatory Guide 7.6[5] describes the design criteria for the structural analysis of shipping cask containment vessels and explains use of the design-by-analysis approach for Class I components from Section III of the ASME Code[6] as a design criteria for the containment vessels. Regulatory Guide 7.8[7] elaborates on the normal and accident tests conditions specified in 10 CFR 71 and recommends the loading combinations for the structural analysis of shipping casks. Both guides may be applied to the structural design of package shielding.

Very careful analysis using the previously discussed guidelines is required to assure that geometry is maintained under all loading conditions. These shielding structural components should be designed according to a well-established design standard such as the ASME Code previously described. Other codes and standards can be used as design criteria provided that they are as conservative as the ASME Code.

Plastic-based shielding materials are often used in containers. Unfortunately, these materials do not lend themselves very well to analysis using the ASME code or other design codes based on linear elastic behavior. In that case, other analytical methods must be used, and/or thorough testing must be done.

All of shielding materials should be examined for compatibility and stability at temperature and in radiation fields.

2.4.4.4 Package internals

As set out in Sect. 2.2 and in Regulatory Guide 7.11 ,[4] the NRC has adopted the philosophy of applying stricter requirements and higher margins of safety to Type B packages with higher levels of radioactivity. Category I containers are used with contents of the highest level of radioactivity, Category II containers are used with contents of moderate levels of radioactivity, and Category III containers are used with contents of still lower radioactivity. The three categories and the associated radiation levels are shown on Fig. 2.2.

The same basic philosophy applies to the design of package internal structures. The three component safety groups and the associated design criteria defined in Regulatory Guide 7.11[4] can also be used for package internal structures. Table 2.3 contains a summary of these design criteria. Package internals structures that perform functions important to maintaining criticality safety, contamment, and shielding are designed with stricter requirements and higher margins of safety. Section 2.3 contains a detailed explanation of Structural Design Criteria for containers. These criteria should be used for the design of all safety-related structures in the package internals.

Regulatory Guide 7.6[5] describes the design criteria for the structural analysis of shipping cask containment vessels and explains use of the design-by-analysis approach for Class I components from Section III of the ASME Code[6] as a design criterion for the containment vessels. Regulatory Guide 7.8[7] elaborates on the normal and accident test conditions specified in 10 CFR 71 and recommends the loading combinations for the structural analysis of shipping casks. Both of these guides may be applied to the design of package internals.

In most DOE containers, the main package internals design concern is protection of the containment boundary from the impact of the package internals and contents in a hypothetical accident. However, package internal design critically safe geometry is of major importance in many containers, and shielding integrity may also be vital.

In summary, very careful analysis using the guidelines described is required to assure that geometry is maintained under all loading conditions. These package internal structural components should be designed according to a well-established design standard, such as the ASME Code described previously. Other codes and standards can be used as design criteria provided that they are as conservative as the ASME Code.

Support of the contents and maintenance of the package geometry can be accomplished in many ways. Often the support is as simple as a conformal resilient elastomer foam insert for the containment boundary. This support insert can also be made of a solid elastomer, if needed for strength or cut resistance. Often polyurethane is used for this application because of its high abrasion resistance and good properties at low and high temperature, but other elastomer materials, such as silicone, are used where temperature extremes are expected. Elastomers insulate the content from the containment boundary, so if the content contains heat-generating materials that require good heat-transfer outward, metal honeycomb material is often used. Unfortunately, these support mechanisms do not lend themselves very well to analysis using the ASME Code or other design codes based on linear elastic behavior. In that case, another analytical methods must be used and/or thorough testing must be done.

Other materials are used to support the contents and perform other functions within the package internals. Metal cans, covers, and spacers are frequently used. In addition, the contents are often bagged in plastic for contamination control, and desiccants are often included with the content. All of the materials should be examined for compatibility and stability at temperature and in radiation fields.

2.4.4.5 Tie-down and lifting devices

The designer should be aware that the forces applied to individual tie-down devices depend very much on the overall tie-down system and its design philosophy. The analysis of the overall tie-down system can be quite complex when the elastic/plastic deformations of cables and straps of a flexible system are considered and the dynamic loadings are included. Depending on the design philosophy, the tie-down system may be designed to fail at certain load levels, thus altering the applied loads to the tie-down devices. A tie-down manual for DOE type B containers has been developed by Smallwood.[10]

Lifting attachments must have a safety factor of three over the yield strength when used to lift the package in the normal configuration. (The yield value used should be the worse case for the material in the Normal Conditions of Transport temperature range. This range should include that temperature increase caused by any internal heat generation from the contents and also the increase from solar insolation if the package is not limited to covered shipment.)

If the lifting attachment fails, it must not impair the ability of the package to meet the other requirements of 10 CFR 71.

Any other structural part of the package which could be used to lift the package must be rendered inoperable during shipment, or meet the requirements described previously.

Tie-down devices which are a structural part of the package must not yield when the package is subjected to a static force with a vertical component of twice the package weight, a horizontal component in the direction of travel of ten times the package weight, and a horizontal component in the transverse direction of five times the package weight. (The yield value used should be the worse case for the material in the Normal Conditions of Transport temperature range. This range should include that temperature increase caused by any internal heat generation from the contents and also the increase from solar insolation if the package is not limited to covered shipment.)

If the tie-down device fails, it must not impair the ability of the package to meet the other requirements of 10 CFR 71.

Any other structural part of the package which could be used to tie down the package must be rendered inoperable during shipment or meet the requirements described previously.

There are many tie-down and lifting device designs, but the most common are the flat steel lugs welded onto the package with a hole for the tie-down hardware attachment. Also common is a commercial hoisting ring or eye bolt inserted into a threaded hole in the package.

Typically, the lifting lug should be at least as thick as the outer shell of the package to which it is welded. The edge distance above the hole should be at least one diameter of the hole. Similarly, the diameter of the hole should not be greater than half of the width of the plate, and the length of the welds should be at least equal to the portion on the lug that is not welded to the outer shell. The bearing stress around the hole should be evaluated. If the lifting lug is stiff, the effects on any external closure bolts resulting from impact loading from free drop conditions should be evaluated. Considerations of a free-drop condition that impact the lifting lug should be given as well as those for the free drop onto the puncture bar.

After the applied forces are determined, the stress analysis methods can be used to evaluate the adequacy of the lifting or tie-down devices. If the lifting or tie-down system is simple, analytical techniques are usually adequate. Critical areas of each design must be recognized and evaluated for the worse possible loads. For example, if the tie-down device is a plate with a hole, the critical areas that should be analyzed are shear stresses on the cross section from the hole to the edge of the plate, bearing stresses on the hole, and bending stresses on the plate due to the tie-down forces. Weldments or bolts should be evaluated according to appropriate welding criteria and bolt pre-load requirements. If the tie-down system uses complex straps or is designed to fail at a particular load, a more detailed analysis, such as finite-element analysis, is warranted.

The design of the lifting or tie-down devices is usually straight-forward. Approximate methods using principles of strength of materials and simple beam theory can be used for stress analyses. Given the approximate nature of the analyses and serious consequences that may occur from the failure of a lifting device, a conservative design approach is necessary. For good engineering practice, a dual load path or a higher safety factor for heavy packages is recommended.

A tie-down or a lifting device that would function in either role, must be analyzed for both cases. The procedure for lifting devices is to perform stress analyses to a factor of safety of three and then to do an actual proof test on each lifting device to 1.5 of the rated load before that device is used for lifting. The actual proof test, while not required by 10 CFR 71, is good practice.

The drum-type containers (Fig. 2.7), called DT containers, contain no lifting device or tie-down attachments and no features making them usable as lifting devices or tie-downs. Lifting is done by normal drum handling methods with fork-truck drum devices or placement of the packages on a pallet, and tie-down is accomplished using a separate tie-down system. As noted earlier, Smallwood[10] provides some guidance for the design of tie-down and lifting devices for DT containers.

2.4.5 Testing and Validation Methods

Section 2.5, which addresses structural validation methods, contains detailed information.

2.4.5.1 Containment system

The primary mechanical loads on the containment boundary are acceleration and vibration loads that are not attenuated by the impact limiters and other package components. Mechanical loads from the package internals and contents are important and should be carefully evaluated. Thermal loads encountered in all phases of transport and from heat generation of the content constitute a second set ofloads. The first step in the analysis of the effect of these loads is to characterize the loading paths, magnitudes, directions, and durations. Thus, to analytically determine the response of the containment boundary, a model of the entire package must be considered.

Regulatory Guide 7.6[5] describes the design criteria for the structural analysis of shipping cask containment vessels. Regulatory Guide7.8[7] recommends the loading combinations to be used in the structural analysis of shipping casks. Regulatory Guide 7.11[4], explains the application of stricter requirements and higher margins of safety to Type B packages with higher levels of radioactivity. The ASME code defines the allowable stresses for the containment boundary and should be applied as advised

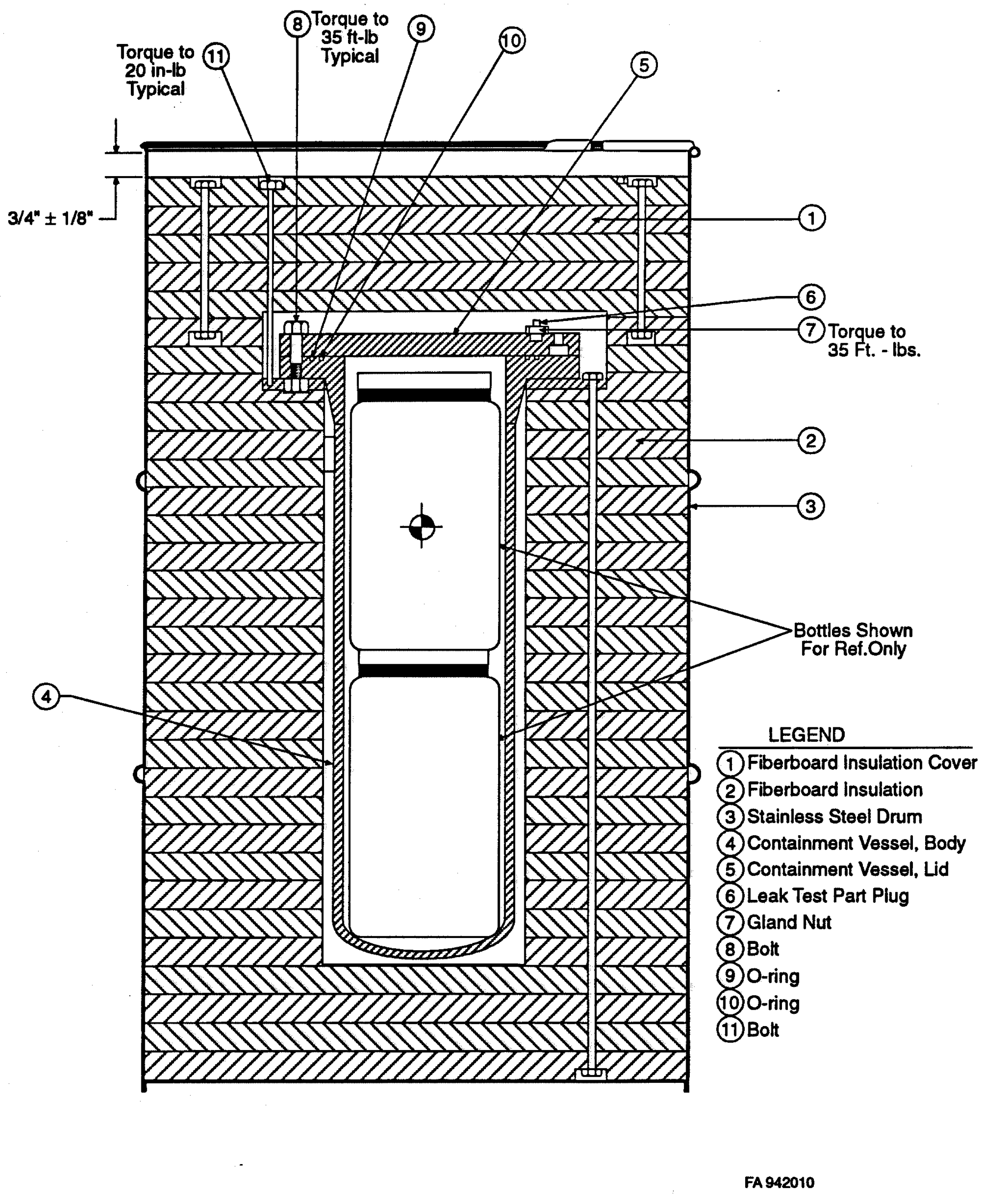


Fig. 2.7. Typical drum-type container.

in Regulatory Guide 7.6,[5] Regulatory Guide 7.8,[7] and Regulatory Guide 7.11[4] depending on the radioactive content of the package. More advanced finite-element or finite-difference methods may be used effectively for both structural and thermal analysis. Any stress analysis should include thermal stresses. Validation methods are discussed in detail in Sect. 2.5.

Testing of prototypes is also used to evaluate the containment design. The testing must be be carefully performed and planned to assure that worst-case loadings as defined in 10 CFR 71, are achieved for both mechanical and thermal loads. Again, the validation methods discussed in Sect. *2.5* can be used to evaluate containment design.

2.4.5.2 Impact limiters/thermal insulation

The primary mechanical loads on the impact limiters/thermal insulation are impact and vibration loads due to normal transport and Hypothetical Accident Conditions. Thermal loads encountered in Hypothetical Accident Conditions and from heat generation of the content constitute a second set of loads. The first step in the analysis of the effect of these loads is to characterize the loading paths, magnitudes, directions, and durations. Thus, to analytically determine the response of the impact limiters/thermal insulation, a model of the entire package must be considered.

The method of analysis used for the impact limiters/thermal insulation depends on their structural configurations. Most of the materials commonly used are not amenable to the linear elastic analysis methods. More advanced finite-element or finite-difference methods may be used effectively for both structural and thermal analysis. Any analysis must include thermal effects of both normal transport and hypothetical accident. This requires that the structural analyst work very closely with the thermal analyst. The thermal behavior of materials normally used in impact limiters/thermal insulation in hypothetical accidents is usually not well characterized. If the materials in the impact limiter/thermal insulation are required for shielding or subcriticality, extra precaution is obviously warranted in the validation. All the validation methods discussed in Sect. 2.5 of this guide can be used for the package internals.

Testing of prototypes is used extensively to evaluate the impact limiter/thermal insulation design. The testing must be carefully performed and planned to assure that worst-case loadings, as defined in 10 CFR 71, are achieved for both mechanical and thermal loads. The validation methods discussed in Sect. 2.5 can be used to evaluate containment design.

2.4.5.3 Shielding

Since the containment shielding is usually not designed for structural considerations alone, an examination is needed of how the structural response of the containment vessel and closures affect the shielding components. This examination is normally conducted during the analysis of the containment and package internals.

Additional detailed information on the analysis of shielding performance is in Chap. 5, Radiation Shielding. The results of any analysis or testing showing a potential degradation in the performance must be factored in to all shielding calculations.

2.4.5.4 Package internals

The primary mechanical loads on package internal components are acceleration and vibration loads that are not attenuated by the impact limiters and other package components. Thermal loads encountered in all phases of transport and from heat generation of the content constitute a second set of loads. The first step in the analysis of the effect of these loads is to characterize their magnitudes, directions, and durations. Thus, to analytically determine the response of the package internals, a model of the entire package must be considered.

The method of analysis used for the packaging internals depends on their structural configurations. For example, many materials commonly used are not amenable to the linear elastic analysis methods of the ASME Code. More advanced finite-element or finite-difference methods may be used effectively for both structural and thermal analysis. Any stress analysis should include thermal stresses. All the validation methods discussed in Sect. 2.5 can be used for the package internals.

Testing of prototypes can be also be used to evaluate the package internal design. The testing must be carefully performed and planned to assure that worst-case loadings, as defined in 10 CFR 71, are achieved for both mechanical and thermal loads. Again the validation methods discussed in Sect. 2.5 can be used for the package internals.

Regulatory Guide 7.8[7] elaborates on the normal and accident tests conditions specified in 10 CFR 71 and recommends the loading combinations for the structural analysis of shipping casks.

2.4.5.5 Tie-down and lifting devices

The primary mechanical loads on lifting and tie-down devices which are a structural part of the package are defined by the dynamic loading requirements of 10 CFR 71.45. The first step in the analysis of the effect of these loads is to characterize their magnitudes and directions. Thus, to analytically determine the loads, a model of the entire package must be considered to determine the center gravity and other mass properties of the loaded package.

The method of analysis used for the lifting and tie-down devices depends on their structural configurations. More advanced finite-element or finite-difference methods may be used effectively for both structural and thermal analysis; however, the analysis of lifting and tie-down devices is usually straightforward. Many validation and test methods discussed in Sect. 2.5 can be used for lifting and tie-down devices.

Testing of prototypes can be also be used to evaluate the lifting and tie-down device design. The testing must be carefully performed and planned to assure that worst-case loadings, as defined in 10 CFR 71, are achieved. Validation methods discussed in Sect. 2.5 can be used.

2.4.6 Quality Assurance

Quality assurance for all packaging activities must conform with the requirements of 10 CFR 71, Subpart H. Quality assurance is applicable to all aspects of the structural design as well as the actual container hardware. This chapter discusses quality assurance for the packaging structural design and components. Chapter 9, Quality Assurance, is the major source of information on quality assurance.

Quality assurance begins with the design. The design calculations should be verified independently. If any computer analytical software is used, it should be baselined against known problems similar to those being solved. Analytical techniques used should be evaluated against test data on similar problems. Material properties should be verified.

During the test phase of the design, records of the configuration, before and after testing, should be maintained. All prototypes should be built using certified materials, dimensions, and fabrication methods specified in the design documents and used in the calculations.

All production containers should be fabricated and maintained using certified materials, dimensions, and fabrication methods similar to those specified in the design documents and used in the calculations.

A record-keeping system must be established and records of the design, the prototype container, the certification effort, and the production containers must be maintained.

2.4.7 Structural Design Examples

Many unique drum-type containers have been designed and are in use. In this section, to show the variety of existing materials, design concepts, and payloads, three different example designs are reviewed. The structural designer is encouraged to draw from past experience and lessons learned, but also to be innovative in selection of materials and design concepts for new applications. The three examples feature a typical DT container primarily for shipments of uranium parts/products, a container for shipment of tritium, and a container primarily for shipment of plutonium metals/powders.

2.4.7.1 DT container

A typical DT container is shown in Fig. 2.7. Features of this design are as follows:

- Austenitic stainless steel inner containment boundary
- Bolted cover, two O-rings (EPDM)
- Top-flange container
- Celotex^{TM} for impact limiter and thermal insulation
- Stainless steel outer drum with vent holes covered by plastic tape (inside of drum)

- Standard locking ring lid design

A lesson learned in the DT program is that when using large, heavy (> 500 lb) drums it is best to use a bolted lid design.

2.4.7.2 H1616-1 container

The H1616-1 container is shown in Fig. 2.8. This container is used primarily for shipment of tritium products and has several unique materials and design features.

- Double containment with the HTV a welded containment
- Type 304 stainless steel secondary containment vessel
- Aluminum tubing pellets for packing (cushioning) material around the HTV
- Foam pads top and bottom of containment vessel
- Fire-retardant polyurethane foam for cushioning material in the 16-gal stainless steel drum
- Ceramic fiber material for the thermal barrier
- Locking ring for drum lid

2.4.7.3 SAFEKEG 2863B container

A section of the SAFEKEG 2863B container is shown in Fig. 2.9. This container was designed to fully meet all current and proposed U.S. and IAEA regulations. Another objective was to have a large margin of safety to withstand the Hypothetical Accident Conditions.

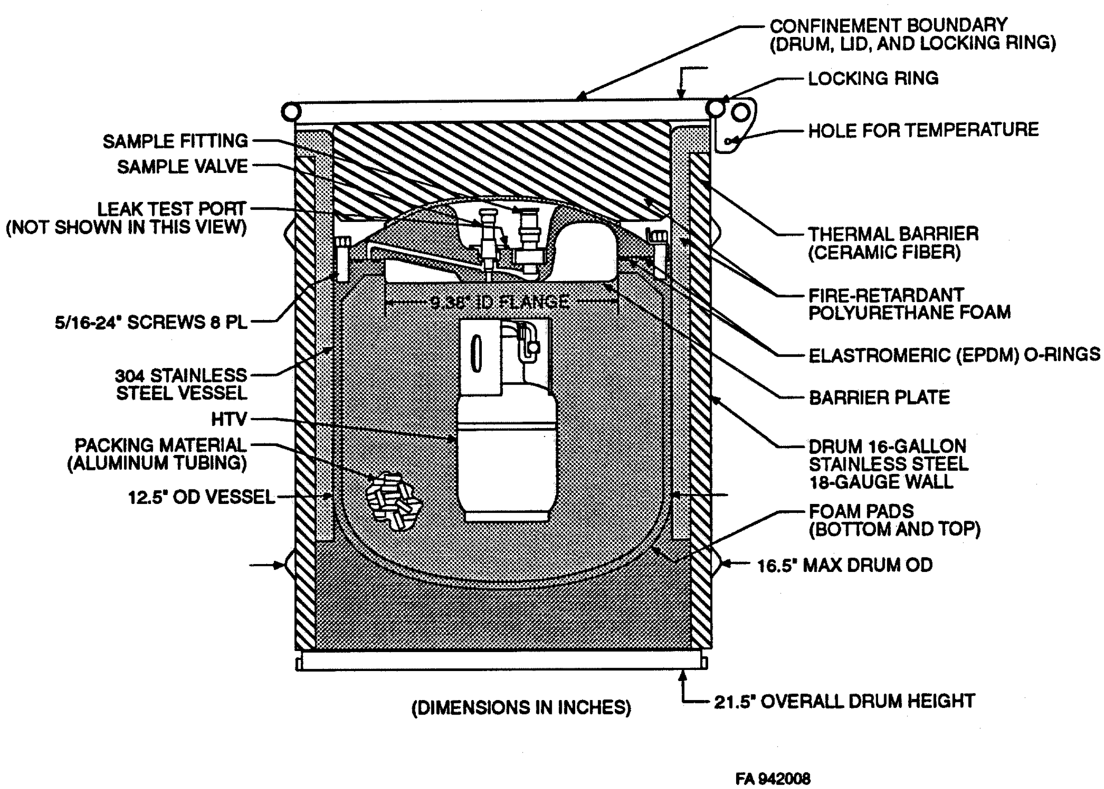
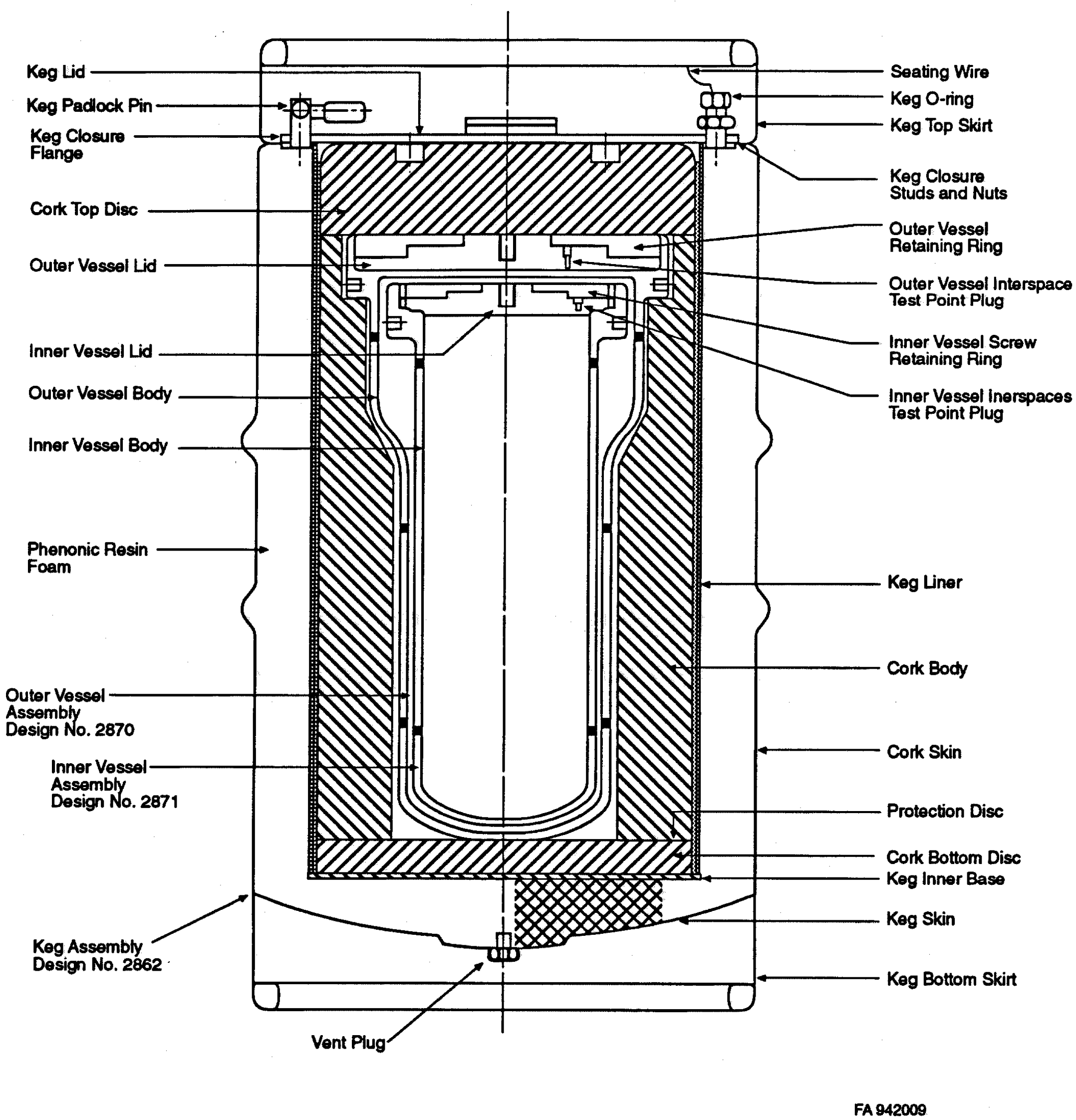


Fig. 2.8. H1626-1 shipping container.



Furure 2.9. SAFKEG 2863B shipping container.

The packaging consists of an outer double skin insulated keg, an insulating cork liner, an outer containment vessel, and an inner containment vessel. The keg is a double-skinned, stainless steel body with the cavity filled with an insulating phenolic resin foam. A flat stainless steel lid bolts to the keg body. The cork liner is made from a reconstituted cork material, and its surface is sealed to enhance its appearance and to resist wear and tear. The outer containment is fabricated from stainless steel. The seal between the body and the lid is effected by two O-ring face seals. The O-ring material is a fluorocarbon elastomer. The lid is held in position by a threaded retaining ring. Both the retaining ring and the lid are set into the body of the container, thus reducing the vulnerability of the closure. The vessel operates at atmospheric pressure. The design, materials, and construction of the inner containment vessel are similar to those of the outer containment vessel.

2.5 VALIDATION METHODS

Structural validation can be accomplished by employing structural analysis techniques or physical tests. In most cases, it is advantageous to combine testing and analysis to validate the design.

2.5.1 Analysis Methods

Analysis methods can be grouped into analytical techniques and numerical methods. In general, analytical techniques are calculations with formulas from engineering reference books. These formulas apply to simple structures with well-defined boundary conditions and loadings and are obtained from the principles of elasticity and strength of materials. When the structures are complex, the boundary conditions change with the applied loads (e.g., contact surfaces), or the load paths are complicated, numerical methods should be employed.

For drum-type packages, the primary load conditions are internal or external pressures on the containment and impact loads acting through the outer shell resulting from the regulatory requirements. Pressure loadings are typically static, uniform loading. The effects on structures are well investigated. In general, the force of pressure on a structure is the product of the pressure and the projected area over which the pressure acts. Formulas for stress and strain[24] and the ASME Code[6] provide many analytical formulas for the stresses and displacements, caused by pressure loading, in plates, shells, and beams. Impact loads, on the other hand, are dynamic and produce nonlinear effects which are not easily determined.

2.5.1.1 Analytical techniques

Most structural formulas are derived by considering a portion of a loaded member as a body in equilibrium under the action of forces that include the resulting reactions. The equations of equilibrium are then used to solve for the reactions and then for the stresses in the loaded member.

Analytical formulas are based on certain assumptions such as the properties of materials, the regularity of the geometry, and approximate boundary conditions. Formulas are derived by mathematical procedures which often involve further approximations and simplification. Consequently, the calculated values of stress, strength, and deformation cannot be considered to be exact and great numerical precision cannot be obtained.

Care must be exercised when applying an analytical formula to ensure that all conditions and limitations of the formula are observed. Because of the approximate nature of the formulas,"bounding" cases should be investigated to ensure that the results are logical and conservative. Analytical methods are useful for acquiring a general understanding of the stresses and deformations of a structure or component, but other techniques such as numerical and experimental methods should be employed for detailed studies of the structural response of the component to a given loading condition.

2.5.1.2 Numerical methods

There are two established numerical methods which are generally denoted as finite-difference and finite-element methods. The finite-difference methods are widely used for field flow problems, such as electromagnetics, heat-transfer, and fluid flow, and are seldom used for structural validation. On the other hand, the finite-element methods are used for both structural static and dynamic analysis and are best suited for complex structures with complicated boundary conditions and loadings. Both of these methods are approximate approaches to describing the geometry and loading conditions.

Many general purpose finite-element programs are available on the market. Examples of some programs are LS-DYNA3D,[20]LS-NIKE3D,[25] ADINA,[26] PRONTO,[23] ANSYS,[22] and ABAQUS,[21]. Whatever program is used by the designer or analyst, it should be benchmarked, verified, and validated to show that it produces result that can be believed. Most of these programs can be used for various types of analysis including linear, nonlinear, static, or dynamic. Linear analysis assumes that after all loads are removed from a structure, the structure returns to its original shape and stress condition, which is typically stress free. Nonlinear analyses allow for plastic behavior of materials and geometric nonlinearities such as contact between components and geometric stiffening. Static loads such as gravity and atmospheric pressure are considered to be constant for long periods of time. Dynamic loads are time varying and can occur in a very short period of time, as in the case of impact analysis. Since the most severe loading package must survive is likely to be the impact load from the 30-foot free-drop test, its structural evaluation is one of the most complicated analyses. Details of impact analysis are in Methods for Impact Analysis of Shipping Containers[27].

For dynamic impact analysis, the finite-element method provides the most accurate technique available from the computation aspect. This procedure is an extension of the traditional energy method of analysis; when the solution technique is based on explicit formulations, the package response provides stresses due to wave propagation and deformation introduced due to external loading. The traditional method for the finite-element analysis is associated with the system response to an initial velocity uniformly applied to the system. An alternative procedure involves the determination of the system response to a prescribed uniform acceleration field that is obtained from an experimental program, where the accelerations may be applied to the boundary of the system or uniformly imposed on the entire model.

The characterization of the material properties of the components of the package must be properly addressed in order to obtain quality results using the finite-element method. For metallic components, power law representation of the stress-strain law provides an adequate model. The parameters needed for input to this model are measured experimentally at the appropriate level of strain rate that is comparable with the strain rates associated with the impact conditions.

Impact limiters that are made of foam-like materials, where the stress-strain relationship is based on a deviatoric and volumetric relationship, can be modeled in a simple form using a static uniaxial load-deflection curve for each axis of the material. If the input data are available, a crushable foam model is used. In the crushable foam model, the pressure-volumetric strain curve and an estimate of a perfect plastic yield function defined by three constants are required.

Wood components of the package are normally modeled using a bi-linear stress-strain relation; the parameter inputs for this model are measured experimentally for each axis of the material model.

For both wood and foam-like materials, the material model should be verified analytically by using a single element model subjected to uniaxial compression. The results are then compared with the appropriate experimentally determined load-deflection curve used to obtain the input parameters.

The interaction among structural components of the package can be properly addressed by using the concept of slide surfaces and the effects of friction (both dynamic and static). Finite-element codes like LS-DYNA3D,[20] PRONTO,[23] and ABAQUS/EXPLICIT[21] employ these techniques.

The solutions obtained from the finite-element method generally address primary impact for a generalized orientation of the package onto the unyielding surface; however, solutions can be extended to include the secondary impact for oblique type impact of the container. Results from the finite-element dynamic impact analysis can be utilized to compare with experimentally obtained results in actual container drop tests.

2.5.2 Experimental Methods

This section provides guidance for planning, conducting, and reporting the structural tests which are used to demonstrate the safety performance of weapon shipping packages. Federal regulations in 10 CFR 71 specify tests to be used to demonstrate the safety performance of packages used to transport radioactive materials. Normal Conditions of Transport and Hypothetical Accident Conditions are specified, and these conditions are summarized in Subsect. 2.2.2. The critical structural tests are the free drops and the puncture or penetration tests. For these tests or drops, the package must strike in a position that produces the most damage to the package. After the drops, the package must be able to meet specific minimum requirements on containment, shielding, and subcriticality.

To meet the intent of the regulations, sufficient evidence must be collected from a drop test program to demonstrate several conclusions about the safety performance of the package. First, all critical drop conditions that can produce the worst damage to the package's containment, shielding, and subcriticality control systems must be identified. Additionally, the worst possible damage must be produced in the drop test program. Finally, the package's ability to meet the regulatory requirements on containment, shielding, and subcriticality must not be jeopardized by the worst possible damage.

To demonstrate the package performance by test alone requires that all potentially damaging test conditions must be executed and the damaged test specimens or models must be evaluated to verify the damaged package's ability to meet regulatory requirements on containment, shielding, and subcriticality. Because the actual radioactive contents in a package are normally not used in the testing, direct evidence cannot be obtained to demonstrate the package's capabilities. The high costs of prototype specimens and drop testing may also prevent all the critical conditions to be tested. Therefore, almost all test programs have to depend on some indirect evidence to demonstrate a package's compliance with regulatory requirements. Indirect evidence is typically from analyses and technical arguments which can vary greatly with the package and the test program.

Structural tests of shipping packages for the transportation of radioactive materials are not standardized or routine. Each package must have its own test plan which is appropriate for the package design, the package behavior under impact, and the approach used to comply with regulatory requirements. The test plan (or test matrix) is the critical ingredient in a successful and meaningful test program for a transportation package.

2.5.2.1 Test planning

Conducting a test is very similar to performing an analysis because both operations employ models to obtain results and input is fed into the model in order to produce desired output. In the case of testing, the input is the test conditions and the output is the measurement of the model response. To achieve a perfect test, techniques must be specified and developed to impose the test conditions exactly; instruments must be devised and calibrated to measure the desired model responses accurately; and models must be designed and constructed to reproduce the hardware precisely. Such a perfect test can never be attained because of various economical and engineering limitations. The reliability of test results is affected by many limitations, approximations and simplifications. These effects must be identified and evaluated in the test plan as follows:

- Define the precise objectives of the test

- Specify the necessary measurements

- Design the test model; identify all omissions, simplifications, and deviations of the model from the prototype of the actual package; and show that the effects are insignificant on the basis of the governing physical phenomena

Transportation package tests can be loosely divided, according to their purposes, into two groups. These groups are impact tests and acceptance tests and are discussed in the following paragraphs.

Impact tests

The impact process excites three structural responses: Waves, vibrations and quasi-static deformations; all three can cause damage to the package. However, the relative significance of their contributions to the damage varies with the package design. In packages without impact limiters, waves are the major contributors to damage, while in packages with soft impact limiters, quasi-static deformations generated by the peak rigid-body acceleration of the impact are the dominant contributors. Vibrations can also cause damage by reinforcing quasi-static deformations.

Understanding the dominant causes for damage is essential to the planning of a structural test. If waves are the dominant causes, damages can occur at locations far from the impact point and near material and geometry discontinuities where waves are reflected. However, if quasi-static deformations prevail, the damages are likely to occur near the impact area and weak sections of the package. Accordingly, the details to be included in the test models for these two cases need to be different. Furthermore, the location, type, and frequency response of instruments used to monitor the impact response should also be different.

Other factors that can have significant influence on the test plan are failure modes and material properties. One type of failure mode, buckling, is not only geometry dependent but also highly sensitive to local stress and boundary conditions. These characteristics of buckling make the use of scale models and partial models very difficult, especially for packages using fin and honeycomb impact limiters. Similar to buckling failures, fracture failures also depend on many ill-defined and difficult-to-control parameters. Fracture failures are not only sensitive to local geometries and stresses but are also determined by the size of existing cracks and defects in the hardware. The use of a scale model is almost impossible for a package which is expected to have failure by fracture. The use of a reduced-scale model is also limited for materials with large microstructures like composite and honeycomb materials. Test results may exhibit size effects when the dimensions of the package model are comparable to the dimensions of the material microstructures. Finally, scale models are not recommended for leak tests because the leak rate depends on many parameters which a scale model cannot accurately reproduce.

The test objectives, the package design, and the potential package failure modes provide the necessary basis for the selection of the critical test conditions. Potential failure modes should be first identified for all the safety-related components to be tested. The components include the containment vessel, closure lid, shielding components, impact limiters, and thermal insulations. Drop conditions which deliver a large portion of the impact energy to potential failure modes in a short time qualify as critical test conditions. Usually, these drops produce either a high rigid body deceleration of the package or a large localized damage at the impact area. Potential test orientations and their failure modes are:

- Direct impact on structural weaknesses (thermal insulations, welds, bolted closure joints, valves, and penetrations)

- Simultaneous impact on large surface area (end drops)

- Impact on stiff package surface area (drops on lifting lugs)

- Impact on package areas not protected or underprotected by impact limiters

- Impact with full package weight over the impact area (center of gravity over corner drops)

- Impact in the most unfavorable direction of the failure mode (side drops for a beam-like mode of failure and drops in the direction of weak buckling strength)

- Impact producing highly localized damage (puncture drop at a weak surface area surrounded by strong area)

The impact conditions should be specified with accompanying environmental and operational conditions. The most unfavorable of these conditions in terms of the containment, shielding, and subcriticality capabilities of the package should be selected. For example, drop tests to demonstrate the containment capability should consider the extreme cold temperature because the cold temperature makes closure bolts more brittle and seals less effective. The tests for containment should also be carried out with the maximum internal pressure for the given environmental conditions and contents.

Acceptance tests

The requirement for containment in a shipping package is commonly demonstrated using equivalent leak tests. Helium is a common medium used for the tests where seal permeability is not a problem. Tests are conducted at a standard pressure, and the measured leak rate is analytically converted to the rate corresponding to the package internal pressure and contents. Unfortunately, not all analytical methods for this conversion are well defined or generally accepted. Therefore, it may be desirable to produce additional evidence of containment in drop tests. One approach is to measure with strain gages the deformation of the closure bolts during and after a drop test. The lack of permanent deformation would provide further confidence of containment. Another measurement is the opening of the joint between the containment vessel and closure lid. This measurement, however, is difficult since in most cases the joint would separate very little.

The requirement for shielding in a shipping package can be demonstrated by placing a radioactive source in the package and measuring the radiation outside the package. However, it is usually more convenient to inspect the damage of the shielding components and determine whether or not the shielding capability of the package has been significantly degraded. Radiography is commonly used for the inspection and the lack of excessive plastic deformation; fracture of the shielding components is evidence of undamaged shielding capability.

The demonstration of the subcriticality of a shipping package must depend on a criticality analysis. This analysis defines the conditions for subcriticality in terms of a number of measurable deformations of the subcriticality-control components of the package. The acceptability of these deformations can be confirmed by post-test inspections.

The flat target area for drop tests should simulate the unyielding surface required by the regulations. To meet this requirement, the target should first have sufficient mass and constraint to prevent appreciable overall rigid-body movement of the target caused by the impact. Second, the target should have sufficient stiffness and support to minimize the overall target deformation under the impact force. The target should also have a hardened surface to minimize the local target deformation surrounding and under the impact area. Finally, the target should be able to disperse the stress waves generated by the impact to minimize the effects of reflecting waves and vibrations on the impact process. A common design of the flat target is a massive, reinforced concrete block having a total mass at least ten times the mass of the package. The impact surface of the block is protected by a thick steel armor plate which is welded or anchored to the steel reinforcements in the concrete. Guidance on designing targets which are unyielding surfaces is provided in IAEA Safety Series Number 6[28].

A flat, unyielding target designed for full-scale tests should be adequate for reduced-scale model tests. However, the small mass for the penetration test and the cylinder or puncture bar for the puncture test should be similarly scaled as the package model for scale model drop tests.

2.5.2.2 Test conduct

Measurements and recordings of the test conditions and model should be made before, during, and after a test. Before and after the test, the test model should be visually inspected, photographed, and checked with a complete dimensional survey. Nondestructive and destructive examinations of the model are conducted if they are needed to substantiate the findings of the test. During the test, high-speed photographs of the impacting model should be taken to confirm the actual test conditions. Temporal records of accelerations, displacements, and strains at selective locations in the test model should also be taken as needed. This provides additional evidence of the safety performance of the package, obtains quantitative results for the verification of analysis method and model, and obtains a precise understanding of the impact behavior of the package. Details of nondestructive and destructive examination methods as well as techniques to measure accelerations, displacements, and strains are discussed in Impact Test Qualification of Shipping Casks[29].

2.5.2.3 Test procedure, records, and report

The test plan should be expanded into a detailed test procedure to ensure that the tests be carried out exactly as planned. The test procedure should specify the sequence and requirements of the following operations:

- Pretest measurements and examinations of the test model

- Setup and calibration of instruments
- Creation of test conditions
- Execution of the test
- Post-test calibration of instruments
- Post-test measurements and examinations of the test model

A useful tool in developing the test plan is a test matrix. This matrix should include all of the structural tests required by the regulations. For each test, all the parameters which are critical to the ability of the package to maintain containment, shielding, and criticality should be listed. These parameters include package configuration, temperature, pressure, geometry, and orientation. The test matrix is made with all the conceivable combinations of the parameters. Tests, analyses, or appropriate justifications are then used to fill in the "boxes" of the matrix.

The specified requirements should include the standards or acceptance criteria to be used for these operations. All deviations from the test procedure must be documented. The instrument settings, the test conditions, and the visual observations must also be documented. Photographs and tapes of the test must preserved for reference and audits.

The test report should provide key information on all the preceding topics. The report should contain at least the followings:

- Objectives of the test

- Design of the test model and a list of all omissions and deviations from the prototype

- Expected dominant impact responses

- Rationale for the test

- Results of pretest measurements and examinations

- Test procedure and setup

- Capability of instruments used and the test data obtained

- Results of posttest measurements and examinations

- Interpretation of the test data and results

- Understanding of the impact response of the test model gained through the study of the test results

- Conclusions of the test and the supporting evidences and arguments

The test report can be included in the SARP or referenced by the SARP. If it is referenced by the SARP, all of the information that demonstrates the package compliance with the regulations must be summarized in the SARP.

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APPENDICES

A STRUCTURAL EVALUATION

B DESIGN CALCULATIONS FOR CONTAINMENT VESSEL

C CALCULATIONS FOR NORMAL VIBRATION DURING TRAILER SHIPMENT OF THE PACKAGE

D CONTAINMENTAL VESSEL STRUCTURAL ANALYSIS

E PRELOAD CONDITION OF CONTAINMENT VESSEL BOLTING

F FATIGUE LOADING OF CONTAINMENT VESSEL BOLTING

G DESIGN AND ANALYSIS OF CLOSURE BOLTS

NOTE: This appendix is intended to be an <u>example</u> of the type of calculations that might be completed for the design of a DOE drum-type shipping package and should not be used as an indication that the requirements are satisfied by these calculations.

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APPENDIX A

STRUCTURAL EVALUATION

A drum-type shipping package used to transport powered uranium oxide has been chosen to indicate the type of structural calculations that may be required for a structural evaluation. This package is a Type B package. It consists of a 35-gallon stainless steel drum having a removable top, and a stainless steel inner containment vessel. The space between the inner surface of the drum and the outer surface of the containment vessel is filled with a fiberboard material that provides positioning of the containment vessel as well as thermal and impact protection. While the packaging materials provide some shielding, the package contains no features intended to provide shielding. This package requires no active cooling system and contains no coolants.

The shipping capacity of the package is up to 6.5 kg (14.3 pounds) of uranium oxide power. The content heat load is limited to 5 Watts per package. The package with 13 kilograms of uranium oxide powder is a Fissile Class I package, based on the criticality evaluation. The package containing up to 6.5 kilograms of uranium oxide powder, meets the requirements of 10 CFR 71.55 and 10 CFR 71.57 for a Fissile Class I package. No transport index is required for criticality safety purposes.

The main functions of the packaging are to provide containment, thermal protection, and nuclear criticality safety. The package consists, primarily, of a 35-gallon drum with removable top, fiberboard insulation, and a stainless steel containment vessel. The 35-gallon drum forms the boundary of the insulation, and the containment vessel during all transport and storage conditions. The package external dimensions are 35 inches (89 centimeters) in height, and 19-5/8 inches (49.8 centimeters) in diameter at the drum rolling hoops. The weight of the package without contents is 98.9 kilograms (218 pounds). The gross shipping weight of the package, including the contents, is 112.2 kilograms (247 pounds). The height of the center of gravity of the loaded package is 20 inches (50.8 centimeters) above the base of the drum.

Drum boundary

The package uses a 35-gallon drum with a removable lid to provide protection and confinement of the fiberboard insulation and containment vessel during normal transport and Hypothetical Accident Conditions.

The 35-gallon drum body and ends are fabricated from 16-gauge, Type 304 stainless steel sheet in accordance with U.S. Department of Transportation (DOT) Specification 17C.[1] The body seams are welded. The drum has two circumferential rolling hoops (chimes) formed into the body. To construct the package, the drum is modified from DOT 17C by adding an identification plate, a security seal, and four, 3/8 inches (1 centimeter) diam holes equally spaced around the circumference near the top of the drum. These holes prevent over pressurization of the drum during use. These vent holes are sealed with BP caplugs, or weatherproof tape, to prevent leakage of water into the drum during transport or storage.

The removable lid is attached to the drum body by a bolted closure ring, or hoop. The closure ring is fabricated from 12-gauge stainless steel with drop-forged lugs. The weld between the closure ring and lugs ensures adequate performance during the Hypothetical Accident Conditions. The closure ring lock bolt is a 5/8 inch, high strength, steel bolt. The lock bolt is tightened to 50 foot-pounds (nominal torque) for transport. The bottom end of the drum is welded to the body, and is not removable. The inner and outer surfaces of the drum are not treated in any way.

Containment

The containment vessel, together with the inner O-ring, provide the containment boundary of the package, preventing the release of the contents to the environment. The containment vessel also prevents moisture from reaching the contents. Some radiation shielding is provided by this vessel, although this is not its primary function.

The containment vessel is designed in accordance with the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section VIII, Division 1.[2] It is fabricated from Type 304 stainless steel, and consists of the body and closure lid. The containment vessel is designed for an internal pressure of 70 pounds per square inch, gauge at ambient temperature.

The containment vessel body is fabricated from 5 inches, schedule 40 seamless pipe, a standard weight ellipsoidal pipe cap, and a 150-pound weld-neck flange. Both the pipe and cap have a 0.258 inch (0.66 centimeter) wall thickness and full penetration butt welds attach the cap and weld-neck flange to the pipe. The containment vessel closure lid is a blind flange with a thickness of about 13/16 inches. All outer and inner surfaces are bead blasted to a matte finish.

The containment vessel has an inside diameter of 5 inches (12.7 centimeters), an inside length of 21.25 inches (54 centimeters). The volume is approximately 6.8 liters (415 cubic inches). The assembled weight of the containment vessel, without contents, is 68.55 pounds (31.2 kilograms). When filled with a maximum of 13 kilograms of uranium oxide powder, the containment vessel weight is about 44.2 kilograms (97.2 pounds).

The containment vessel flanged joint is attached with eight 3/4 inches high strength steel bolts that are tightened to 30 feet-pounds (nominal torque). Eight 3/4 inches hex nuts are welded to the underside of the flange to allow bolt tightening from above.

The flat face of the weld-neck flange contains two machined O-ring grooves. Two Buna-N polymer O-rings (Parker compound N163-70[3] or equivalent) will be used. O-ring properties are listed in Parker Seals Technical Report No. KT1726.

The inner O-ring, together with the containment vessel structure and bolted closure lid, form the containment boundary. The O-ring seal design permits assembly verification leak testing of the containment vessel by measuring the leak rate in the space between the inner and outer O-rings. The closure lid is provided with a leak test port for post-load leak testing. During transport, the leak test port is sealed with a commercial grade high pressure plug and gland nut.

The containment vessel lid contains two counterbored holes, outside the O-ring seal area for insertion of ball lock pin tools. These tools facilitate handling the lid and vessel during loading and unloading operations.

Thermal Protection and Heat Dissipation

The external surface temperature of the package is not affected by the contents thermal limit of *5* Watts, nor does the heat output result in temperatures that approach the normal operating temperature limits of any materials used in the package. Therefore, the package does not have design features for cooling or dissipation of heat. The package is passively cooled. No coolants are utilized within the package or are there active cooling systems.

The drum is filled with a fiberboard material that surrounds the containment vessel and insulates it during the hypothetical fire accident condition. The drum ensures that the insulation material remains in place during accident and normal transport conditions. The insulation material also provides vibration and impact protection, and centers the containment vessel within the drum.

The insulation material is an industrial cane fiberboard that complies with ASTM Standards C-208-72and C-209-84[4],[5]. The fiberboard has a density of 15 to 18 pounds per cubic feet. There is a nominal thickness of 5.8 inches (14.7 centimeters) of fiberboard between the containment vessel and drum wall. Insulation thickness between the containment vessel and the drum ends is about 4.8 inches (12 centimeters) at the top, and 4 inches (10 centimeters) at the bottom.

Fiberboard assemblies are constructed from disks cut from 1/2 inch sheet stock. These disks are cemented together to form 2 inches thick disk assemblies. These disks are stacked and joined together with tie-rod bolts to simplify handling.

Nuclear Criticality Safety

Nuclear criticality safety is provided by process control of the fissile material content, and by geometrical control of the contents. Detailed criticality analysis demonstrate that the package remains subcritical with a full content load of 13 kilograms of uranium oxide powder. The analysis also demonstrates that the package remains subcritical with water in the containment vessel.

Shielding

The package does not contain material specifically intended to provide shielding. Shielding from the contents is provided, to some extent, by the packaging materials of construction and the distance from the uranium oxide powder to the package surface. During the Hypothetical Accident Conditions, the containment vessel, and insulation, remain confined within the drum; however, for conservatism, the shielding evaluation considers that the drum and insulation are not present after the accident.

Neutron Reflectors, Absorbers, and Moderators

There are no materials in the package that are intended to act as a neutron reflector, absorber, or moderator for criticality safety purposes. Certain of the materials of construction possess these characteristics, and are considered in the criticality analysis. The thermal insulation acts as a neutron reflector to the contents of a single package, and as a neutron moderator in any array of packages. The stainless steel of the containment vessel and drum acts as neutron reflectors to the contents of a single package, and as neutron absorbers in any array of packages. The nuclear properties of the materials of construction, and of the contents, have been considered in the criticality safety evaluation.

Lifting and Tiedown Devices

No fixture, ports, or devices are provided for lifting the package, and there are no protrusions on the package that could be used for that purpose. There are no tie-down fixtures or devices on the package nor are there any other features that could be used for that purpose.

Lifting of the package is performed using standard industrial drum grapples that grip the drum circumference. These devices are typically of an ice tong clamp design, but lifting can be performed using fork lift clamps, slings, or a pallet under the drum.

The containment vessel and its lid are lifted and handled using ball lock pin tools inserted in two counterbored holes in the containment vessel closure lid.

Structural Description

This package consists of a stainless steel 35-gallon drum, fiberboard insulation with sealant, stainless steel containment vessel, bolts, O-rings, and contents (See Fig. A1). The cavity created in the fiberboard insulation is 6-1/2 inches inside diameter and about 24 inches inside height. All exposed surfaces of the fiberboard are painted with a weatherproof coating to reduce moisture absorption from the atmosphere and to reduce dust. The drum provides the outer boundary, and the containment vessel and the inner O-ring provide the single containment boundary of this package (Fig. A2).

The containment vessel is fabricated from standard schedule 40 seamless stainless steel pipe, an ellipsoidal lower head (a 5 inch, standard weight pipe cap), and a 5 inch, 150 pounds welding neck flange (Fig. A3). The flange is machined to provide two concentric O-ring grooves in the flat face. A 5 inch, 150 pounds blind flange, machined to 13/16 inch thick, is provided as the vessel lid to complete the assembly. The blind flange is provided with lift features and a leak-check port between the O-ring grooves. The two flanges, with the O-rings in place, are joined together with 8, 3/4-inch, 10 UNC-2A, 2-1/2 inch long, high-strength, steel bolts. There are no penetrations or connections with fittings into the sealed containment vessel. To meet the requirements for package certification, the containment vessel

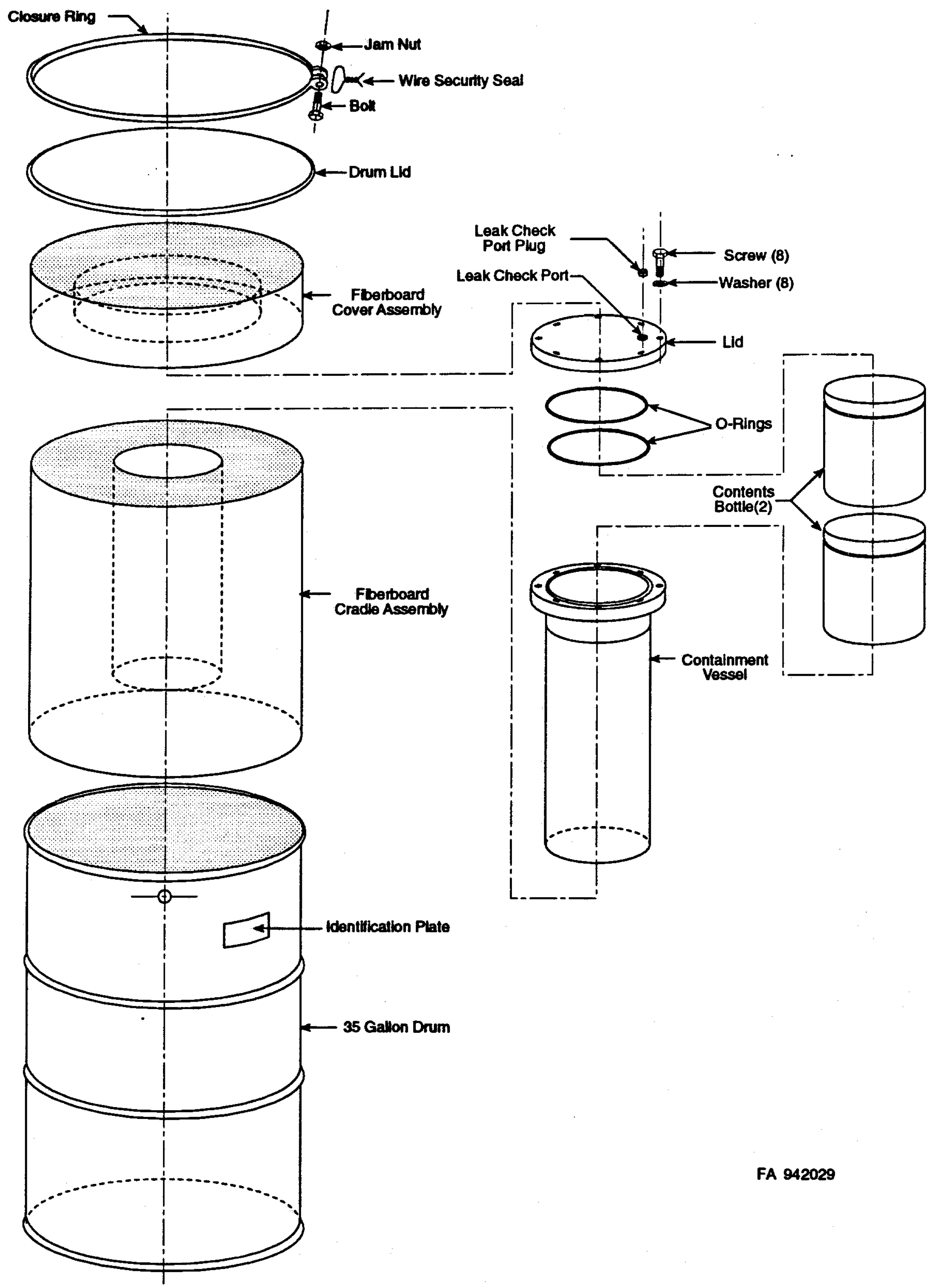

Fig. A.1. Typical drum-type shipping package.

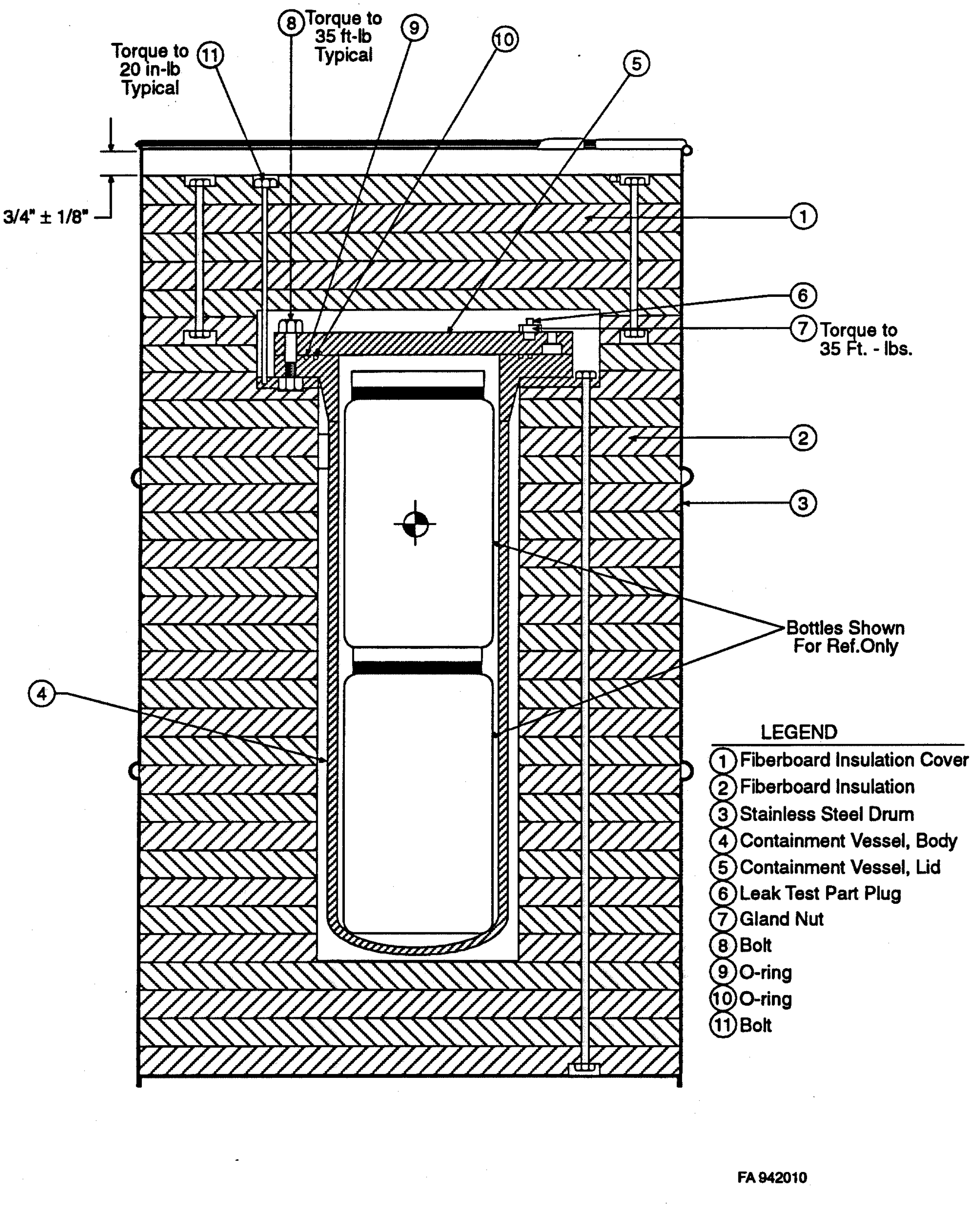

Fig. A2. Shipping package drum.

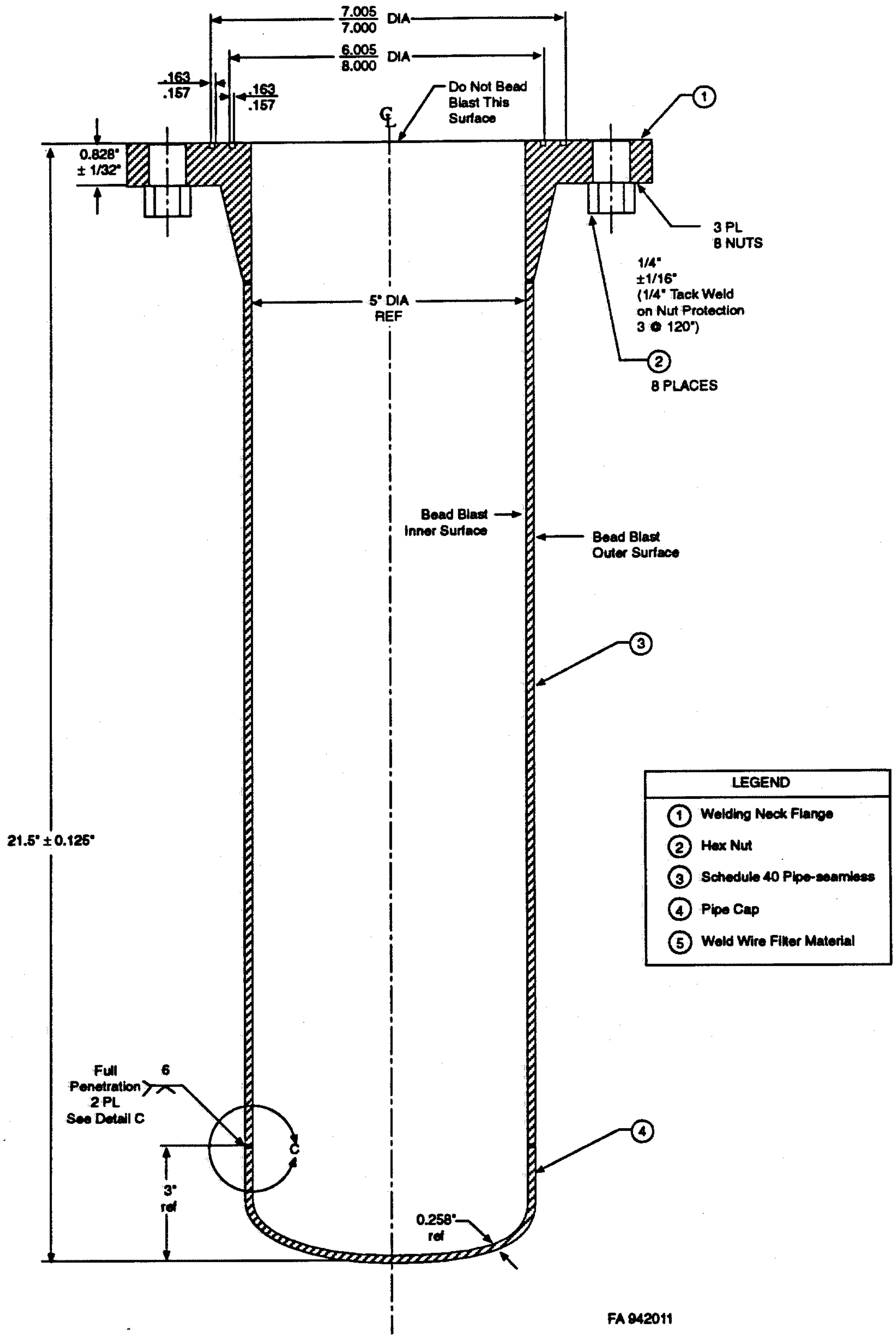

Fig. A3. Shipping package containment vessel

must remain intact during all conditions of transport. This integrity must be demonstrated by test or other acceptable methodology for Normal Conditions of Transport and for Hypothetical Accident Conditions.

Design Criteria

The outer boundary is a stainless steel 35-gallon drum with construction and thickness as noted on Fig. A4. The containment vessel, defined as the containment boundary, is designed and fabricated in accordance with Section VIII, Division 1 of the ASME Boiler and Pressure Vessel Code. Allowable stress limits for the containment vessel as specified in the ASME Code, Section VIII, Division 1 (operating range of minus 20 to 200 degrees Fahrenheit) are as follows:

Forging, F304L stainless steel (flange)	14,300 psi
Forging, F304 stainless steel (lid)	16,600 psi
Bolts and nuts, Grade B8C	17,900 psi
Fitting, WP304L stainless steel (bottom head)	13,300 psi

The calculated stresses (Appendix B) in all components of the containment vessel are below the allowable limits at the design conditions. Comparison of a number of calculated stresses to allowable stresses are tabulated in Table A1.

Comparison of allowable external and internal pressures to design pressures as well as the required wall thickness to the actual thickness are also given in Appendix B. All allowable pressures and all actual wall thickness are well above the 10 CFR 71 design requirement of 21 pounds per square inch external pressure and the defined design requirements of 70 pounds per square inch internal pressure.

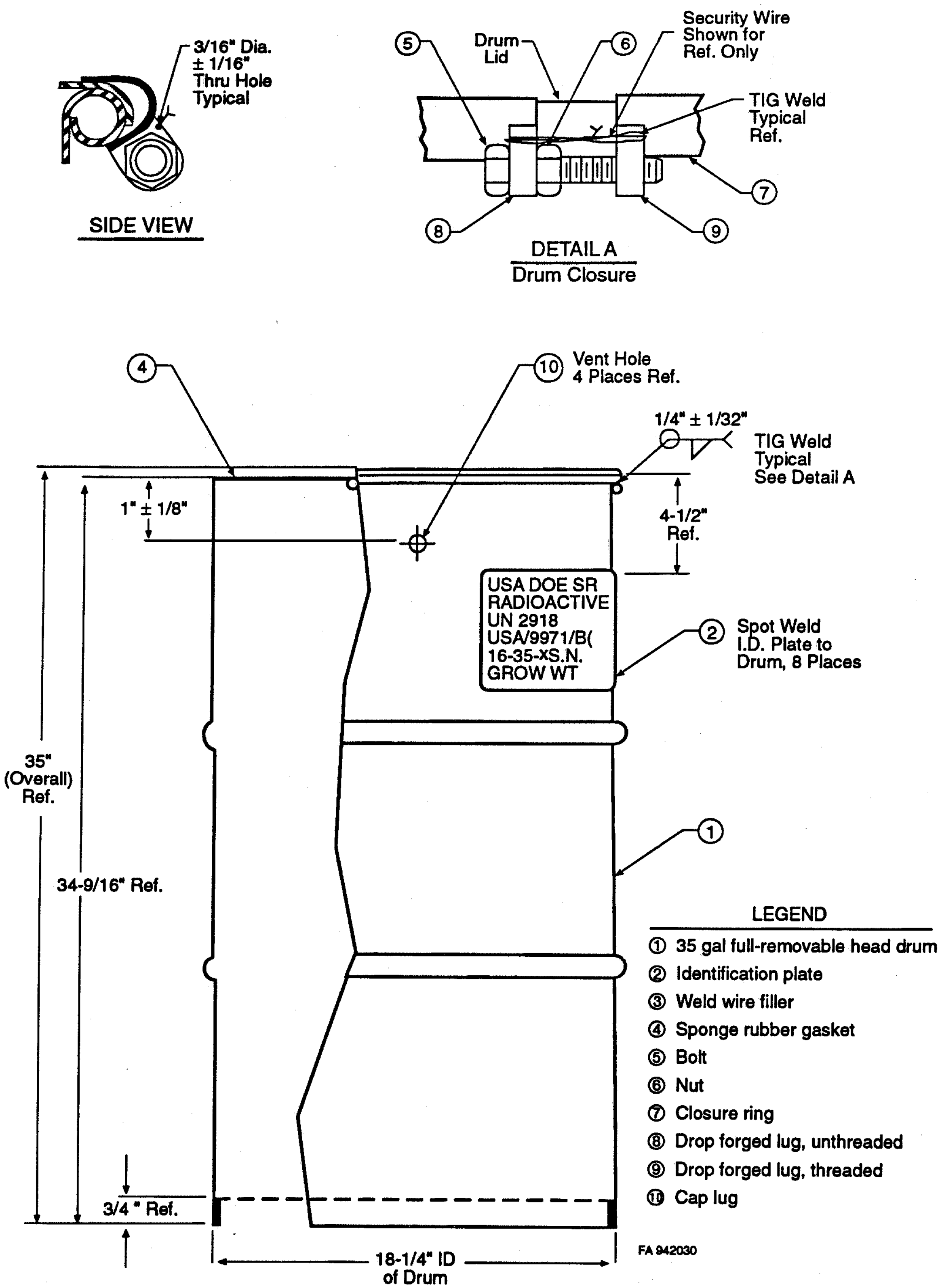

Fig. A4. Shipping package drum.

Table A1. Stress comparison

Item	Design Condition	Calculated Stress (psi)	Allowable Stress (psi)
Longitudinal hub stress	71 psig internal	11,176	14,300
Radial flange stress	71 psig internal	13,031	14,300
Tangential flange stress	71 psig internal	4,671	14,300
Longitudinal compressive stress	21 psig external	119	13,000
Stress in cylinder wall	71 psig internal	836	14,300
Stress in lower head	71 psig internal	800	13,300
Bolt stress	71 psig internal and 30 ft-lb torque	8,504	17,900

The stresses used in the design of all metal components are in the elastic range of the material properties. Brittle or fatigue failures are not anticipated under any design, transport, accident, or storage condition. Specific design information for packaging is given in subsequent paragraphs.

Drum

The drum is stainless steel with a ring clamp closure, manufactured as shown on Fig. A4. Four 3/8 inch equally spaced holes are drilled in the top sidewall to prevent pressure buildup in the drum. The holes are sealed from the inside with plastic plugs (BPF 3/8 inch Caplugs[R]) to provide a moisture barrier for Normal Conditions of Transport. The drum body and heads are fabricated from 16-gauge stainless steel. A rolled 12 gauge stainless steel closure ring with drop forged lugs provides closure. The drum lid closure ring is secured with a 5/8 inch high-strength steel bolt tightened to 50 feet-pounds (nominal torque).

Insulation

The drum is lined with fiberboard insulation that complies with ASTM C-208 and has a density of 15 to 18 pounds per square feet. The insulation has a nominal thickness of 5-13/16 inch on the sidewall of the drum, 4 1/4 inch on the bottom of the drum, and 4-15/16 inch on the top of the drum. All exposed surfaces of the insulation are coated with a weatherproofing compound to reduce moisture absorption from the atmosphere and to reduce dust. The insulation has a normal operating temperature limit of 250 degrees Fahrenheit degrades above 280 degrees Fahrenheit, and ignites at 425 degrees Fahrenheit in air.[6]

Containment Vessel

The containment vessel is designed and constructed in accordance with the ASME Boiler and Pressure Vessel Code, Section VIII, Division 1. Title 10 CFR 71.73(c) requires that the containment system be immersed in water with to an external water pressure equivalent to at least a 50-foot head of water, which equates to an external pressure of 21 pounds per square inch gauge. The design analysis (Appendix B) shows that the containment vessel is conservatively rated for the external pressure requirement as well as a pressure differential of 70 pounds per square inch, accordance with the ASME Code.

The containment vessel is sealed at the flanges with concentric elastomer O-rings. An evacuation port is located between the O-rings to facilitate post-load leak testing. A package assembly verification air leak rate of 10^{-3} atmosphere-cubic centimeter per second must be demonstrated before the package is released for transport. This assures effective O-ring sealing. The evacuation port located in the top flange is sealed after the leak test with a pressure plug and gland nut.

WEIGHTS AND CENTERS OF GRAVITY

The weights of the packaging components for the test unit for shipment are given as 264 pounds. The center of gravity for the shipping package is shown in Fig. A5.

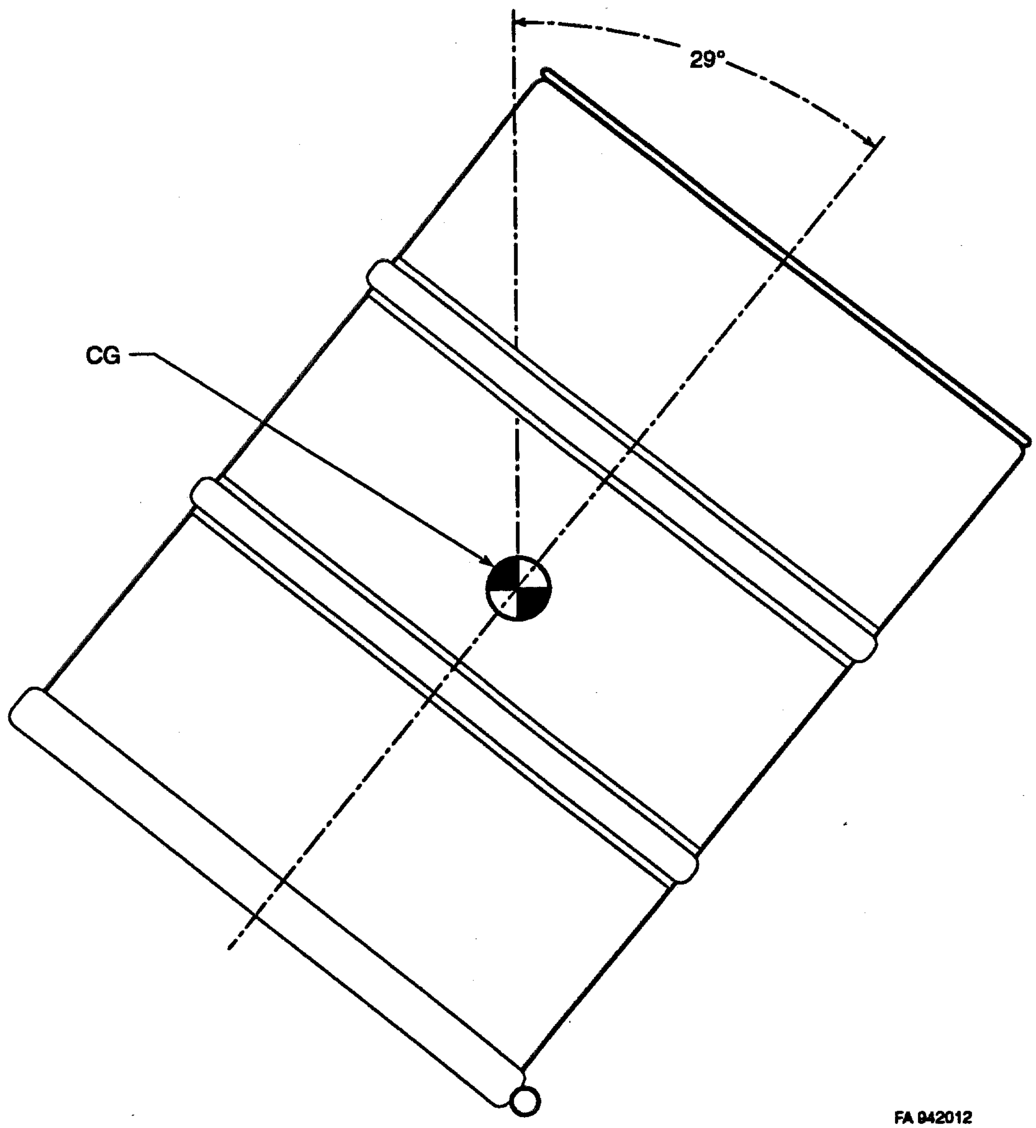

Fig. A5. Shipping package center of gravity.

MECHANICAL PROPERTIES OF MATERIALS

The mechanical properties of the packaging are presented in Tables A2-A6. Design temperature ranges are listed where required to establish allowable stresses used in the containment vessel design calculations (Appendix B).

GENERAL STANDARDS FOR ALL PACKAGES

This section demonstrates compliance with 10 CFR 71.43, *General Standards for All Packages.*

Minimum Package Size

Requirement. The smallest overall dimension of a package shall not be less than 4 inches.

Analysis. The drum inside diameter is 18-1/4 inch and the inside height is about 35 inches. The containment vessel minimum outside diameter is about 5-1/2 inches, and the overall height is about 22-1/3 inches. Therefore, the package meets this requirement.

Tamperproof Feature

Requirement. The outside of the package shall incorporate a seal that cannot be readily broken and that, while intact, provides evidence that the package has not been opened by unauthorized persons.

Analysis. The removable head of the drum is attached to the body by a closure ring and high-strength alloy bolt with lock nut. Two 3/16 inch-diameter holes are drilled through the closure ring lugs

Table A2. Mechanical properties of drum [a]

Materials of construction	
Drum	304 Stainless steel, SA-240
Closure ring	304 Stainless steel, SA-167
Closure lugs	304F Stainless steel, SA-182
Closure bolts	304 Stainless steel, SA-194
Ultimate strength (psi)	85,000
Yield strength (psi), 0.2% offset	35,000
Elongation in 2 in. (%)	50
Modulus of elasticity (psi)	28.0×10^6
Minimum thickness (gauge)	
Body	16
Heads	16
Closure ring	12
Service temperature range (°F)	-40 to 1475

[a] Pages. 17 and 20.[7]

Table A3. Mechanical properties of insulation [a,b]

Material	Industrial cane fiberboard
Tensile strength (psi)	
Parallel to surface	7200
Perpendicular to surface	40,000
Compression	500
Linear expansion, 50-90% RH average maximum (%)	0.5
Modulus of rupture, average (%)	40
Modulus of elasticity	
Water absorption by volume maximum (%)	10
Normal operating temperature (°F)	250
Degradation temperature (°F)	285
Ignition temperature in air (°F)	425
Density (lb/ft^3) (as delivered)	15 to 18
Modulus of elasticity (psi): compression; tension	500;40,000

[a] ASTM C-208.[4]
[b] DP-1292.[6]

Table A4. Mechanical properties of containment vessel [a,b]

Materials of construction	Type 304L stainless steel; SA 312 pipe, SA 184 flange, and SA 403 pipe cap
Design temperature range (°F)	-20 to 200
Ultimate strength, minimum (psi)	85,000
Yield strength (psi), 0.2% offset	30,000
Elongation in 2 in. (%)	50
Modulus of elasticity (psi)	28.0×10^6
Allowable stresses (psi)	
Seamless pipe	14,300
Flange	14,300
Lid	16,600
Lower head	13,300
Coefficient of thermal expansion (in./in./°F)	0.0000098
Service Temperature Range (°F)	-40 to 1475

[a] ASME Code.[2]
[b] Pages 17 and 20.[7]

Table A5. Mechanical properties of O-rings [a]

Material composition	Buna-N or Nitrile
Normal service temperature range (°F)	-45 to 250
Permissible exposure time at 400 °F (min)	45
Hardness Durometer (Shore A)	70 ± 5
Elongation, minimum (%)	100
Fabrication method	Molded
Coefficient of thermal expansion (in./in/°F)	0.000062
Vendor seal compound number (Parker)	N163-70

[a] Parker O-ring Handbook.[3]

Table A6. Mechanical properties of bolts for the containment vessel [a,b,c]

Material	High-strength stainless steel ASTM A193 (Bolts), ASTM A194 (nuts)
Grade	B8C (bolts), 8C (nuts), Class 1
Composition	Type 347
Service, temperature range (°F)	-40 to 1475
Ultimate strength, minimum (psi)	75,000
Yield strength, minimum (psi), 0.2% offset	30,000
Allowable stress (psi)	17,900
Elongation in 2 in. (%)	30
Modulus of elasticity (psi)	30×10^6
Heat treatment conditions	Quenched and tempered
Coefficient of thermal expansion (in./in./°F)	0.000007

[a] ASTM A193.[8]
[b] ASTM A194.[9]
[c] Page 5-5.[10]

for use with wire-type security seals. The security seal consists of a steel wire with a copper crimp closure. The requirement is satisfied by the installation of the seal with each use.

As an additional tamperproof feature, the containment vessel bolt heads have 1/8 inch holes for wire seals. The security wire is threaded between the bolts and insures that the bolts are not loosened by unauthorized persons.

Positive Closure

Requirement. Each package must include a containment system that can be securely closed by a positive fastening device that cannot be opened unintentionally.

Analysis. The closure system of the drum with tamperproof features, provides assurance that the outer boundary will not be unintentionally breached. The containment boundary is sealed using eight 3/4-inch bolts with nuts welded to the flange to ensure that this boundary will be breached only through a deliberate effort, and then only after the outer boundary is breached.

Chemical and Galvanic Reactions

Requirement. A package must be of materials and construction that assure there will be no significant chemical, galvanic, or other reaction among the packaging components or between the packaging components and the package contents, including possible reaction resulting from leakage of water into the containment vessel to the maximum credible extent.

Analysis. Starting with the outer component, the packaging consists of the drum (Type 304 stainless steel), insulation (cane fiberboard), containment vessel (Type 304 stainless steel), bolts (stainless steel), plastic bag (polyethylene), and the contents (polyethylene or polypropylene bottles).

All metal components of the packaging are stainless steel; thus, coupling by galvanic reaction is not expected to occur. The assembled components are protected from the weather and inspected at the time of packaging; therefore, the package will not contain any free water at the time it is loaded for transport. The only moisture present will be the relative humidity or moisture absorbed by the insulation (10 weight percent maximum). Therefore, during Normal Conditions of Transport, there will be no significant moisture present to cause corrosion or rusting problems.

During immersion under Hypothetical Accident Conditions, water can enter the holes at the top of the drum, be absorbed into the insulation, and fill all void spaces within the drum. The insulating value of the insulation material would be lost and the fiber and organic glue composite could become mushy over an extended period of time, but these are not the prime considerations under these conditions. The most important consideration is that the containment vessel remain intact and leaktight, as demonstrated by the analysis and testing.

Since the containment vessel will remain intact even if the drum is filled with water, the package is acceptable to the maximum credible extent from the standpoint of chemical, galvanic, or other reactions.

LIFTING AND TIE-DOWN STANDARDS FOR ALL PACKAGES

This subsection addresses the requirements of 10 CFR 71.45, *Lifting and Tie-Down Standards for All Packages.*

Lifting Devices

The package as delivered for transport has no lifting devices. The tools to be used for lifting the containment vessel lid during loading and unloading operations are not considered a component of the package structure. Therefore, the lifting-device requirements of 10 CFR 71.45 are not germane.

Tie-Down Devices

The package as delivered for transport has no tie-down devices. Therefore, the tie-down requirements of 10 CFR 71.45 are not applicable. During transport, the package is either blocked or braced in a manner conventional for drums or is restrained within a cargo-restraint transporter.

NORMAL CONDITIONS OF TRANSPORT

This subsection demonstrates compliance with 10 CFR 71.71, *Normal Conditions of Transport.* It is shown that this package will experience no loss in shielding effectiveness or spacing and no release of radioactive content or leakage of water into the containment vessel during exposure to the Normal Conditions of Transport.

As specified in 10 CFR 71.71(b) the tests for Normal Conditions of Transport shall be conducted at the most unfavorable ambient temperature within the range of minus 20 to 100 degrees Fahrenheit. The drum and the containment vessel are fabricated from Type 304 stainless steel; which is particularly suitable for low-temperature service. The Izod strength of a material measures it's ability to resist brittle fracture. The Izod impact strength for Type 304 stainless steel remains constant over a large range specifically minus 320 to 70 degrees Fahrenheit.[11] Tests on this package were performed at 70 to 90 degrees Fahrenheit ambient temperature. At these temperatures, the strength properties of the 304 stainless steel will remain constant and the tests will provide essentially the same results as tests at minus 20 degrees Fahrenheit.

As specified in 10 CFR 71.71(c) the package service temperature must be between minus 40 and 100 degrees Fahrenheit. Solar insolation at 100 degrees Fahrenheit must also be considered. With solar insolation, the maximum temperature of the containment vessel approaches 200 degrees Fahrenheit. Thermal cycling of a package over the temperature range of minus minus 40 to 200 degrees Fahrenheit is considered a rare event, multiple thermal cycling of the same package is considered incredible. Thus, in no case will the fatigue endurance limit for Type 304 stainless steel be exceeded from thermal cycling.

Heat

Requirement. Exposure to an ambient temperature of 100 degrees Fahrenheit in still air and insolation as stated in 10 CFR 71.71.

Analysis. An increase in ambient temperature to 100 degrees Fahrenheit will have no effect on the ability of the containment vessel to provide containment.

Carbon steel confinement boundaries (drums) of similar construction have been qualified for this condition by test.[12] This drum is stainless steel, but service characteristics are essentially the same as carbon steel at this temperature.

The service temperature range of the O-rings is from minus 40 to 250 degrees Fahrenheit. The vendor certified continuous service life at 250 degrees Fahrenheit or less is at least 1000 hours[3] If the package is exposed to solar radiation at 100 degrees Fahrenheit still air, the O-ring temperature will be less than 200 degrees Fahrenheit. This temperature represents a conservative upper value for the packaging under normal conditions.

Summary of Pressures and Temperatures

An ambient temperature of 70 degrees Fahrenheit is assumed for the packaging at assembly. Since there are four ventilation holes near the top of the drum, the drum will not become pressurized as the temperature increases. The containment vessel is sealed; thus, the internal pressure will change with temperature. Maximum calculated pressures at various temperatures are tabulated below:

External temperature (ºF)	Internal pressure (psia)
-40	11.65
70	14.7
100	15.5
150	16.9
200	18.31

Differential Thermal Expansion

Differential thermal expansion during Normal Conditions of Transport is insignificant.

Stress Calculations

Stress levels imposed on the package during Normal Conditions of Transport are insignificant as demonstrated in Appendix B. Fatigue failure of the containment vessel could only come about by stress levels exceeding the endurance limits. During normal conditions, stresses are only imposed by changes in internal pressure of the containment vessel as the temperature varies over the operating range of -40 to 200 degrees Fahrenheit.

The hoop stress in the wall of the containment vessel over the operating temperature range varies from compression to tension as shown below:

pressure (@ -40^{0}F) = 11.65 psia -3.05 psig

pressure (@ 200^{0}F) = 18.31 psia = 3.61 psig

$$\text{hoop stress} = \frac{PD}{2t},$$

where:

P = pressure (psig),

d = 5.047 in.,

t = 0.258 in,

$$\text{hoop stress @ } -40^\circ\text{F} = \frac{(-3.05)(5.047)}{2(0.258)} = -30 \text{ psi, (compression)}$$

$$\text{hoop stress @ } 200^{\circ}\text{F} = \frac{(3.61)(5.047)}{2(0.258)} = 35 \text{ psi. (tension)}$$

The variation in hoop stress in the wall of the containment vessel over the operating temperature range will occur slowly over time as temperature changes are not expected to be rapid during Normal Conditions of Transport. Since the magnitude and rate of change of the hoop stress are small during Normal Conditions of Transport, no compromise of the package integrity is expected.

Comparison with Allowable Stresses

The fatigue or endurance limits for austenitic stainless steel are normally assumed to be about 1/2 the ultimate tensile strength (page 5-10[10]). For Type 304 stainless steel, one-half the ultimate tensile strength is 42,500 pounds per square inch. The tensile and compressive hoop stresses of the magnitude shown above are insignificant when compared with the endurance limit of 42,500 pounds per square inch.

As noted in the previous subsection, the hoop stresses during Normal Conditions of Transport are insignificant. Even at the maximum test temperature and internal pressure, the stresses in the containment vessel were insignificant when compared with the allowable wall stress of 14,300 pounds per square inch.

Cold

Requirement. An ambient temperature of minus 40 degrees Fahrenheit in still air and shade, as required by 10 CFR 71.71(c)(2).

Analysis. The drum and containment vessel are fabricated from Type 304 stainless steel. Carbon steel drums of similar construction have been qualified for a low temperature of minus 40 degrees Fahrenheit[12] However, as discussed earlier, stainless steel is more suitable for low-temperature service than carbon steel, particularly regarding impact strength. Stainless steel does not show a transition from ductile to brittle failure. Section VIII of the ASME Code,[2] Part UHA-51 exempts impact testing of Type 304 stainless steel for applications above minus 425 degrees Fahrenheit. The Izod impact strength for Type 304 stainless steel remains constant at 110 feet-pounds from minus 320 to 70 degrees Fahrenheit (page 66[10]). The tensile strength increases from about 85,000 to 246,000 pounds per square inch between 70 and minus 320 degrees Fahrenheit, and the yield strength increases about 10 percent over the same range. Therefore, the stainless steel drum is more acceptable than similar carbon steel drums which have been qualified for minus 40 degrees Fahrenheit.

The thermal insulation is an industrial cane fiberboard made from sugar cane fibers. The fibers are interfelted and bonded in random fashion to provide the specified thickness. The board is about 97 percent cellulose and contains a maximum of 10 weight percent moisture. The only moisture available for freezing at minus 40 degrees Fahrenheit would be moisture that collected in the package during assembly. Since there will be no free water present for freezing and since the insulation is a bonded mass of random fibers, the properties of the insulation will not change appreciably at low temperatures. The fibers may become less flexible when subjected to the cold temperature, but this will not affect the ability of the insulation to position the containment vessel and absorb impacts.

The bolts will be manufactured from an alloyed steel for low-temperature service as specified by ASME A193.[8]

The specified O-rings have a service temperature range of minus 45 to 250 degrees Fahrenheit per the *Parker O-ring Handbook*.[3]

Appendix E provides calculations to demonstrate that the O-rings do not unseat due to differential thermal expansion at minus 40 degrees Fahrenheit.

As demonstrated by the information presented above and in Appendix E, the packaging is acceptable for Normal Conditions of Transport at minus 40 degrees Fahrenheit.

Reduced External Pressure

Requirement. An external pressure of 3.5 pounds per square inch, absolute, as required by 10 CER 71.71(c)(3).

Analysis. Reducing the external pressure from ambient pressure to 3.5 will have no effect on the drum because the plugs sealing the ventilation holes will fail, allowing the internal pressure of the drum to equalize. At this reduced pressure, the differential pressure across the wall of the containment vessel would be 11.2 pounds per square inch. The containment vessel is designed and fabricated in accordance with Section VIII of the ASME Boiler and Pressure Vessel Code for a pressure differential of 70 pounds per square inch (Appendix B). Therefore, the packaging is acceptable for Normal Conditions of Transport at an external pressure of 3.5 pounds per square inch, absolute.

Increased External Pressure

Requirement. An external pressure of 20 pounds per square inch, absolute is required by 10 CFR 71.71(c)(4).

Analysis. Increasing the external pressure from ambient pressure to 20 pounds per square inch, absolute would have no effect on the drum because the plugs covering the ventilation holes would fail, allowing the internal pressure of the drum to equalize. The containment vessel is designed and fabricated in accordance with Section VIII of the ASME Boiler and Pressure Vessel Code for an external pressure of 85.7 pounds per square inch, absolute at 70 degrees Fahrenheit (Appendix B). Therefore, the packaging is acceptable for Normal Conditions of Transport at an external pressure of 20 pounds per squar einch, absolute.

Vibration

Requirement. Vibration normally incident to transportation, as required by 10 CFR 71 .71(c)(5).

Analysis. An analysis has been completed for normal vibration while being subjected to transport loads in a Department of Energy safe-secure trailer or any other transport (Appendix C). The analysis shows that a maximum acceleration level of 1.2 g is expected during shipment. Since this level is only slightly higher than normal gravity, it is not considered detrimental to the packaging. Conventional shipping over roadways has shown that transportable goods withstand these types of vibratory loadings without adverse effects, even is situations of much less packaging integrity than this package.

Procedures will be followed to assure that the packaging is assembled as specified. The drum, lid, and lock ring are refurbished as required before each use. The lock ring bolt is tightened to 50 feet-pounds (nominal torque) and secured with a lock nut. The 3/4-inch containment vessel head bolts are tightened to 30 feet-pounds (nominal torque).

Calculations are provided in Appendix F to demonstrate that the endurance limit of the bolts of the containment vessel far exceeds the maximum stress during transport.

The package is acceptable for vibration normally incident to transport in a safe-secure trailer.

Water Spray

Requirement. A water spray that simulates exposure to rainfall of approximately 2 inches/hour for at least 1 hour, as required by 10 CFR 71.71(c)(6).

Analysis. The neoprene gasket on the lid of the drum and the four ventilation hole plugs should maintain a water tight perimeter. Even if prolonged exposure to rain did result in water penetrating the drum, the containment vessel would remain intact and leak tight; thus, the package is acceptable for use under the water spray conditions of the Normal Conditions of Transport.

Free Drop

Requirement. A free drop of 4 foot onto a flat, essentially unyielding, horizontal surface in a position for which maximum damage is expected, as required by 10 CFR 71.7 1(c)(7). For Fissile Class II packages, this test shall be preceded by a free drop from a height of 1 foot onto each quarter of each rim. These tests shall be made between 1 1/2 and 2 1/2 hours after the conclusion of the water spray test.

Analysis. Drop tests must be conducted on a prototype package with simulated contents and a greater weight than the proposed actual contents. Minimal damage must be seen following these tests and no breaks in the outer surface of the drum. On this basis, the package meets the requirements of 10 CFR 71.71(c)(7).

Corner Drop

Requirement. For wood and fiberboard packages, 10 CFR 71.7 l(c)(8) requires a free drop onto each corner of the package in succession, or in the case of a cylindrical package, onto each quarter of each rim, from a height of 1 foot onto a flat, essentially unyielding, horizontal surface. Preceding the 4-foot free drop, a drop onto each quarter of each rim from a height of 1 foot onto a flat, essentially unyielding, horizontal surface.

Analysis. This test is not applicable because the confinement boundary is a metal drum. This test applies only to wood and fiberboard packages.

Compression

Requirement. A compression load of five times the package gross weight or the equivalent of 1.85 pounds per square inch multiplied by the vertically projected area of the package, whichever is greater. This load shall be uniformly applied to the top and bottom surfaces of the package. The load must be applied for 24 hours, as required by 10 CFR 71.71(c)(9).

Analysis. This test must be conducted on a prototype package. The minimum requirement for this package is about 1250 pounds. On the successful completion of this test, the package meets the requirements of 10 CFR 71.71(c)(9).

Penetration

Requirement. Impact of the hemispherical end of a vertical steel cylinder of 1 1/4 inches diameter and 13-pounds mass dropped from a height of 40 inches onto the exposed package surface that is expected to be most vulnerable to puncture, as required by 10 CFR 71.71(c)(10).

Analysis. This test must be performed on a prototype package. On the successful completion of this test, the package meets the requirements of 10 CFR 71 (c)(10).

HYPOTHETICAL ACCIDENT CONDITIONS

This subsection demonstrates compliance with 10 CFR 71.73, *Hypothetical Accident Conditions.* It shows that the package will experience no loss in shielding effectiveness or spacing and no release of radioactive content or leakage of water into the containment vessel during Hypothetical Accident Conditions.

Title 10 CFR 71.51 requires that the shipping package satisfy the standards under Hypothetical Accident Conditions specified in 10 CFR 71.73. For the tests specified in 10 CFR 71.73, three test packages are to be subjected to the three different tests: free drop, puncture, and thermal. In addition, another undamaged package undergoes the general immersion test required by 10 CFR 71.73(c)(5).

Title 10 CFR 71.73(b) requires that the Hypothetical Accident Conditions tests, except for the water immersion tests, be conducted at the most unfavorable ambient temperature within the range of minus 20 to 100 degrees Fahrenheit. This requirement was previously discussed for Normal Conditions of Transport in which it was concluded that the tests performed at 70 to 90 degrees Fahrenheit ambient temperatures should provide essentially the same results as those made at any ambient temperature between minus 20 and 100 degrees Fahrenheit.

Free Drop

A free drop of 30 foot onto a flat, unyielding, horizontal surface, striking the surface in a position for which maximum damage is expected, as required by 10 CFR 71.73(c)(l).

Analysis

A hypothetical free drop was analyzed. The analysis simulated a fully-loaded shipping package, weighing approximately 260 pounds and dropped onto it's corner from a height of 30 foot onto an unyielding surface. The package was aligned so that the center of gravity was over the impact point.

In the simulation, at the moment immediately before impact, all the potential energy of the package was converted to kinetic energy. The maximum relative velocity between the package and the impact surface was 527.4 inches per second. This relative velocity was broken into two orthogonal components; one velocity component parallel to the longitudinal axis of the package, and the other velocity component perpendicular to the longitudinal axis. The lid of the package was pointed downward so that the bolts holding the lid to the containment vessel were loaded in tension. This geometry is considered as the most conservative impact case and would result in maximum loads on the containment vessel. Therefore only this one impact case was analyzed.

The outer drum of the package is made of stainless steel sheet (16 gauge). During impact, the drum will permanently deform but very little energy will be dissipated through this plastic deformation of the drum. Therefore, the drum is conservatively assumed to remain elastic during impact. The fiberboard and the containment vessel are allowed to deform beyond the elastic limits. However, the calculated stresses in the fiberboard and containment vessel are well below the elastic limit. This analysis is thus conservatively simplified to linear dynamic impact analysis.

A finite-element analysis was performed. Since the package is symmetrical about the longitudinal axis, only half of the package was modeled.

The solution to the dynamic impact analysis involved equating the kinetic energy of the moving object to strain energy of the object undergoing stress deformation. Since the ground surface is an unyielding surface, it is assumed that all the strain energy went into the package. During impact, multiple shock strain waves were generated and propagated within the object. All the strain waves were dissipated when permanent deformation took place. The energy dissipated in the form of permanent deformation is equivalent to the total kinetic energy before impact and can be found by a static analysis. However, the maximum transient dynamic stresses induced by the propagating shock wave may be many times greater than the stress found by simple static analysis. Therefore, a computer program that could account for the dynamic state of the stress was used. The finite-element program ADINA (Automatic Dynamic Incremental Nonlinear Analysis)[13] was used to perform this analysis. The program ADINA is a main frame program which is licensed and maintained by the code developer. The outer drum, fiberboard, containment vessel and aluminum plate as modeled are shown in Appendix D.

The maximum stress of the vessel during impact was determined to be 400 pounds per inch. This is well within the allowable stress limits, which are 14,300 pounds per inch for the top flange area. The maximum acceleration of the drum immediately before impact was determined to be 461 g's. The maximum acceleration of the containment vessel was determined to be 270 g's. A discussion of the analysis and the results are presented in Appendix D.

Prototype Testing

Three shipping packages containing a mock-up of the maximum weight content shall be drop tested from 30 foot in accordance with 10 CFR 71.73(c)(l).

The first test package weighing about 264 pounds is to be tilted at about 29 degrees from vertical and dropped with the lid down onto the juncture of the lock ring and the drum sidewall seam. The line of impact is to be through the approximate center of gravity, thus providing the maximum corner impact load. One concern for this impact orientation is that the drum and insulation would compress near the top corner of the containment vessel resulting in less insulation of the O-ring seal area during a fire. A second concern is that the drum lid might be partially separated from the drum body, directly exposing the insulation; then, in subsequent thermal tests, the Buna-N O-rings between the flanges of the containment vessel could be damaged. The use of this drop position was also based on actual testing of prototype carbon steel containers with locking rings for transport of Type B packages,[6] in which 20 Type B prototype containers were tested. The carbon steel drum sizes varied from 30 to 140 gallons, with a weight range of 130 to 880 pounds.

The test plan for the second test package, weighing about 264 pounds, is to be a straight drop onto the bottom of the drum. An end drop, either top or bottom, could result in the highest overall deceleration and the most abrupt shock.

The third test package is to be dropped from 30 feet onto its side. The side drop orientation could be less severe than either a top or bottom-end drop since the side drop presents a larger area at impact, thus distributing the available kinetic energy.

Summary of Results

The weights of the test packages were about 264 pounds. The tests shall be conducted at the maximum package content weight to determine the maximum potential damage. Since the velocity of a package at impact is independent of the package, the ratio of kinetic energies is directly proportional to the weight.

The test configuration is applicable to the actual configuration for several reasons. The load path of the test mass is equivalent to the actual mass since they are both very similar in configuration and consistency. Applicability is further substantiated by the fact that the kinetic energy in an actual configuration will be less than the test configuration giving conservatism to the test configuration.

The containment vessel must not be damaged in any of the drop tests. The deformation of the drum in the three tests must not decrease the effective center-to-center spacing required for criticality safety purposes. The model used for the criticality analyses shall be shown to be adequate for the damage incurred during package testing, and shows the shipping package is safe during Hypothetical Accident Conditions. No brittle fracturing should be expected or observed.

Puncture

A free drop of 40 inches, from a position to obtain maximum damage, onto the upper end of a solid, vertical, cylindrical, 6 inch-diameter, mild steel bar mounted on an unyielding horizontal surface, as required by 10 CFR 71.73(c)(2). The bar must be a minimum of 8 inches long with the top end rounded to 1/4 inch maximum radius.

Analysis

The three test units previously dropped from 30 feet are to be dropped from 40 inches in accordance with 10 CFR 71.73(c)(2). They are to be dropped in three different orientations as described in the test plan.

Prototype Testing

The three puncture tests described above are to be completed after the 30 foot drop tests using the same three test units.

Summary of Results

The results of the prototype tests as described above are to be summarized.

Thermal

Exposure of the whole specimen for not less than 30 minutes to a heat flux not less than that of a radiation environment of 1475 degrees Fahrenheit, with an emissivity coefficient of 0.9 minimum and no artificial cooling, as required by 10 CFR 71.73(c)(3).

Analysis

A computer analysis of the package subjected to hypothetical accident fire conditions is to be conducted. The analytical model is to be discussed in the thermal chapter (Chap. 3).

Prototype Testing

A full-scale thermal test is to be conducted on one package containing a 20 kilograms mock-up. The test is to be performed following the free-drop and puncture tests in accordance with 10 CFR 71.73(c)(3). The test package to be used for the thermal test is that which experienced the most damage during the drop and puncture tests. Prior to the thermal test, the test package is preheated to simulate a 5 Watts content heat load and a 100 degrees Fahrenheit ambient temperature. The test package is to be placed, upright, in a jet fuel pool fire for 30 minutes. The test unit is to be instrumented with temperature indicators placed on the surfaces of the containment vessel.

Summary of Results

A maximum temperature is to be recorded. A leak test of the containment vessel after cool down from the thermal test is also to be completed. The containment vessel surfaces, flange faces, O-rings, and sealing surfaces are to be shown by the air leak test as not damaged. If the package is completely intact following the 30 minute exposure to the pool fire, the requirements of 10 CFR 71.73(c)(3) have been met.

The O-rings have a service temperature range of minus 45 to 250 degrees Fahrenheit in accordance with Parker.[3] The manufacturer conservatively based O-ring temperature ratings on 1000 hours of continuous life at an operating temperature of 250 degrees Fahrenheit. The manufacturer further states (Fig. A3-6[3]) that the O-rings can operate at a temperature as high as 350 degrees Fahrenheit for about 5 hours. No damage would be expected to the O-rings from the tests.

Summary of Pressures and Temperatures

An ambient temperature of 70 degrees Fahrenheit is assumed for the packaging at assembly. Since there are four ventilation holes near the top of the drum, the drum will not be pressurized as the temperature increases. The containment vessel is sealed and the internal pressure will increase with temperature. The calculated pressures at the maximum test temperatures are tabulated below:

External temperature (°F)	Internal pressure (psia)
70	14.7
150	16.9
200	18.3

Differential Thermal Expansion

The containment vessel is constructed of Type 304 stainless steel. The insulation to containment vessel minimum gap is 0.375 inch radially; therefore, no radial interference is expected from the thermal growth. Interferences in the vertical directions are minimal since there is adequate clearance between the containment vessel and the insulation.

Assuming the pretest temperature of the containment vessel was 70 degrees Fahrenheit and observing the temperature that the entire containment vessel flange reached during the thermal test, the linear expansion of the flange thickness should have been about 0.0013 inch, as shown below:

$$\begin{aligned}\delta &= \alpha\, l\, \Delta T\ , \\ &= 9.8 \times 10^{-6}(1.625)(150 - 70)\ , \\ &= 0.0013 \text{ in.},\end{aligned}$$

where

α = coefficient of linear expansion (in./in/^{o}F),

l = thickness of bolted flange (in.),

ΔT = temperature differential (^{o}F).

If all the flange bolts had reached the same temperature, the expansion would have been about 0.0014 inch, as shown below:

$$\begin{aligned}\delta &= \alpha\, l\, \Delta T\ , \\ &= 7.0 \times 10^{-6}(2.5)(150 - 70)\ , \\ &= 0.0014 \text{ in.}\end{aligned}$$

where:

l = bolt length (in.)

Since the flanges will be seated face to face, the bolt load will not increase as a result of the thermal expansion. According to these calculations the bolts will loosen very slightly, however, since the bolts are tightened to 30 feet-pounds, the slight potential loosening will not be enough to affect the containment integrity of the containment vessel. During Normal Conditions of Transport the temperature of the containment vessel can potentially be higher because of the solar incidence required by 10 CFR 71. Linear expansion during Normal Conditions of Transport would only be slightly more than shown here by these calculations. For normal conditions, the amount of loosening due to the thermal expansion of the containment vessel bolts would not compromise the integrity of the containment.

Stress Calculations

The principal effect of the elevated temperature on stress levels is caused by the increase in the internal pressure. An analysis of this effect is presented in the thermal chapter (Chap. 3). The hoop stresses in the wall are to be well below the allowable stress of the containment vessel walls which is 14,300 pounds per square inch.

The containment vessel bolts will be subjected to stress as a result of the torque, increased internal pressure, and thermal expansion. Calculations are provided in Appendix F to demonstrate that the stress in the bolts from these sources do not exceed the yield strength of 30,000 pounds per square inch.

During the pool fire test containment vessel temperatures are to be recorded on the top flange. Temperatures are also to be recorded on the bottom head. If low temperatures and low temperature gradients are present, the thermally induced stresses in the stainless steel containment vessel are minor and no damage would be expected.

The air leak check on the containment vessel is to be recorded following the thermal test. This leak rate is used to demonstrate that the thermally induced stress, if any, cause no permanent deformation and will not affect the containment capability. Therefore, the reliability of the containment vessel to perform its containment function is demonstrated.

Comparison with Allowable Stresses

If the differential stresses resulting from temperatures recorded during Hypothetical Accident Conditions are negligible, the tangential stress during Hypothetical Accident Conditions is minor when compared to the allowable stress limit of 14,300 pounds per square inch for the wall of the containment vessel. If these stresses are low they do not affect the integrity of the packaging.

Immersion—Fissile Material

Requirement. In those cases for which water leakage into the containment vessel has not been assumed for criticality analysis, the specimen must be immersed under a 3 feet head of water for a period of not less than 8 hours and in an attitude for which maximum leakage is expected, as required by 10 CFR 71.73(c)(4).

Analysis. If the criticality chapter assumes leakage of water into the containment vessel and complete flooding of all void volume, this test is not considered to be necessary.

Immersion—All Packages

Requirement. A separate, undamaged specimen must be immersed under water at a pressure equivalent to a 50 feet head of water for not less than 8 hours, as required by 10 CFR 71 .73(c)(5). This requirement may be satisfied by an external pressure of 21 pounds per square inch (gauge).

Analysis. Immersion under water with pressure equivalent to a 50 feet head of water would result in water entering the drum, because the caplugs covering the four ventilation holes would not prevent water from entering the drum. The containment vessel has been designed for an external pressure of 21 pounds per square inch, gauge and an internal pressure of 70 pounds per square inch, gauge. The design incorporates an O-ring seal of verified integrity to provide assurance that no water will penetrate the containment boundary.

An undamaged shipping package shall be subjected to water at a 50 feet head of water for at least 8 hours. Inspection of the package after the immersion test should show no signs of water leakage into the containment vessel.

Summary of Damage

As a result of the testing of the shipping packages under Hypothetical Accident Conditions, no drum and insulation damage is expected. The resultant damage shall not reduce the effective center-to-center spacing to a point of criticality concern. As required in 10 CFR 71.73(c)(4) the containment vessel must be submerged under a 3 feet head of water, following the thermal test, for at least 8 hours with no leakage permitted. The container does not have to be subjected to this test if the criticality analysis demonstrates that the package can be full of water with no adverse effects.

The structural integrity of the shipping package must be demonstrated by calculation and full-scale testing to meet all the applicable requirements of 10 CFR 71.73 for transport.

SPECIAL FORM

The package does not include special form radioactive material. Hence, the requirements of 10 CER 71.75 and 71.77 are not germane.

FUEL RODS

The contents do not utilize cladding for the containment of radioactive materials. Therefore, the requirement is not germane.

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APPENDIX B

DESIGN CALCULATIONS FOR CONTAINMENT VESSEL

The design of the containment vessel has been reviewed with respect to the requirements of ASME Rules for Construction of Pressure Vessels Section VIII, Division 1, 1992 Edition.[2] The calculations here will show that design meets the stress requirements of the ASME Code.

Applicable Code Sections:

These calculations reflect the following ASME Code sections:

- UG-23 Maximum Allowable Stress Values;
- UG-27 Thickness of Shells Under Internal Pressure;
- UG-28 Thickness of Shell and Tubes Under External Pressure;
- UG-32 Formed Heads, Pressure on Concave Side; (formed heads are according to ASA B16.9);
- UG-33 Formed Heads, Pressure on Convex Side; (formed heads are according to ASA B16.9);
- UG-34 Unstayed Flat Heads and Covers;
- UG-43 Methods of Attachment of Pipe and Nozzle Necks to Vessel Wall (thread engagement requirements have been met);

- UW-12 Joint Efficiencies (full radiographic examiniations of welds are required);

- Appendix 2, Rules for Bolted Flange Connections with Ring Type Gaskets (Sections: 2-3, 2-5, 2-6, 2-7, and 2-11); and

- Appendix 5, Charts for Determining Shell Thickness of Cylindrical and Spherical Vessels Under External Pressure.

Pressure relief device requirements (UG-125) do not apply since this container is used under ambient temperature conditions. Because the containment vessel is installed in an insulated outer container, an additional hazard such as exposure to fire will not cause the internal pressure to exceed 5 pounds per square inch, gauge, which is only 7 percent of the 71 pounds per square inch, gauge design pressure.

Design Conditions and Allowable Stresses:

The worst case accident scenario is a 50 feet head of water which translates to an external pressure of 21 pounds per square inch, gauge; thus this will be the design external pressure. This vessel will also be designed for 71 pounds per square inch, gauge internal pressure and hydrostatically tested at 105 pounds per square inch, gauge as per ASME Boiler and Pressure Vessel Code, Section VIII, Division 1. This analysis assumed that the vessel body is 304L stainless steel and the lid is Type 304 stainless steel.

Allowable stresses for this vessel are from the ASME Code, Section VIII, Division 1 (1992 Edition). The temperature range of minus 20 to 200 degrees Fahenheit (See ASME Code, Section II, Part D Table UHA-23[14]) were applied. The stresses are as follows:

SA-312 seamless pipe, TP304L stainless steel (wall)	S_p = 14,300 psi
SA-182 forging, F304L stainless steel (flange)	S_f = 14,300 psi
SA-193 bolts, Grade B8C, Class 1	S_b = 17,900 psi
SA-182 forging, F304 stainless steel (lid)	S_h = 16,600 psi
SA-403 fitting, WP304L stainless steel (cap)	S_c = 13,300 psi
SA-194 nuts, Grade 8C	S_n = 17,900 psi

Flathead—Cover: Section UG-34[2]

Lid OD = 10 in.

S_h = 16,600 psi (allowable flathead stress at 200°F)

S_b = 17,900 psi (allowable bolt stress at 200°F)

d_m = 6.342 in. (mean gasket diameter)

t_{min} = 0.813 in. (minimum flathead thickness)

P = 71 psig (internal design pressure)

P_e = 21 psig (external design pressure)

C = 0.3 (from Fig. UG-34[2])

h_G = 1.079 in. (see calculations on following page)

Note: The dimensional nomenclature for the Containment Vessel flange is shown in Fig. B1.

+

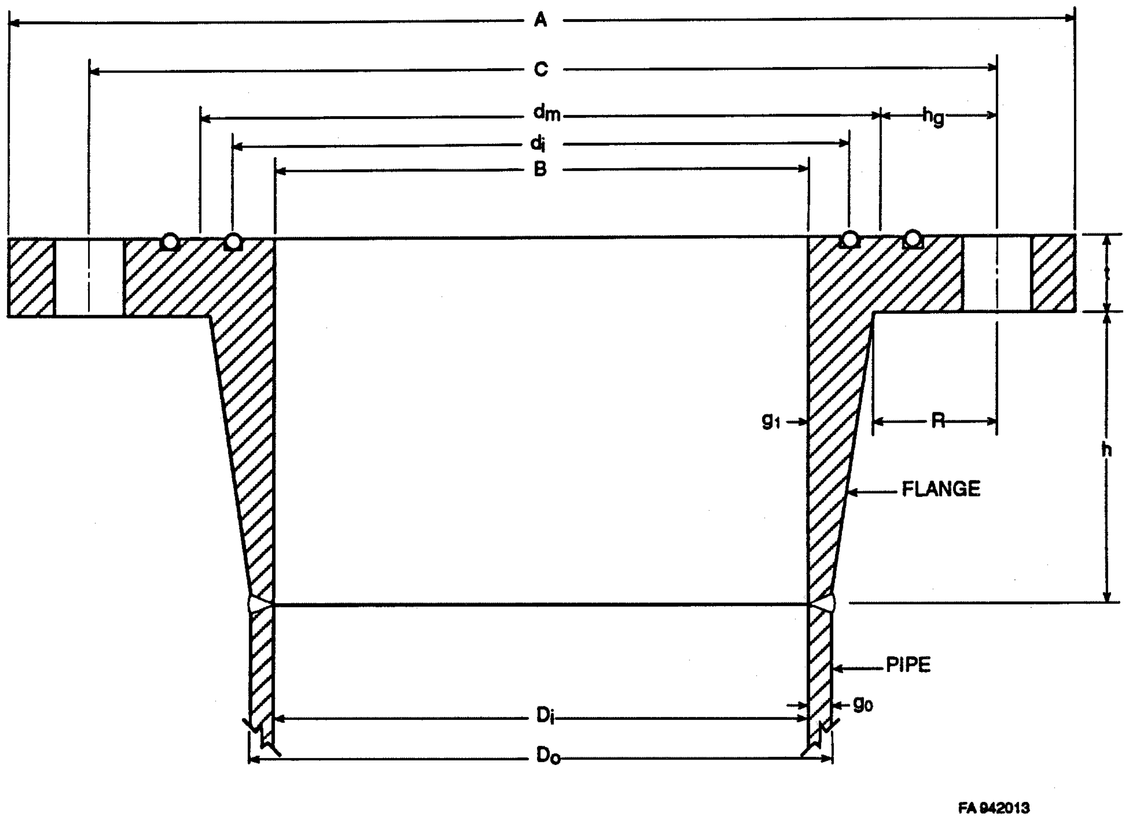

Fig. B1. Dimensional nomenclature for the containment vessel flange

8 bolts, 3/4 inch-10 UNC-2A with 0.3340 square inch stress area per bolt (page 8-10[11]), root diameter area for all 8 bolts,

$$A_b = 8(0.3340) = 2.672 \text{ in.}^2$$

Gasket loading is 75 pounds per inch of circumference per O-ring or 150 pounds per inch on a mean double O-ring diameter of 6.342 inches (a constant from Fig. A4-14 of the Parker O-ring Handbook, 30 percent compression, 70 plus or minus 5 Shore hardness, 0.139 inch cross-sectional diameter[3]). The gasket seating load, W_{m2}, Section 2-5(c)(2),[2] equals the circumference of the mean O-ring seal diameter times 150 pounds per inch.

$$W_{m2} = \pi (G)(b)(y)$$

$$W_{m2} = 3.14(6.342)(150) ,$$

$$= 2987 \text{ lb (gasket bolt load)} ,$$

$$G = d_m = 6.342 \text{ in. mean gasket diameter} ,$$

$(b)(y)$ = 150 lb/in. (compression per linear inch for 0.139 in. cross-sectional diameter O-rings[3])

$$A_{m2} = W_{m2}/S_b \qquad S_a = S_b$$

$$= \frac{2987}{17{,}900}$$

$$= 0.167 \text{ in.}^2$$

$$W = (A_{m2} + A_b)(S_b)/2 \text{ (gasket seating load)}$$

$$= (0.167 + 2.672)(17{,}900)/2$$

$$= 25{,}409 \text{ lb.}$$

Flange subject to internal pressure,

$$W_{m1} = 0.785\,(G)^2\,P + W_{m2}\ ,$$

$$= 0.785\,(6.342)^2\,(71) + 2987\ ,$$

$$= 5{,}229 \text{ lb (operating bolt load)}\ ,$$

$$G = d_m = 6.342 \text{ in. mean O-ring diameter,}$$

$$A_{m1} = W_{m1}/S_b\ , \qquad S_a = S_b\ .$$

$$= 5{,}229/17{,}900\ ,$$

$$= 0.292 \text{ in.}^2$$

Flange design bolt load, Section 2-5(e)[2]

$$W = (A_{m1} + A_b)(S_b)/2\ , \text{ (operating load)}$$

$$= (0.292 + 2.672)(17{,}900)/2\ ,$$

$$= 26{,}528 \text{ lb}\ .$$

Therefore, use W = 26,528 pounds since operating load exceeds gasket seating load.

The minimum required flathead cover thickness at 70 pounds per square inch, gauge internal pressure:

$$t = d_m\,[CP/(SE) + 1.9Wh_G/(S_hEd_m^3)]^{0.5}\ ,$$

$$t = 6.342\,[0.3(71)/(16{,}600(1)) + 1.9(26{,}528)(1.079)/(16{,}600(1)6.342^3)]^{0.5},$$

$$= 6.342[0.0013 + 0.0128]^{0.5}\ ,$$

$$= 6.342(0.1187)\ ,$$

$$= 0.753 \text{ in.}$$

The actual minimum flathead cover thickness is 0.813 inch. This thickness is acceptable since it is greater than 0.753 inch.

Moment Arms for Flange Loads: Table 2-6 (Integral, Weld-Neck Flange)[2]

$$h_D = R + 0.5\, g_1 \,,$$
$$= 1.031 + 0.5(0.695) \,,$$
$$= 1.379 \text{ in.}$$

$$R = \frac{C - B}{2} - g_1 \,,$$

$$= \frac{8.5 - (5.047)}{2} - 0.695$$

$$= 1.031 \text{ in.}$$

$$h_G = \frac{C - G}{2} \,,$$

$$= \frac{8.5 - 6.342}{2} \,,$$

$$= 1.079 \text{ in.}$$

$$h_T = \frac{R + g_1 + h_G}{2} \,,$$

$$= \frac{1.031 + 0.695 + 1.079}{2} \,,$$

$$= 1.403 \text{ in.} \,,$$

where:

g_1 = 0.695 in. (hub thickness at back of flange),

C = 8.5 in. (bolt circle diameter),

B = 5.047 in. (inside diameter of flange),

G = 6.342 in. (mean gasket diameter) = d_m

Flange Moment, M_o , Due to Gasket Seating: Section 2-11[2]

$$
\begin{aligned}
M_o &= Wh_G \ , \\
&= 25{,}409(1.079) \ , \\
&= 27{,}416 \text{ in.-lb from gasket seating} \ ,
\end{aligned}
$$

Flange Moment, M_o , Due to Internal Pressure and Gasket Seating: Sections 2-3 and 2-6[2]

$$
\begin{aligned}
M_o &= M_D + M_T + M_G, \\
&= 1958 + 1153 + 26{,}205 \ , \\
&= 29{,}316 \text{ in.-lb} \ .
\end{aligned}
$$

Solving for the components of M_o,

$$
\begin{aligned}
M_D &= H_D \, h_D \ , \\
&= (0.785)\,(5.047)^2\,(71)\,(1.379) \ , \\
&= 1420\,(1.379) \ , \\
&= 1958 \text{ in.-lb} \ . \\
M_T &= H_T \, h_T \ , \\
&= [0.785\,(6.342)^2\,(71) - 1420]\,(1.403) \ , \\
&= (2242 - 1420)1.403 \ , \\
&= 1153 \text{ in.-lb} \ . \\
M_G &= H_G \, h_G \ , \\
&= (W-H)h_G, \qquad \text{(Section 2-3 and 2-5[2])} \\
&= (26{,}528 - 2242)(1.079) \ , \\
&= 24{,}286\,(1.079) \ , \\
&= 26{,}205 \text{ in.-lb} \ .
\end{aligned}
$$

where:

H = Force on head due to internal design pressure of 71 psig.

Flange Moment, M_o Due to External Pressure: Section 2-11[2]

$$H_D = 0.785\ B^2\ P_e\ ,$$

$$= 0.785\ (5.047)^2\ (21)\ ,$$

$$= 420\ \text{lb}\ .$$

$$H = 0.785\ G^2\ P_e\ ,$$

$$= 0.785\ (6.342)^2\ (21)\ ,$$

$$= 663\ \text{lb}\ .$$

$$H_T = H - HD\ ,$$

$$= 663 - 420\ ,$$

$$= 243\ \text{lb}\ .$$

$$M_o = H_D\ (h_D - h_G) + H_T\ (h_T - h_G)\ ,$$

$$= (420)\ (1.379 - 1.079) + (243)\ (1.403 - 1.079)\ ,$$

$$= 126 + 78\ ,$$

$$= 204\ \text{in.-lb}\ ,$$

where:

B = 5.047 in.,

P_e = 21 psi,

G = 6.342 in. (mean gasket diameter) ,

h_D = 1.379 in.,

h_G = 1.079 in.,

h_T = 1.403 in.

The flange moment (M_o) is 29,316 inch-pounds from internal pressure and gasket seating; M_o is 204 inch-pounds from external pressure. Since the worst case flange moment is from the internal pressure and gasket seating, the value for M_o used in the remaining flange calculations will be 29,316 inch-pounds.

Flange Stress: Section 2-7[2]

Longitudinal Hub Stress:

$$S_H = \frac{f\,M_o}{L g_1^2 B},$$

$$= \frac{(1)\,(29{,}316)}{1.076(0.695)^2\,(5.047)},$$

$$= 11{,}176 \text{ psi},$$

where:

A	= 10 in.
M_o	= 29,316 in.-lb,
g_o	= 0.258 in. (thickness of hub at small end),
B	= 5.047 in.,
h	= 2.563 in. (hub length),
g_1	= 0.695 in.,
f	= 1 (Fig. 2-7.6[2]),
h/h_o	= 2.25,
g_1/g_o	= 2.69.
A/B	= 1.981

$$L = (te + 1)/T + t^3/d,$$

$$= (0.806(0.53) + 1)/1.5 + 0.806^3/4.18$$

$$= 0.951 + 0.125$$

$$= 1.076$$

where:

t = 0.813 - 0.007 = 0.806 (minimum flange thickness)

T = 1.5 (Fig. 2.7.1[2]),

Solving for the e factor,

$$e = F/h_o ,$$
$$= 0.60/1.141 ,$$
$$= 0.53 \text{ in.}^{-1} ,$$

where:

$F = 0.60$ (Fig. 2-7.2[2]),

$h_o = [Bg_o]^{0.5} = 1.141$ in.

Solving for the d factor,

$$d = U\, h_o\, g_o^2/V ,$$
$$= (3.3)(1.141)(0.258)^2/0.06 ,$$
$$= 4.18 \text{ in.}^3 ,$$

where:

$U = 3.3$ (Fig. 2-7.1[2]),
$V = 0.06$ (Fig. 2-7.3[2]).

$$K = A/B ,$$
$$= 10.00/5.047 ,$$
$$= 1.981 ,$$

A longitudinal hub stress of 11,176 pounds per square inch is acceptable since it is less than the allowable stress of $1.5 \times 14{,}300 = 21{,}450$ pounds per square inch [Section 2-8(a)(1)[2]].

Radial Flange Stress:

$$S_R = \frac{(1.33\, te + 1)\,(M_o)}{Lt^2B} ,$$

$$= \frac{[1.33(0.806)(0.53) + 1](29{,}316)}{(1.076)\,(0.806)^2\,(5.047)} ,$$

$$= 13{,}031 \text{ psi} .$$

A radial flange stress of 13,031 pounds per square inch is acceptable since it is less than the allowable stress of 14,300 pounds per square inch.

$$\frac{S_H + S_R}{2} = (11{,}176 + 13{,}031)/2 \text{ , [(Section 2-8(a)(4)}^{[2]}] ,$$

$$= 12{,}104 \text{ psi} .$$

This stress value is less than the allowable stress of 14,300 pounds per square inch.

Tangential Flange Stress:

$$S_T = \frac{Y(M_o)}{t^2B} - ZS_R ,$$

$$= \frac{3.0(29{,}316)}{(0.806)^2\,(5.047)} - (1.7)(13{,}031) ,$$

$$= 4671 \text{ psi} ,$$

where:

$Y = 3.0$ (Fig. 2-7.1[2]),
$Z = 1.7$ (Fig. 2-7.1[2]).

This flange stress is acceptable since it is less than the allowable stress of 14,300 pounds per square inch. Also,

$$\frac{S_H + S_T}{2} = [11{,}176 + 4671]/2 , \qquad \text{[(Section 2-8(a)(4))}^{[2]}]$$

$$= 7{,}924 \text{ psi} .$$

This stress value is acceptable since it is less than the allowable stress of 14,300 pounds per square inch.

Cylindrical Shells Internal Pressure: Section UG-27(c)(1)[2]

$$t = \frac{PR}{SE - 0.6P} \quad \text{(minimum allowable thickness)}$$

$$= \frac{71\ (2.524)}{(14{,}300)\ (1) - 0.6\ (71)} ,$$

$$= \frac{179}{14{,}257} ,$$

$$= 0.0126 \text{ in.}$$

where:

P	= 71 psig,
R	= 2.524 in. (nominal internal radius),
$S = S_p$	= 14,300 psi,
E	= 1.0 (seamless).
t	= 0.258 in. (nominal thickness of containment vessel) ,
t	= $0.258 \times 0.875 = 0.226$ in. (containment vessel thickness for calculations) ,

where:

0.875 is mill tolerance allowance.

A thickness of 0.226 inch is acceptable since it is greater than the minimum allowable thickness of 0.012 inch.

The maximum allowable internal pressure, P_a, for 0.226-inch wall thickness:

$$P_a = \frac{SEt}{R + 0.6t} , \qquad (S = S_p)$$

$$= \frac{14{,}300(1.0)(0.226)}{2.524 + 0.6(0.226)} ,$$

$$= \frac{3232}{2.66} ,$$

$$= 1215 \text{ psig} .$$

The design internal pressure of 71 pounds per square inch, gauge is acceptable since it is less than the maximum allowable pressure of 1215 pounds per square inch, gauge.

Cylindrical Shells — External Pressure: Section UG-28(c)[2]

$$P_{max} = \frac{4B}{3(D_o/t)} ,$$

$$= \frac{4\ (10{,}663)}{3(24.6)} ,$$

$$= 578 \text{ psig} ,$$

where:

D_o = 5.563 in. (outside diameter),
L = effective cylinder length (Fig. UG-28[2])

L = Total length - cap thickness - 2/3 h

where:

h = Cap height = 1.48 in.

Solving for L,

L = 21.625 - 0.258 - 2/3 (1.48)
= 20.38 in.

$$\frac{L}{D_o} = \frac{20.38}{5.563} ,$$

$$= 3.663 .$$

$$\frac{D_o}{t} = \frac{5.563}{0.226} ,$$

$$= 24.6 .$$

A = 0.0028 (from Table 5-UG0-28.0, Appendix 5[2]),

B = 10,663 (from Fig. 5-UHA-28.1, Appendix 5[2]).

This cylinder is acceptable since the maximum allowable external pressure is 578 pounds per square inch, gauge, which is greater than the design external pressure of 21 pounds per square inch, gauge.

Cylindrical Shell — Maximum Longitudinal Compressive Stress: Section UG-23(b)[2]

According to Section UG-23(b),[2] the lower of the maximum allowable tensile stress or factor B shall be used as the maximum allowable longitudinal compressive stress. The maximum allowable tensile stress for the pipe is 14,300 pounds per square inch. Factor A is calculated as follows:

Solving for A,

$$A = \frac{0.125}{R_o/t}, \qquad \text{(UG-23[2])}$$

$$= \frac{0.125\ (0.226)}{2.781},$$

$$= 0.010,$$

where:

$R_o = D_o/2.$

From Fig. 5, UHA-28.1 (Appendix 5),[2]

$B = 13{,}000$ psi .

Since B is less than the allowable stress B (13,000 pounds per square inch) will be used as the allowable stress in this case. The actual longitudinal compressive stress (S) due to the external design pressure of 21 pounds per square inch, gauge is:

$$S = \frac{PD_o^2}{(D_o^2 - D_i^2)}, \qquad \text{(p. 504[15])}$$

$$= \frac{21(5.563)^2}{(5.563^2 - 5.047^2)},$$

$$= 119 \text{ psi} .$$

This cylinder is acceptable since the actual stress of 119 pounds per square inch is less than the value of B (13,000 pounds per square inch).

Formed Heads — Pressure Concave Side: Section UG-32[2]

$$t_{min} = \frac{PD}{2SE-0.2P}, \qquad \text{(minimum allowable thickness)}$$

$$= \frac{71(5.047)}{2(13{,}300)(1) - 0.2(71)},$$

$$= 0.0135 \text{ in.}$$

where:

P	= 71 psi,
D	= 5.047 in. (ID),
$S = S_c$	= 13,300 psi,
E	= 1,
t	= 0.226 in. (thickness of lower head multiplied by 0.875)

The thickness of the lower head is acceptable since it is greater than the minimum allowable thickness of 0.0135 inch.

$$P_{max} = \frac{2SEt}{D + 0.2t}, \text{ (maximum allowable internal pressure)}$$

$$= \frac{2(13{,}300)(1)(0.226)}{5.047 + 0.2\,(0.226)},$$

$$= 1181 \text{ psig} .$$

The lower head is acceptable since the design internal pressure of 71 pounds per square inch, gauge is less than the maximum allowable internal pressure of 1181 pounds per square inch, gauge.

Formed Heads — Pressure Convex Side: Section UG-33, UG-28(d)[2]

$$D_o/2h_o = 5.563/2\ (1.48)$$

$$= 1.88$$

$$K_o = 0.85 \text{ from Table UG-33.1}^{[2]}$$

$$A = 0.125/(R_o/t)\ ,$$

$$= 0.125/(4.73/0.226)\ ,$$

$$= 0.006\ ,$$

where:

t = 0.226 in.
R_o = $D_o(K_o)$ = 5.563(0.85) = 4.73 in.
D_o = 5.563 in. (Outside diameter of cap skirt)
h_o = 1.48 in. (cap height)

$$P_{max} = \frac{B}{R_o/t}\ , \text{ (maximum allowable external pressure)}$$

$$= \frac{12{,}500\ (0.226)}{4.73}\ ,$$

$$= 597 \text{ psig}\ ,$$

where:

B = 12,500. (Appendix 5, Fig. 5-UHA-28.1[2])

The lower head is acceptable since the external design pressure of 21 pounds per square inch, gauge is less than the maximum allowable external pressure of 597 pounds per square inch, gauge.

Bolt Stress and Torque: Section 2-5 (e)[2]

Flange design bolt load, W = 26,513 pounds (gasket seating with internal pressure) and

Load per bolt, F = (26,513/8) = 3314 pounds (to achieve allowable bolt stress) for

torque, T $= 0.2(F)(d)$, (Equation 6.16, p. 247[16])

$= 0.2(3314)0.75$,

= 497 in.-lb maximum torque = 41 ft-lb
(max torque to achieve allowable stress)

where:

d = 0.75 in. (bolt diameter).

Bolt Load at 71 pounds per square inch internal pressure:

Load H, due to operating pressure of 71 psig:

$$H = P(\text{area}) ,$$

$$= \frac{71(3.14)(6.342)^2}{4} , \quad (\text{using } d_m)$$

$$= 2{,}242 \text{ lb} .$$

Gasket seat loading = 2987 pounds (see previous calculations).

Total required bolt load at 71 pounds per square inch, gauge internal pressure:

$$H_T = \text{operating pressure load} + \text{gasket seating load} ,$$

$$= 2242 + 2987 ,$$

$$= 5229 \text{ lb} .$$

Torque Required, T_o , 70 pounds per square inch, gauge internal pressure and gasket seating:

Load per bolt, $F_o = H_T/8$,

$= 5229/8$,

$$= 654 \text{ lb/bolt} .$$

$$T_o = 0.2(F_o)d , \quad \text{(minimum torque)}$$

$$= 0.2(654)0.75 ,$$

$$= 98.1 \text{ in.-lb (minimum required)} = 8.2 \text{ ft-lb} .$$

Specify a design torque of 30 $\pm$ 2 ft-lb (384 in.-lb). Bolt load at this torque,

$$F = \frac{T}{0.2d} ,$$

$$= \frac{384}{(0.2)(0.75)} ,$$

$$= 2560 \text{ lb/bolt} ,$$

where:

T = 384 in.-lb (max torque),

d = 0.75 in.

Actual bolt stress:

$$S_B = F/A_s ,$$

$$S_B = \frac{2560 + 2242/8}{(0.334)} \quad \text{(torque + pressure loads)}$$

$$= \frac{2840}{(0.334)} ,$$

$$= 8{,}504 \text{ psi} ,$$

where:

A_s = 0.334 in.^{2} (bolt stress area),

d = 0.75 in. (bolt diameter).

The bolts are acceptable since the actual stress of 8,504 pounds per square inch is less than the allowable stress of 17,900 pounds per square inch.

Minimum Thread Engagement, Section UG-43(g)[2]

$$t = 0.75dS_b/S_n \,, \quad \text{(minimum thread engagement)}$$

$$= 0.75(0.75)(17{,}900/17{,}900) \,,$$

$$= 0.56 \text{ in. deep,}$$

where:

d = 0.75 in.,

S_b = 17,900 psi (bolt allowable stress),

S_n = 17,900 psi (nut allowable stress).

This minimum engagement length would apply to tapped holes. However, as specified in Section UG-13, for pressure vessel designs using bolt/nut configurations for pressure head attachment, the minimum engagement length shall be the full depth of the nut.

Actual thread engagement length = t_a

max lid thickness	= 0.8438 in.,
max flange thickness	= 0.8438 in.,
max nut thickness	= 0.665 in.,
bolt length	= 2.50 in.
max washer thickness	= 0.177 in.

$$t_a = 2.50 - (0.8438 + 0.8438 + 0.177)$$

$$= 2.50 - 1.86$$

$$= 0.64 \text{ in.}$$

The minimum thread engagement is 0.665 inch, which is the largest thickness of a 3/4 inch-10UNC hex nut. From the above calculation, 0.64 inch of bolt is actually available for engagement. When fully engaged, the bolt will extend through 96 percent of the nut. This engagement is considered acceptable since this analysis used the maximum of all component tolerances.

Bolt Stress Due to Increased Internal Pressure - Actual operating pressure

This calculation determines the total stress in the containment vessel bolts at the maximum possible thermal condition of about 200 degrees Fahrenheit. The maximum internal operating pressure at 200 degrees Fahrenheit is calculated as follows:

$$P_1V_1/T_1 = P_2V_2/T_2 ,$$

where:

T_1 & T_2 are temperature in Rankine

$V_1 = V_2$,

$P_1 = 14.7$ psia ,

$T_1 = 70°F + 459.6 = 529.6R$.

At the elevated temperature,

$T_2 = 200°F + 459.6 = 659.6$,

$P_2 = (14.7)(659.6)/529.6$,

$P_2 = 18.31$ psi .

$\Delta P = P_2 - P_1 = 3.61$ psi (internal pressure rise).

Taking a conservative approach and assuming that the containment vessel lid is infinitely stiff, all the load due to the internal pressure differential will cause increased bolt loading.

Force due to pressure is calculated as follows:

$$F_P = \Delta P(\text{Area}) ,$$

where:

Area of O-ring with a mean diameter of 6.342 in. ,

$$\text{Area} = \pi D^2/4 = (3.14)(6.342)^2/4 ,$$

$$= 31.57 \text{ in.}^2$$

$$F_P = (3.61)(31.57) ,$$

$$F_P = 114 \text{ lb} .$$

$$\text{Load/bolt} = 114/8 = 14 \text{ lb per bolt} .$$

The stress on the bolt due to increased pressure is therefore,

$$S_{B\Delta P} = 14/0.334 = 42 \text{ psi at the threads,}$$

Bolt stress due to differential thermal expansion is,

$$S_{B\Delta T} = E\Delta\alpha(\Delta T) = (30 \times 10^6)(\alpha_F - \alpha_B)(130) ,$$

$$S_{B\Delta T} = 10{,}920 \text{ psi} .$$

where:

$$\alpha_F = 9.8 \times 10^{-6} \text{ in/in/°F (For flange, Table A4)},$$

$$\alpha_B = 7.0 \times 10^{-6} \text{ in/in/°F (For bolt, Table A6)},$$

$$E_B = E_F = 30 \times 10^6 \text{ psi (Table A6)}.$$

Bolt stress due to torque requirements is,

$$S_{BT} = 8{,}504 \text{ psi (previous section)}$$

Total stress in the bolt at 200 degrees Fahrenheit is then,

$$S_{Btotal} = S_{B\Delta T} + S_{BT} + S_{B\Delta P},$$

$$S_{Btotal} = 10{,}920 + 8{,}504 + 42$$

$$S_{Btotal} = 19{,}466 \text{ psi}.$$

This combined loading will produce a stress in the bolts which exceeds the maximum allowable stress of 17,900 pounds per square inch. However, since this is an extremely infrequent condition, the S_{Btotal} can be realistically compared to the yield strength of the bolts which is 30,000 pounds per square inch (Table A6). Therefore, containment and safety will not be compromised by this condition.

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APPENDIX C

CALCULATIONS FOR NORMAL VIBRATION DURING TRAILER SHIPMENT OF THE PACKAGE

The package is delivered for transport as an assembled unit. Each package must be self-contained as far as vibration protection is concerned. Consequently, the analysis for normal vibration during shipment in a standard or safe-secure trailer must also consider individual packages.

Having the packaging mass, insulation thickness, bearing area, insulation stiffness, and vibration characteristics of the trailer (safe-secure trailer used in this analysis), the mean square response of the container and its contents can be determined.

The power spectral densities (PSD) for the safe-secure trailer are shown at various frequencies (Fig. Cl). The parent curves for this figure are given in the Sandia Report SAND83-0480.[17] From this report, it can be seen that the vertical PSD dominates the horizontal PSD.

The containment vessel is supported by about 4 inches of insulation below it. The containment vessel lower bearing area is 19.6 square inches (5-inches diameter). The Young's modulus (E) of the insulation is 500 pounds per square inch, in compression, as measured by The Celotex Corporation.[18]

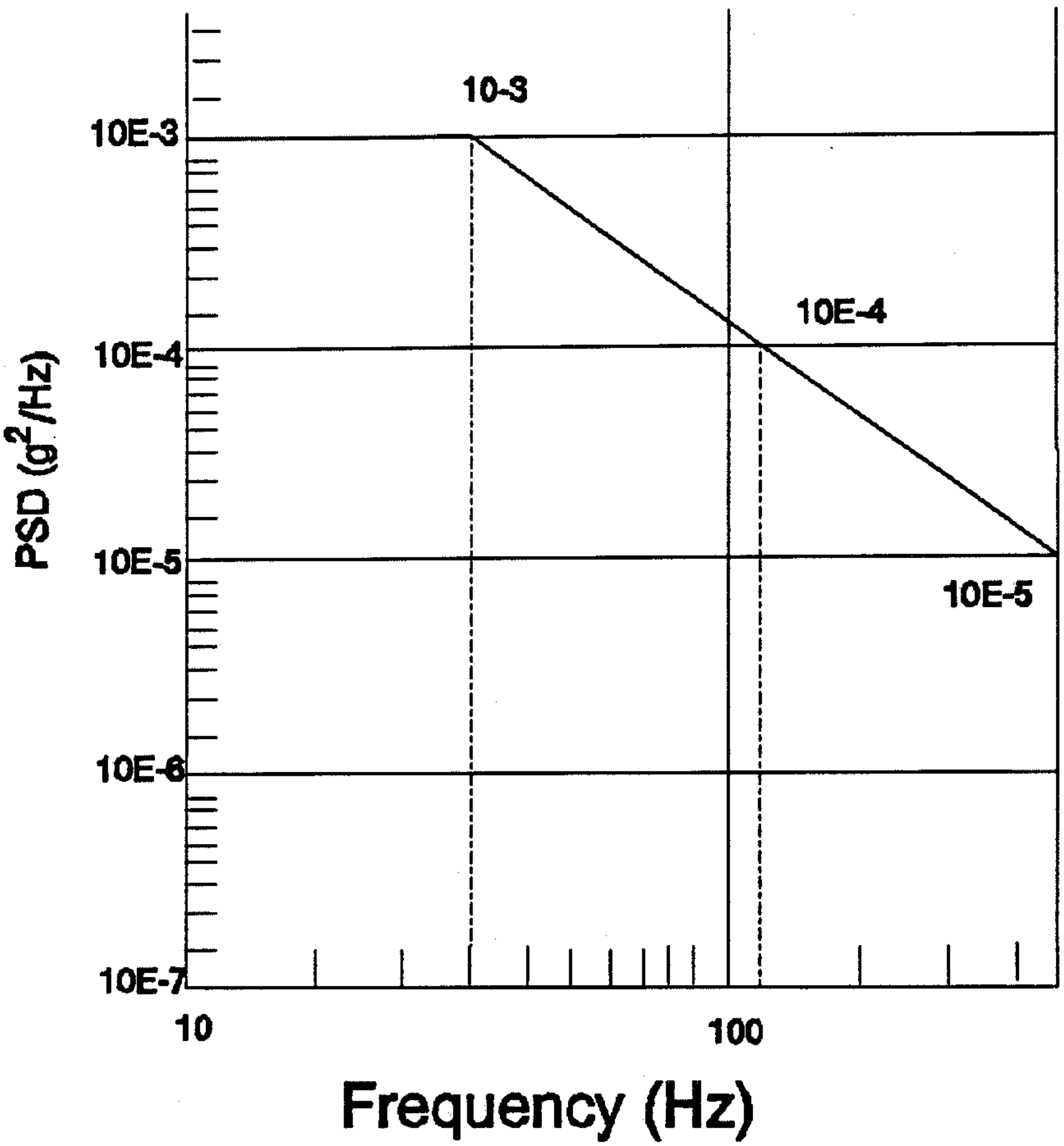

Fig C1. Safe-secure trailer power spectral density.

Excerpted and redrawn for an envelope of 10 to 500 Hz (Fig. 3.30 [17]).

The spring rate of the insulation in the vertical direction can be calculated as follows:

$$k = AE/L\ ,$$

$$= \frac{(19.6)\ (500)}{4}\ ,$$

$$= 2{,}450\ \text{lb/in.}\ ,$$

where:

$A = 19.6\ \text{in.}^2$,

$E = 500$ psi,

$L = 4$ in.

The natural frequency of the container (Hertz) and insulation spring-mass system can then be calculated:

$$f = (1\ /\ 2\pi)*(kg\ /\ W)^{0.5}\ , \qquad \text{(Equation 8-5)[19]}$$

where:

$k = 2{,}450(12) = 29{,}400$ lb/ft,

$g = 32.2\ \text{ft/sec}^2$.

W = Net weight of package

From the shipping weight:

Total shipping weight = 249 lb

Weight of drum = 47.0 lb

Net weight of package (W) = 249 - 47 = 202 lb

Then,

$$f = 1/(2\pi)[29{,}400(32.2)/202]^{0.5} = 11 \text{ Hz} .$$

Conservatively estimating damping as 10 percent of critical, an amplification factor (Q) can be calculated:

$$Q = [(1 - r^2)^2 + (2rd)^2]^{-0.5}, \qquad \text{(Equation 8-14)[19]}$$

$$= [(1 - 1^2)^2 + (2 \times 1 \times 0.1)^2]^{-0.5} ,$$

$$= (0.2^2)^{-0.5} ,$$

$$= 5 ,$$

where:

d = damping coefficient = 0.1,

r = ω/ω_r, where worst case is $\omega = \omega_r$, and $r = 1$.

The packaging is in resonance at 11 Hertz, and vibration above this frequency will be mechanically filtered (isolated). Neglecting any amplification at resonance and using Fig. C1 for the PSD at these frequencies, the broadband, mean square g-level of input excitation on the packaging being transported in a safe-secure trailer can be calculated. The mean square acceleration response of an oscillator (in this case the package) can be defined as:

$$\text{mean square} = (\pi/2)(f)(Q)(PSD) . \qquad \text{(Equation 24.17)[20]}$$

Then,

$$\text{mean square at 11 Hz} = (1.57)(11)(5)(0.001)g^2 ,$$

$$= 0.086g^2 .$$

Thus,

$$\text{root mean square at 11 Hz} = 0.294g.$$

The maximum vibration is three to four times the root mean square, or about 0.9 g to 1.2 g. This magnitude of vibration is of little concern for the containment vessel or product bottles.

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APPENDIX D

CONTAINMENT VESSEL STRUCTURAL ANALYSIS

Assumptions

The impact of the package during accident drop condition was simulated using the finite-element analysis method. The package assembly includes the outer stainless steel drum, the cane fiberboard insulation, the stainless steel containment vessel and the aluminum support plate. There is an air gap between the outer drum and the fiberboard. There is another air gap between the fiberboard and the containment vessel. The air gaps are included in the finite-element model to allow the adjoining solid parts to slide relative to each other during impact. The most conservative accident condition was analyzed, in which case the package is dropped from 30 feet onto an unyielding surface with the corner over the center of gravity. The containment vessel flange head is pointed towards the impact surface as shown in Fig. D1. In this configuration, the fiberboard cushion is the thinnest, therefore, the highest induced dynamic stress would be resulted. The fiberboard is assumed to remain confined within the drum during and after the impact. Since the package assembly is symmetric about its neutral (longitudinal) axis, only half of the package is modeled for simplicity. The package is assumed to be dropped in a stand still position from 30 feet above the impact surface. Just before impact, all the potential energy in the cask assembly is converted into kinetic energy as the maximum impact velocity is achieved. In reality, the outer drum will deform after impact, however, it was conservatively assumed that the drum remains elastic during the impact, therefore no plastic energy dissipation is accountable.

The maximum impact velocity V of the cask assembly before impact is calculated by the formula shown below.

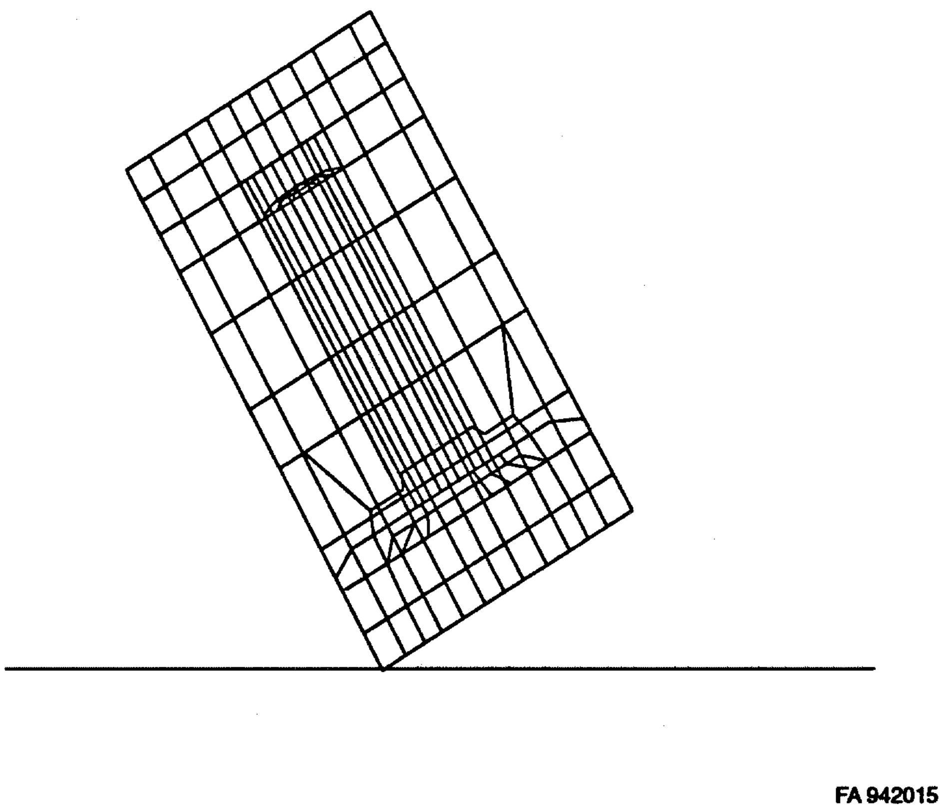

Figure D1. Corner drop of cask assembly.

$$V = (2 \times g \times h)^{0.5}$$

$$= (2 \times 32.2 \times 30)^{0.5}$$

$$= 43.95 \text{ ft/sec.}$$

$$= 527.4 \text{ in/sec.}$$

where:

g = gravitational constant = 32.2 ft/sec^2

h = height of the cask assembly above ground = 30 ft.

The center of gravity of the cask assembly is found to be at an axial distance of 14.94 inches from the top of the outer drum. The outer drum has a diameter of 19.25 inches and a height of 33.5 inches. The line connecting the C.G. to the point of impact intersects the drum lid at an inclined angle of 58.6 degrees. At impact, the inclined angle between the side of the drum and the ground surface is 58.6 degrees. The impact velocity can be resolved into two orthogonal components. The velocity component V_x perpendicular to the longitudinal axis is found as:

$$V_x = V \times \cos 58.6°$$

$$= 527.4 \times 0.521$$

$$= 275 \text{ in/sec.}$$

The velocity component V_y parallel to the longitudinal axis of the drum is found as.

$$V_y = V \times \sin 58.6°$$

$$= 527.4 \times 0.854$$

$$= 450 \text{ in/sec.}$$

Finite-Element Model

The finite-element model of the package was divided into five regions according to their material properties. The outer stainless steel drum was modeled as shell elements. The air gaps between the outer drum and fiberboard, as well as the air gaps between the containment vessel and the fiberboard, were modeled as 8-node solids with material properties extremely light weight and very low stiffness. The containment vessel was modeled as 8-node solids with material properties representing stainless steel. The aluminum plate supporting the containment vessel was modeled as solids with material properties representing aluminum metal. The fiberboard was modeled as 8-node solids with material properties derived from published data (ASTM C-208).[4]

The finite-element model of the package shown in Fig. D2 includes the following components:

- the outer drum
- the fiberboard
- the containment vessel
- the aluminum support plate

Dynamic Stress Results

In a preliminary analysis, the dynamic stresses of the cane fiberboard during impact were found to be within the linear range, therefore, a linear dynamic impact analysis was performed. The impact velocities were entered in the program as initial conditions of all the finite-element nodes at time zero (the very first moment of impact). The analysis program ADINA[14] used a time-step integration computing technique to find the propagating strain wave and dynamic stresses within the package assembly generated

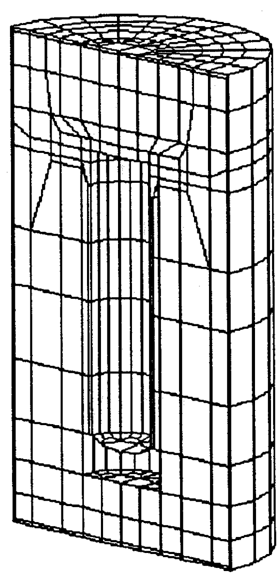

Fig. D2. Finite element model of package assembly.

by the impact. Figure D3 shows a portion of the containment vessel finite-element model representing the area near the flange head. The locations of four elements 40, 46, 52 and 56 are noted on the figure. The stresses of these four elements were examined, in particular, because they are located in the region of highest stress on the containment vessel. The transient dynamic stress histories within the four elements are graphically presented in the following pages as functions of stresses versus elapsed time from the moment of impact.

Figures D4 through D9 graphically display the transient dynamic stress histories within the four finite-elements of interest at the containment vessel flange area. The subscripts x, y, and z are the direction indices representing the radial (x), longitudinal (y), and hoop directions (z). The stress Sxx represents the tensile stress in the radial direction. The Stress Syy represents the tensile stress in the longitudinal direction. The stress Szz represents the hoop stress in the circumferential direction. The stresses Sxy, Sxz and Syz represent the shear stresses along the radial-longitudinal, radial-circumferential, and longitudinal-circumferential, surfaces respectively. All stresses presented in the figures are in units of pounds per square inch. Figure D10 shows the reaction forces at the point of impact. Figures D11 through D14 display the dynamic stress histories of all stresses within each finite-element of interest.

The maximum stresses in each stress category and the corresponding element number are listed below.

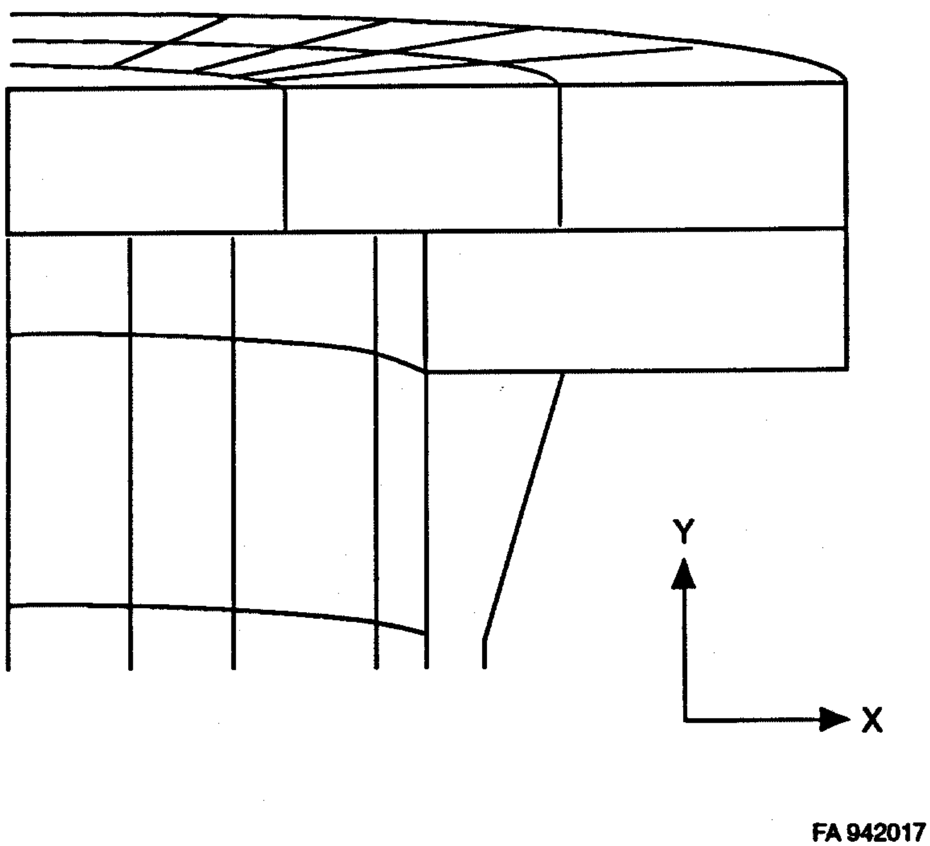

Figure D2. Locations of the finite elements within the containment vessel flange region.

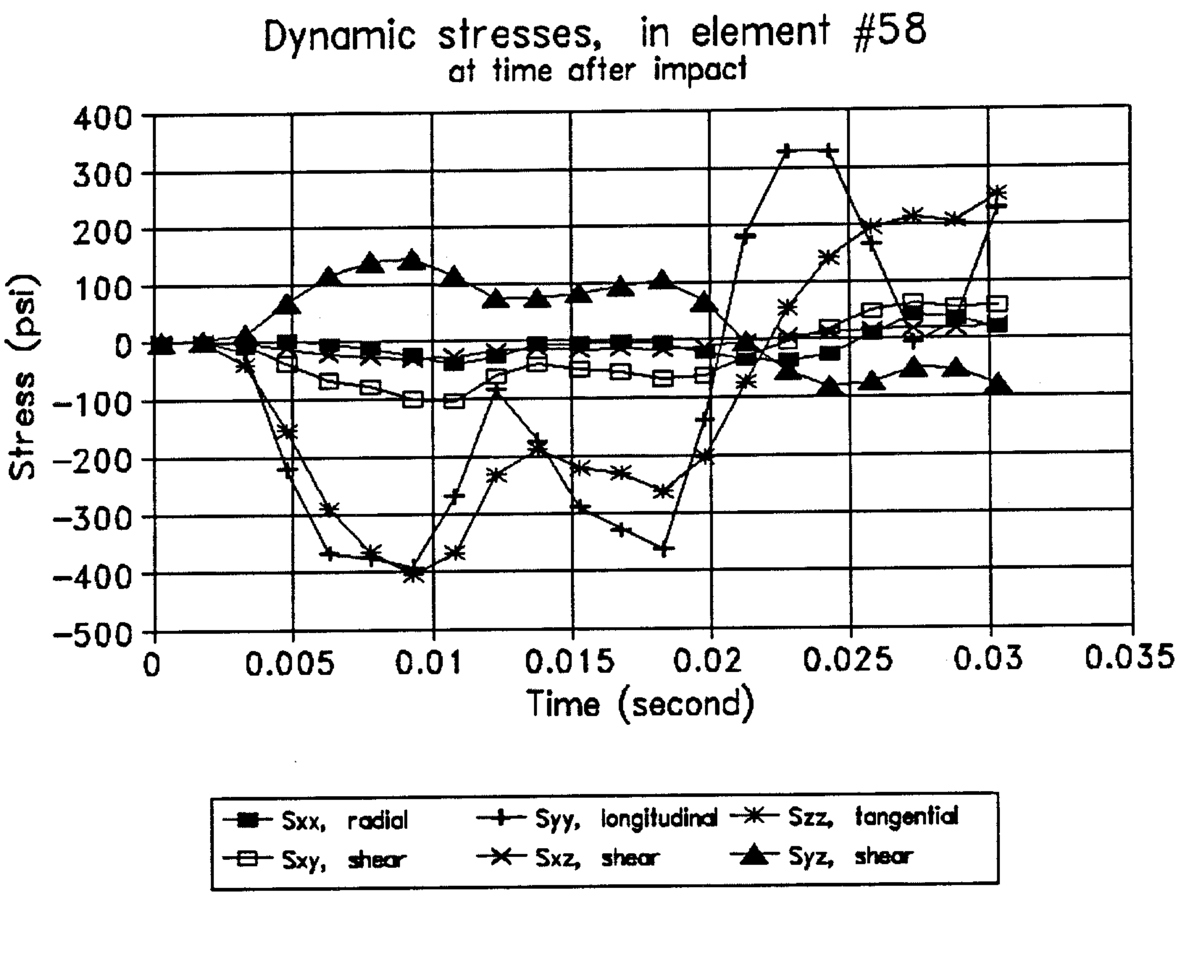

Fig. D4. Time history of transient dynamic stress (radial normal stress) Sxx.

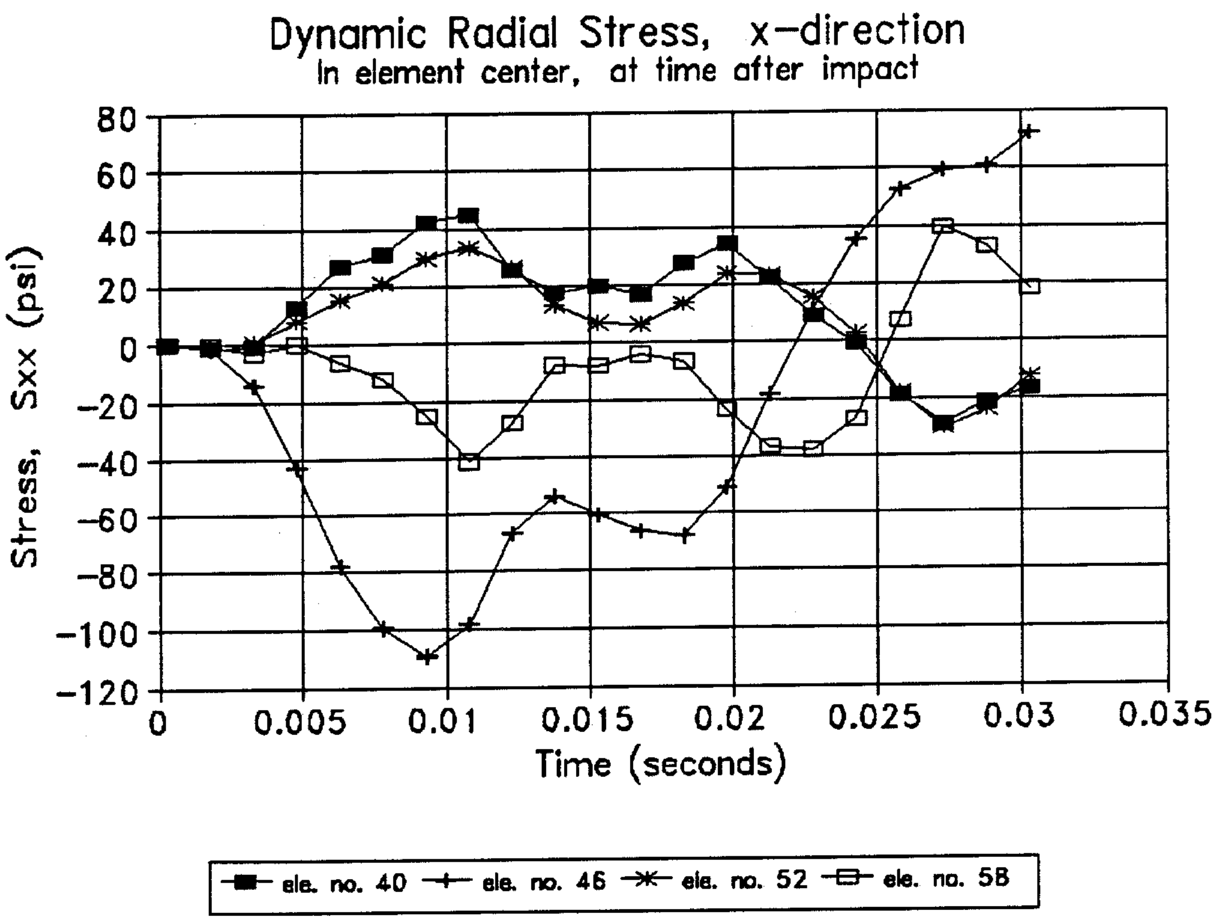

Fig. D5. Time history of transient dynamic stresses (radial normal stress) Syy.

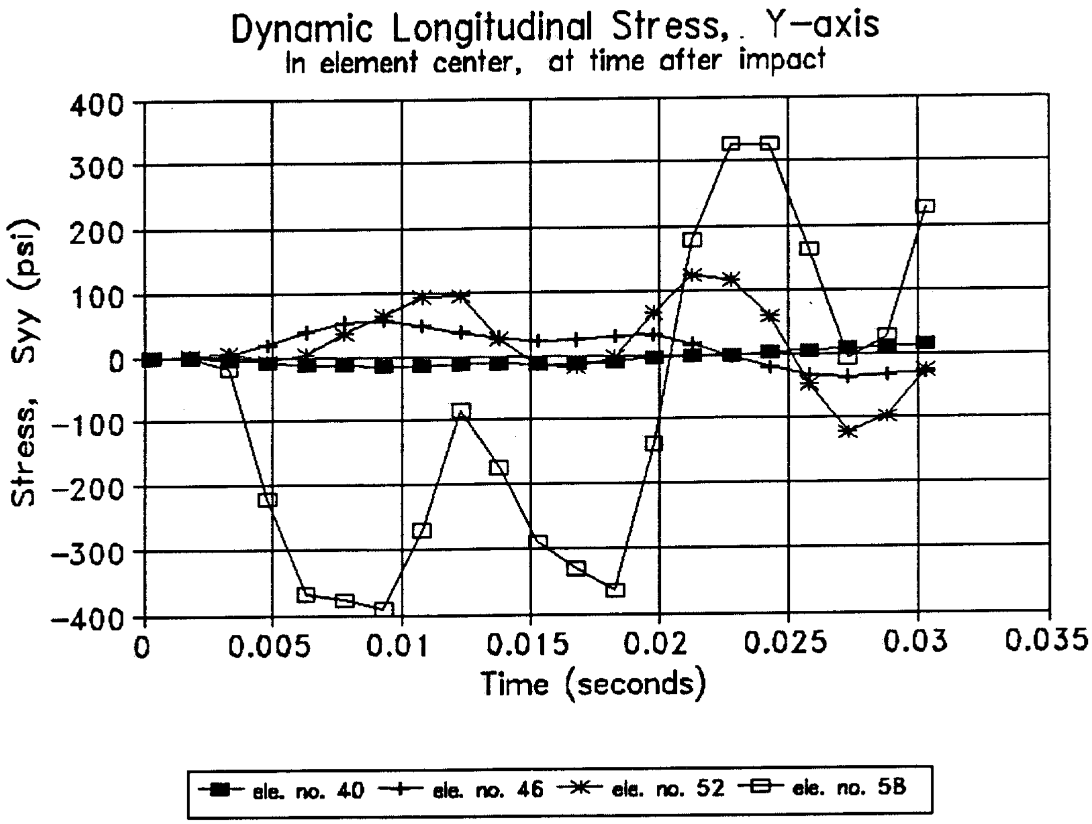

Fig. D6. Time history of transient dynamic stresses (radial normal stress) Szz.

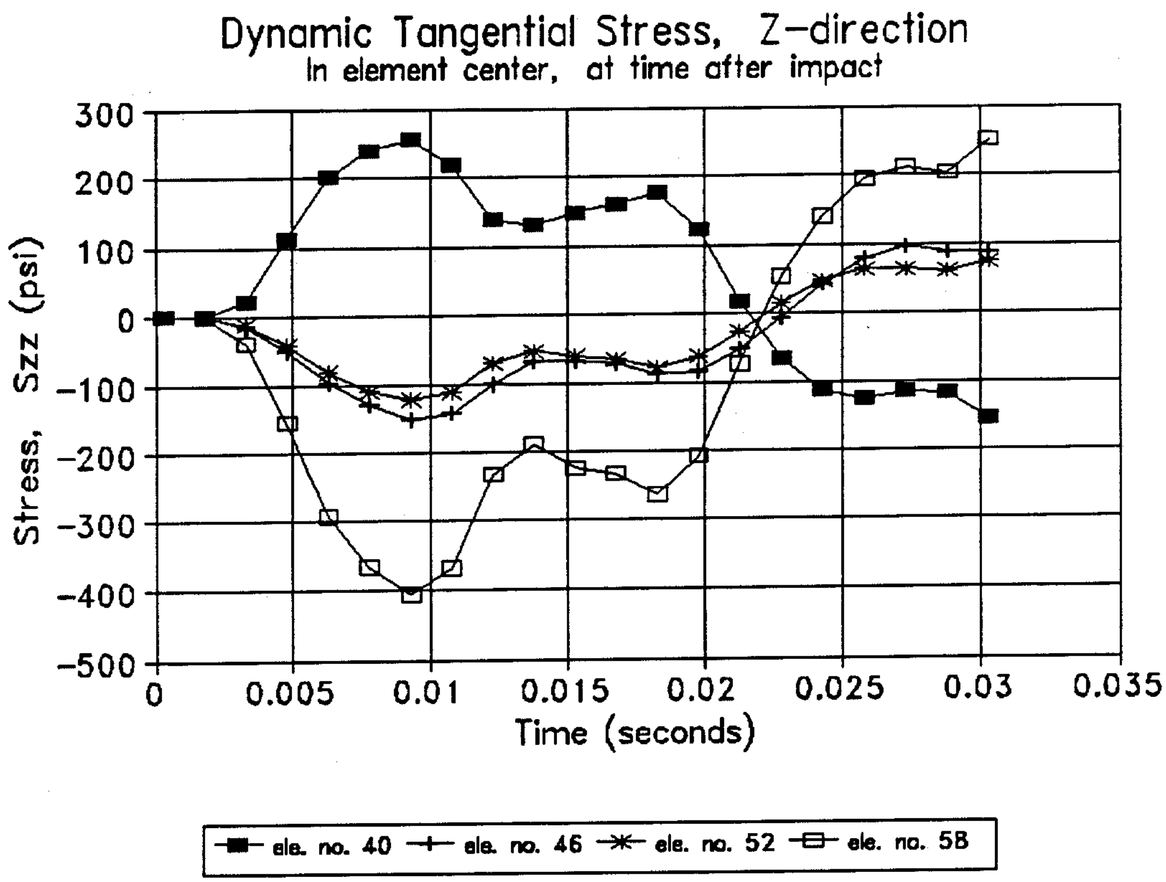

Fig. D7. Time history of transient dynamic stresses (radial normal stress) Sxy.

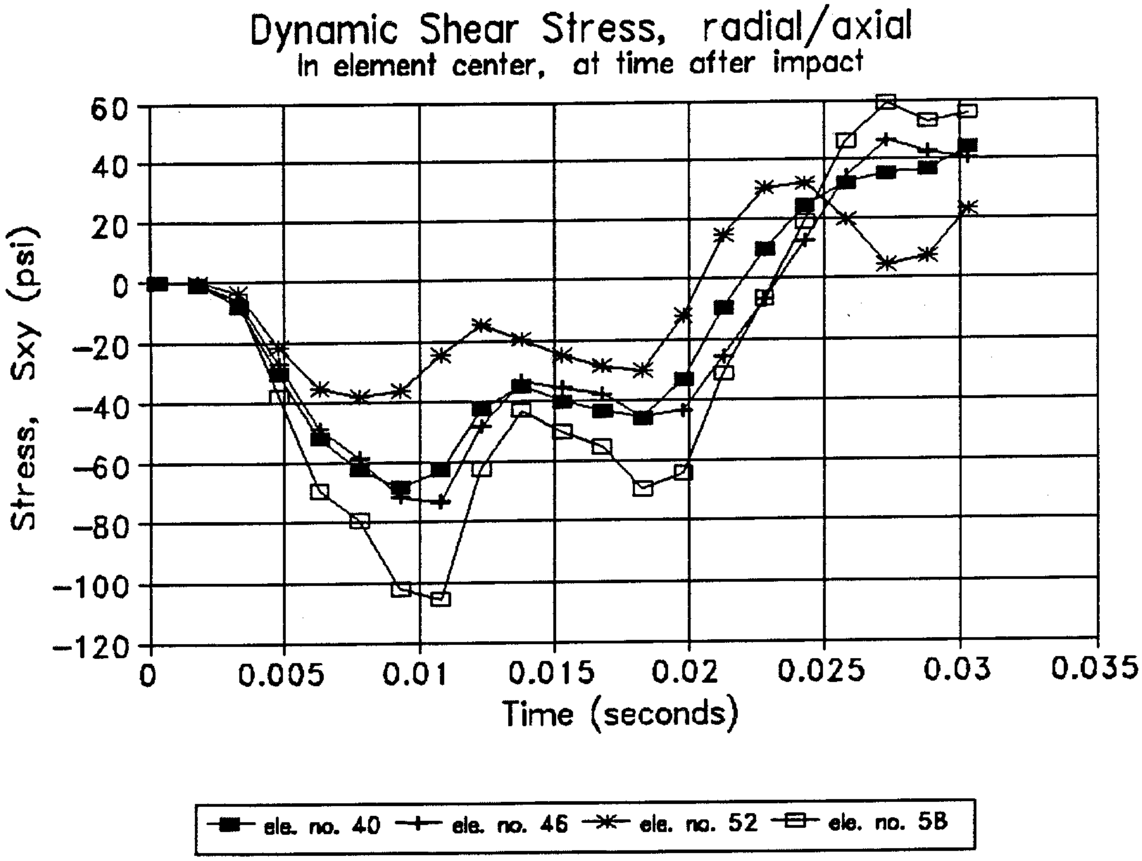

Fig. D8. Time history of transient dynamic stresses (radial normal stress) Sxz.

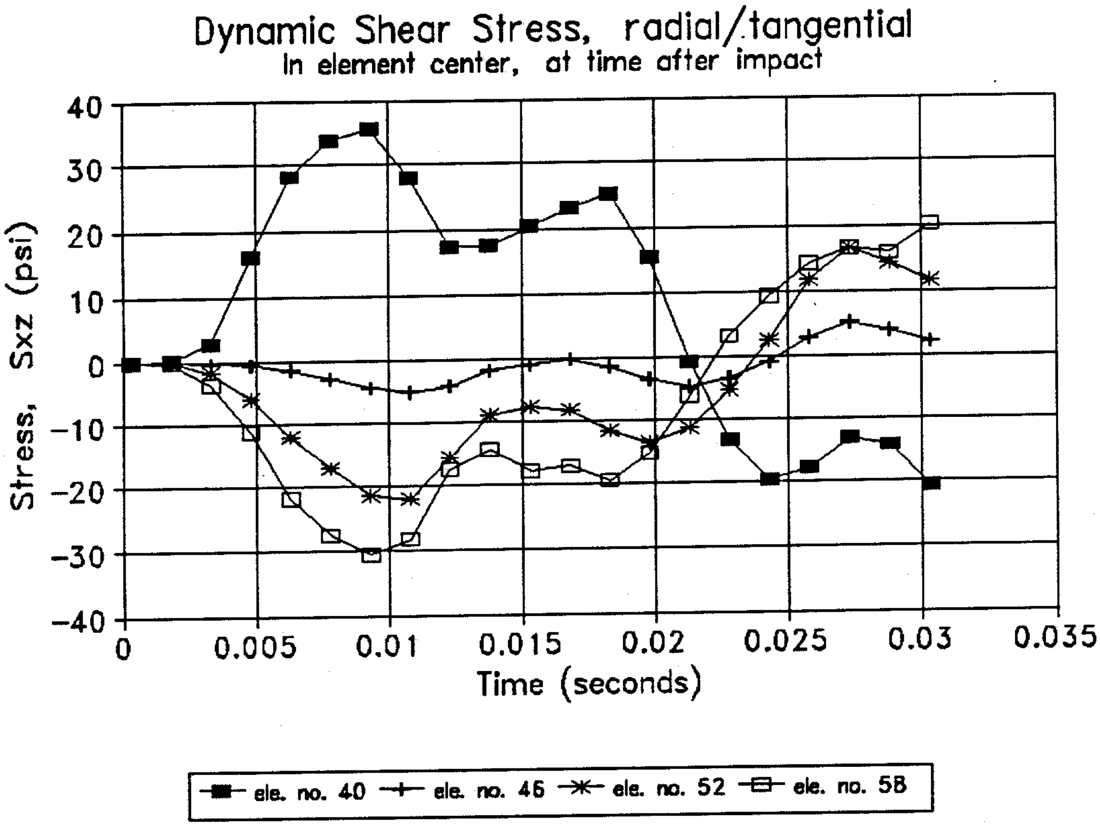

Fig. D9. Time history of transient dynamic stresses (radial normal stress) Syz.

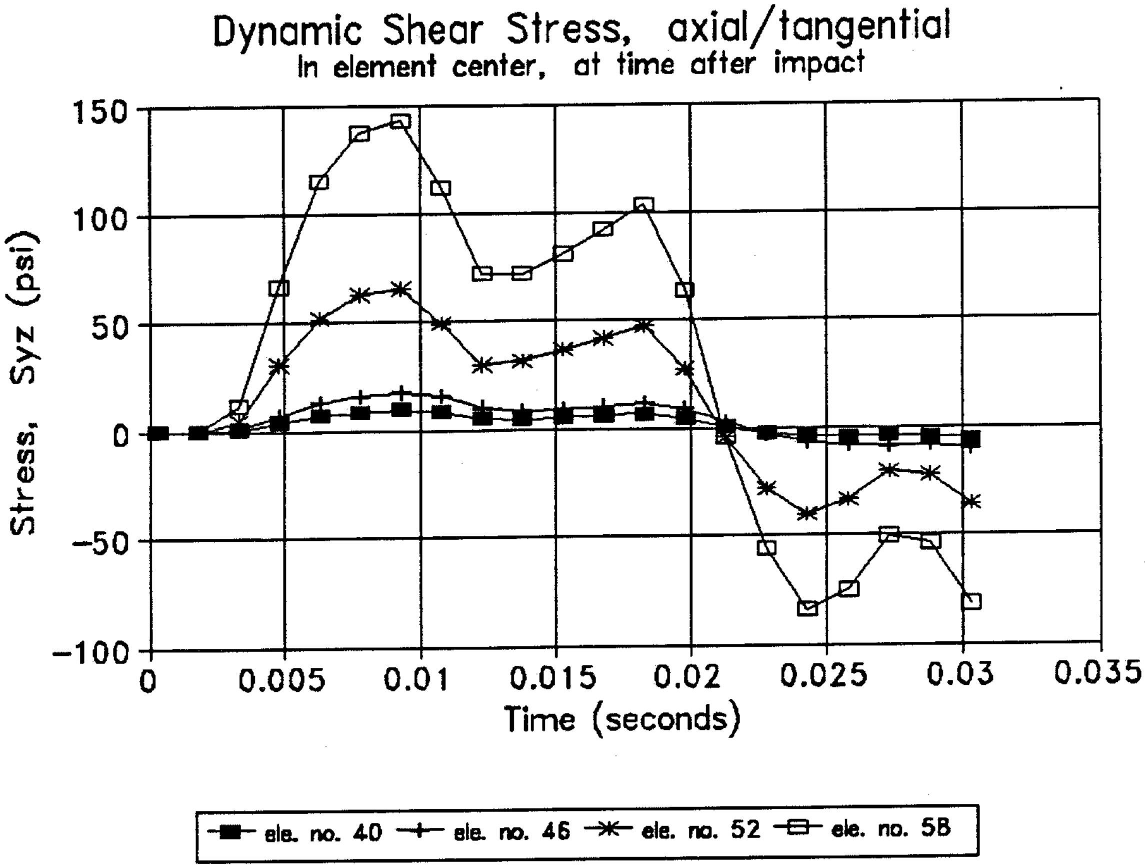

Fig. D10. Time history of reaction forces at point of impact.

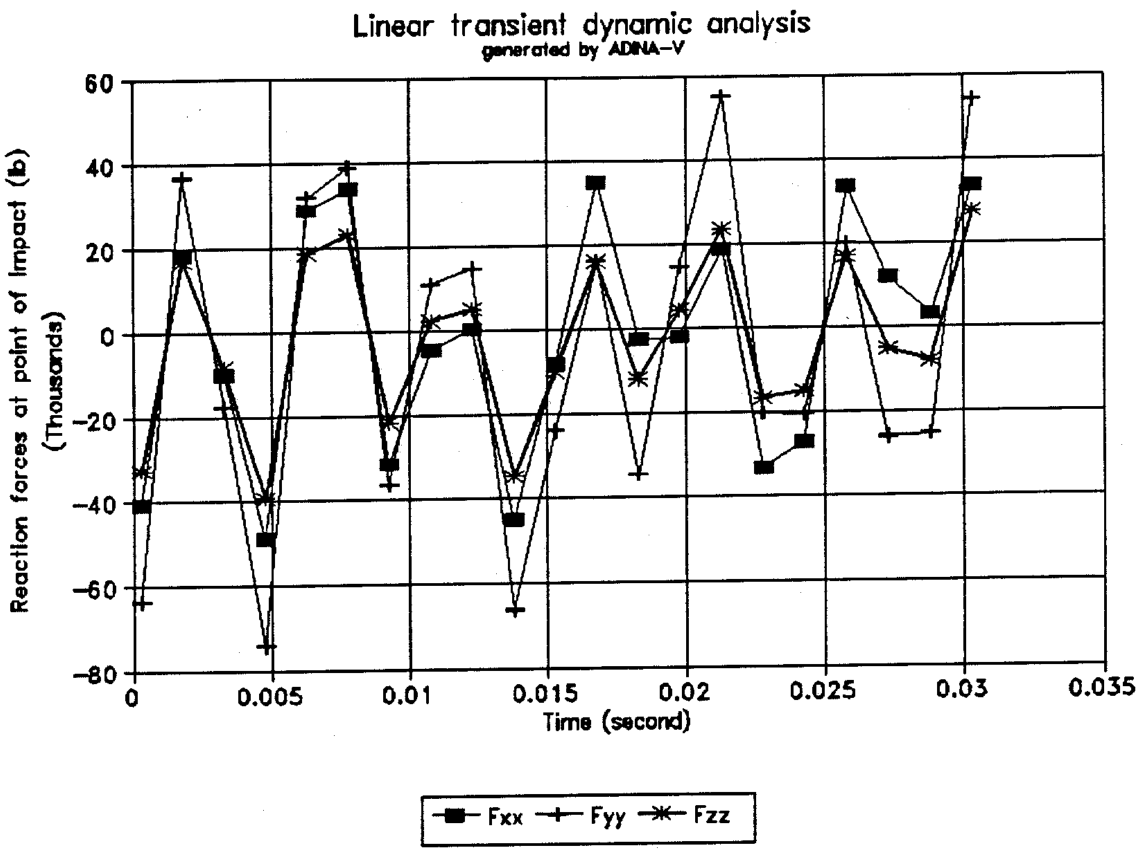

Fig. D11. Time history of transient dynamic stresses within finite-element no. 40.

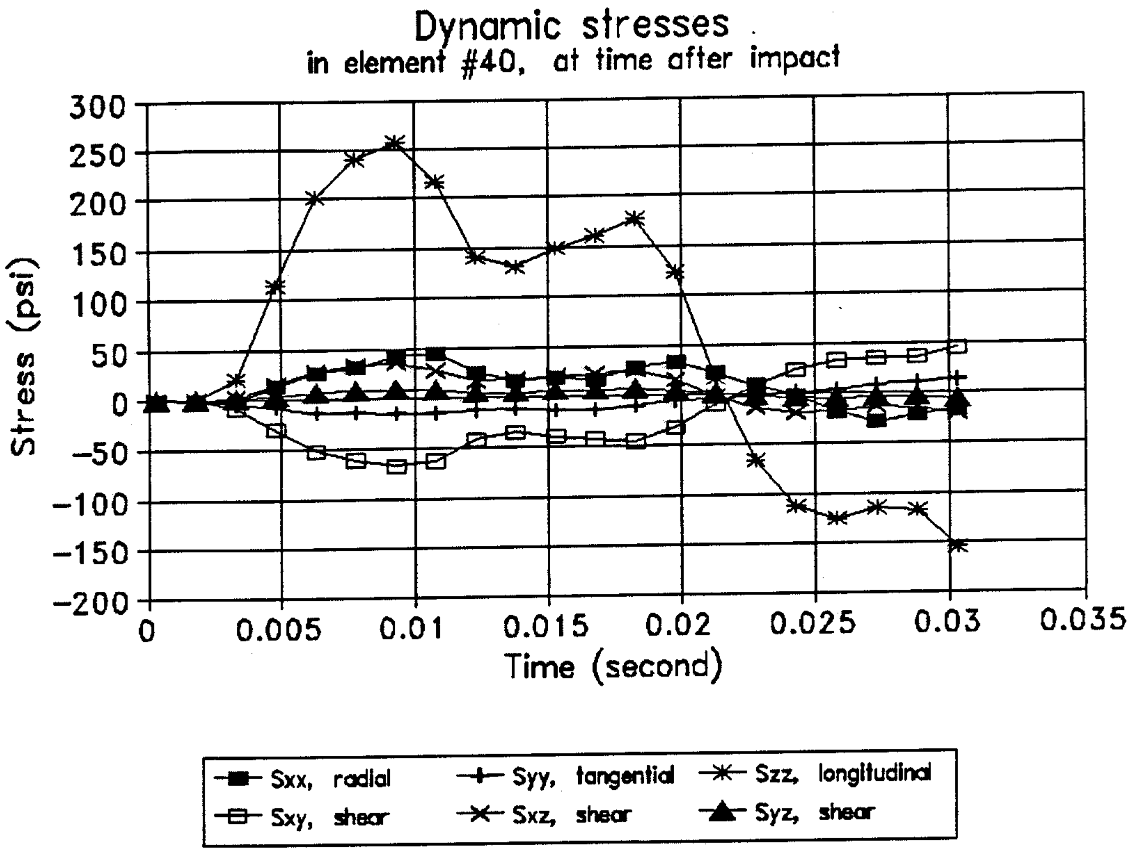

Fig. D12. Time history of transient dynamic stresses within finite-element no. 46.

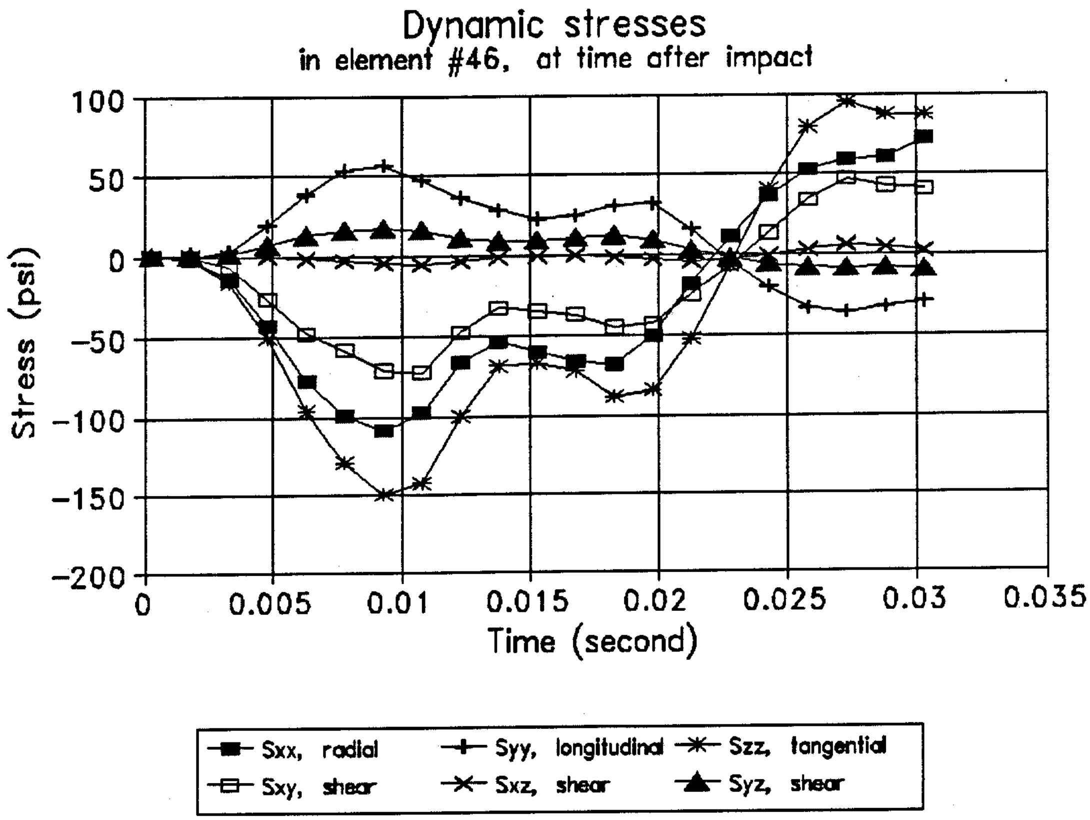

Fig. D13. Time history of transient dynamic stresses within finite-element no. 52.

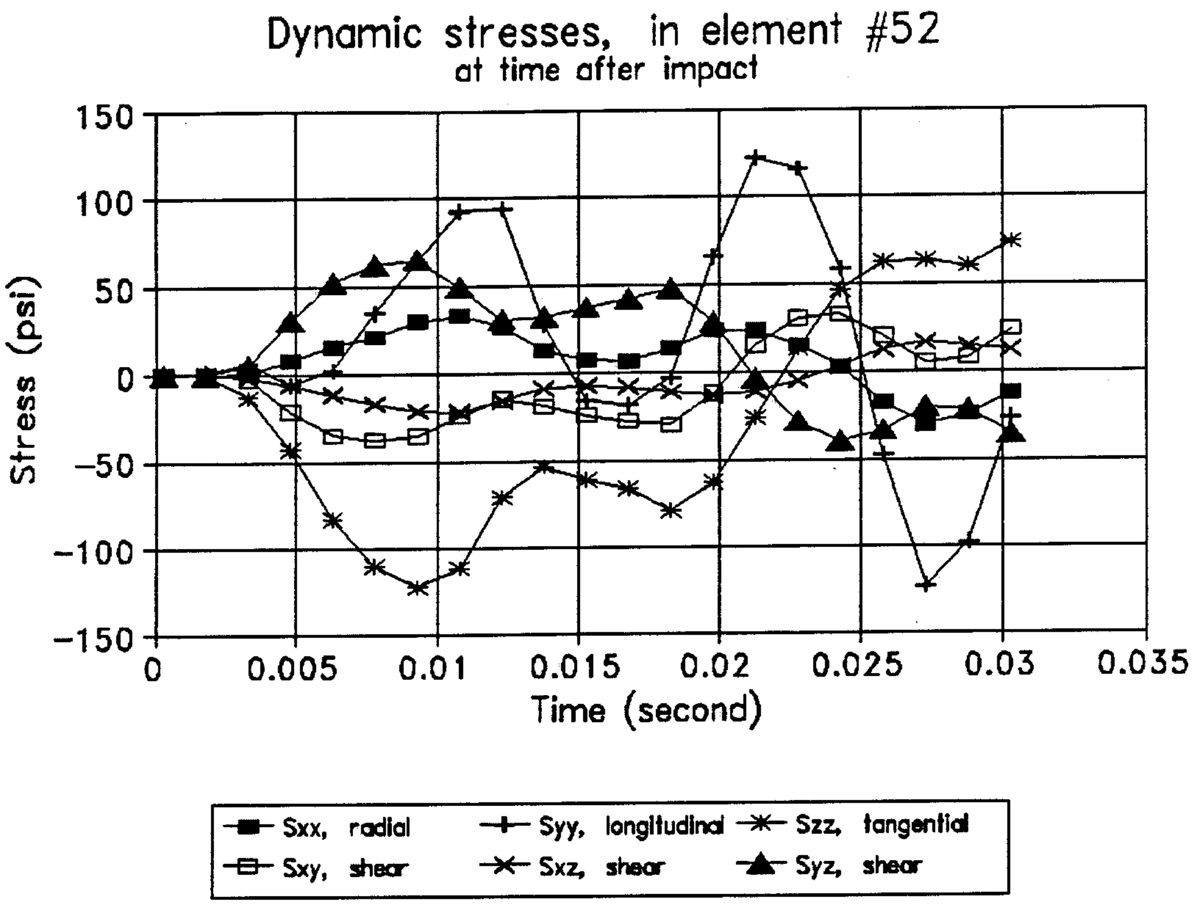

Fig. D14. Time history of transient dynamic stresses within finite-element no. 58.

stress type	maximum stress (psi)	element number	reference figure
Sxx, radial, normal	-100	46	Fig. D4
Syy, longitude, normal	-400	58	Fig. D5
Szz, tangential, normal	-400	58	Fig. D6
Sxy	-105	58	Fig. D7
Sxz	+140	58	Fig. D8
Syz	+ 35	40	Fig. D9

The maximum stresses occur at element 58, where the containment vessel flange head transitions into the flange neck. The normal stresses are compressive stresses. Since they occur at different time intervals, no attempt was made to find the stress intensities by combining normal stresses.

According to Table 2.1,[2] the maximum allowable stress value at 250 degrees Fahrenheit for 304L stainless steel SA-182 (flange material) and SA-312 (vessel pipe material) is 14,300 pounds per square inch. For hypothetical accident drop condition, the allowable stress is permitted to increase by 50 percent. Therefore, the allowable tensile stress for the package during accident drop is 21,450 pounds per square inch. The maximum stress experienced in the vessel flange is 400 pounds per square inch. Therefore, the design margin for this containment vessel is on the order of 53.

Finite-Element Model Input

The finite-element program ADINA was used to solve the package impact analysis. The finite-element model consists of 1324 nodes and 1266 elements. More than 100 printed pages are required if the complete input listing is presented. In order to conserve paper, the complete input listing of the finite-element model is not presented herein. The input listing and output have been downloaded from the mainframe computer and saved on floppy disks. However, the control cards and the material properties are presented in here and can be reviewed. The geometry information listing is abbreviated but the accuracy has been verified by reviewing the graphical presentation of the model.

```
/JOB
ADINA.
USER,S446001,NEWPASS,KCF.CRY
CHARGE,MA3300CC,ADMIN.
JOB,JN=ADINA,CL=L05,T=100.
ACCOUNT,AC=S446001,PW=NEWPASS.
GETLIB,AP=ADINA5,LEV=5NL5.
ASSIGN,DN=$IN,A=FT05.
ADINA5.
/EOR
LIST OFF
TRANSIENT ANALYSIS OF AN AXISYMMETRIC CASK DROPPED FROM 30 FT. 5/02/91
 13240001110  6    0    1  101   0.0003          .0    4    0    0    0    0
0
C***  MASTER CONTROL
99999   0    0    0    0    0    0   50 3000
C***  3  LOAD CONTROL
    0    0    0    0    0    0    0    0    0    0    0    0    0    0 0 1 0
0
C***  4  MASS AND DAMPING CONTROL
    1    0    0    0         .0       .0
C***  5  EIGENVALUE SOLUTION CONTROL
    0    0    0    0    0    0
C***  6  TIME INTEGRATION METHOD CONTROL
    0    20.500000000.25000000   0    0    0    0    0
C***  7  INCREMENTAL SOLUTION CONTROL
C***0   -3  0 000 15.001000000.010000000.0500000000.50000000       .1       .0

C***  8  PRINT-OUT CONTROL
    1    1    5    0    1    1    0
C***  9  PORTHOLE SAVE CONTROL
    0    0    0   60   60             0    0    0    0
C***  BLOCK DEFINITION CARD FOR PRINT-OUT TIME STEPS
    1    0    5
C***  BLOCK DEFINITION CARDS FOR NODAL QUANTITIES PRINT-OUT
  223  223    1  228  228    1  229  229    1  241  241    1  249  249    1
C***  TIME FUNCTIONS
    0    2
C***  NODAL POINT DATA
    1    0    0    1    0    0    0    0.000      0.000     0.000 1    0 0   00
0
    2    0    0    1    0    0    0    1.600      0.000     0.000 1    0 0   00
0
    ...
    ...
    ...
 1323    0    0    0    1    1    1   -2.500     28.200    -4.330 1    0 0   00
0
 1324    0    0    0    1    1    1   -1.294     28.200    -4.830 1    0 0   00
0
C***  INITIAL CONDITIONS, initial velocity defined
    1    0    0    0    0
C***  INITIAL DISPLACEMENTS
 1324    0         0         0         0         0         0         0
0
C***  INITIAL VELOCITIES
    1    1       275.      450.       0.        0         0         0
0
   56    0       275.      450.       0.        0         0         0
0
   58    1       275.      450.       0.        0         0         0
0
 1324    0       275.      450.       0.        0         0         0
0
C***  INITIAL ACCELERATIONS
```

```
 1324
C***  ELEMENT GROUP 1,  3-D SHELLS, S.S. DRUM
C***  column 52 should be 1, to print out stress at location 21 using stress
table
   7 252   0   0   0   0   4   4   2   0   0   0   0   1   1   1   0       0
C***  MATERIAL PROPERTY
    1     .0010395       0
29000000.     0.3             0
C*** SHELL THICKNESS TABLE CARD
   .05         .05        .05        .05
C***  ELEMENT DATA CARDS
    1    4    1    1    1         0         0    0    0         0
    1   10    7    2
    2    4    1    1    1         0         0    0    0         0
   10   16   15   18
    ...
    ...
    ...
  252    4    1    1    1         0         0    0    0         0
  843  844  818  830
C*** ELEMENT GROUP 2, 3-D SOLIDS, AIR GAP
   3 366   0   0       0   8   0   0   0   1       1   0   1   1   0   0   0
C*** MATERIAL PROPERTY CARD
    1  .0000001
10.          .1
C*** STRESS OUTPUT TABLE CARD
   21    0
C*** ELEMENT DATA CARD
    1    8    8    0    1    0         0         0    0    0         0
  150  151  153  156    1    2    7   10

    2    8    8    0    1    0         0         0    0    0         0
  156  164  161  162   10   18   15   16
    ...
    ...
    ...

  366    8    8    0    1    0         0         0    0    0         0
  610  580  586  616 1234 1209 1214 1239

C*** ELEMENT GROUP 3, 3-D SOLID, FIBER BOARD
   3 480   0   0       0   8   0   0   0   1       1   0   1   1   0   0   0
C*** MATERIAL PROPERTY CARD
    1  .0000328
  500.       .1
C*** STRESS OUTPUT TABLE CARD
   21    0
C*** ELEMENT DATA CARD
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  165  166  168  171  150  151  153  156

    2    8    8    0    1    0         0         0    0    0         0
  171  179  176  177  156  164  161  162
    ...
    ...
    ...

  480    8    8    0    1    0         0         0    0    0         0
  694  712  646  634 1304 1319 1264 1254

C***  ELEMENT GROUP 4.1,  3-D SHELLS, INNER VESSEL, LOWER PART
   7  18   0   0   0   0   4   4   2   0   0   0  -1   1   1   1   0       0
C***  MATERIAL PROPERTY
    1     .001312        0
29000000.     0.3             0
```

```
C*** SHELL THICKNESS TABLE CARD
   .26        .26        .26        .26
C***  ELEMENT DATA CARDS
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  266  267  271  275
    2    4    0    1    1         0         0    0    0         0
  275  271  270  274
    ...
    ...
    ...
   18    4    0    1    1         0         0    0    0         0
  276  949  950  266
C*** ELEMENT GROUP 4.4, 3-D SOLID, INNER VESSEL, UPPER PART
   3 126   0   0       0   8   0   0   0   1       1   0   1   1   0   0   0
C*** MATERIAL PROPERTY CARD
    1  .001312
29000000.      .3
C*** STRESS OUTPUT TABLE CARD
   21    0
C*** ELEMENT DATA CARD
    1    8    8    1    1    0         0         0    0    0         0
D
  294  303  299  295  280  289  285  281

    2    8    8    1    1    0         0         0    0    0         0
  303  302  298  299  289  288  284  285
    ...
    ...
    ...
  126    8    8    1    1    0         0         0    0    0         0
  352  358  574  676 1019 1024 1204 1289

C*** ELEMENT GROUP 5, 3-D SOLIDS, ALUMINUM PLATE
   3  24   0   0       0   8   0   0   0   1       1   0   1   1   0   0   0
C*** MATERIAL PROPERTY CARD
    1  .000194
15.000000      .3
C*** STRESS OUTPUT TABLE CARD
   21    0
C*** ELEMENT DATA CARD
    1    8    8    0    1    0         0         0    0    0         0
  569  575  557  425  223  224  221  220

    2    8    8    0    1    0         0         0    0    0         0
  570  576  558  426  569  575  557  425
    ...
    ...
    ...
   24    8    8    0    1    0         0         0    0    0         0
  580  562  568  586 1209 1194 1199 1214
```

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APPENDIX E

PRELOAD CONDITION OF CONTAINMENT VESSEL BOLTING

The following calculations determine the preload condition of the containment vessel bolting of the shipping package during shipment at minus 40 degrees Fahrenheit. To demonstrate that the containment vessel O-rings do not unseat during Normal Conditions of Transport at minus 40 degrees Fahrenheit, the following assumptions have been made:

- The package is assembled at 70 degrees Fahrenheit.
- The package is shipped at minus 40 degrees Fahrenheit.
- The gasket seating load W, m^2 does not change appreciably as a function of temperature over the range associated with Normal Conditions of Transport in 10 CFR 71.

The total strain (ε_T) in the fasteners when assembled at 70 degrees Fahrenheit consists of the strain associated with the gasket load (ε_G) and the strain due to seating the blind flange to the containment vessel body flange (ε_F). This total strain is induced by the torque requirements of 30 plus or minus 2 foot-pounds.

$$\varepsilon_T = \varepsilon_G + \varepsilon_F.$$

Total strain in a bolt (ϵ_T) is calculated as follows:

$$F_T = T/(0.2)(d),$$

$$= 336/(0.2)(0.75) \; ,$$

$$= 2240 \text{ lb.}$$

where:

T = 336 in.-lb (minimum torque of 28 ft-lb),

d = 0.75 in (bolt shank diameter).

Total stress in a bolt is calculated as follows:

$$\sigma_B = F_T/A_1 \; ,$$

$$\sigma_B = 2240/0.442 = 5{,}068 \text{ psi} \; .$$

where:

A_1 = cross-sectional area of bolt shank ,

$A_1 = \pi(.375)^2 = 0.442 \text{ in.}^2$.

Then, total strain is equal to:

$$\epsilon_T = \sigma_B/E ,$$
$$= 5{,}068/30 \times 10^6$$
$$\epsilon_T = 0.00017 \text{ in./in.}$$

where:

$E = 30 \times 10^6$ psi (Modulus of Elasticity, Table A6) .

The resulting change in bolt length due to this strain is:

$$\Delta L_B = L_1\epsilon_T = (1.747)(0.00017) = 0.000297 \text{ in.}$$

where:

L_1 = Length of bolt between head and nut ,

L_2 = Thickness of flange and washer combination ,

Assume $L_1 = L_2 = 1.747$ in.

Strain in a bolt resulting from gasket loading (ϵ_G) is computed as follows:

$$F_G = W_{m2}/8 ,$$
$$F_G = 2{,}987/8 = 373 \text{ lb} .$$

where:

$$W_{m2} = 2{,}987 \text{ lb.}$$

Then, stress is calculated as follows:

$$\sigma_G = F_G/A_1 ,$$

$$= 373/0.442 ,$$

$$\sigma_G = 844 \text{ psi} .$$

Then strain is equal to:

$$\epsilon_G = \sigma_G /E ,$$

$$= 844/(30 \times 10^6) ,$$

$$\epsilon_G = 0.000028 \text{ in/in} .$$

Thus, the strain in the bolt associated with the face to face seating of the lid and body is:

$$\epsilon_F = \epsilon_T - \epsilon_G ,$$

$$= (0.00017 - 0.000028) ,$$

$$\epsilon_F = 0.000142 \text{ in./in.}$$

When the package is shipped at minus 40 degrees Fahrenheit, the original preload (strain) in each bolt is partially relieved as computed below:

Coefficients of thermal expansion for flange and bolt:

$$\alpha_F = 9.8 \times 10^{-6} \text{ in/in/°F (For flange, Table A4)},$$

$$\alpha_B = 7.0 \times 10^{-6} \text{ in/in/°F (For bolt, Table A6)}.$$

The change in bolt length from 70 to minus 40 degrees Fahrenheit is equal to:

$$\Delta L_i = \alpha_i (\Delta T) L_i ,$$

$$\Delta L_B = \alpha_B (\Delta T) L_1 ,$$

$$\Delta L_B = (7.0 \times 10^{-6})(-110)(1.747) ,$$

$$\Delta L_B = -0.00135 \text{ in. (contraction)} .$$

The change in thickness of the lid flange and washer combination from 70 to minus 40 degrees Fahrenheit is:

$$\Delta L_F = \alpha_F (\Delta T) L_2 ,$$

$$= (9.8 \times 10^{-6})(-110)(1.747) ,$$

$$\Delta L_F = -0.00188 \text{ in (contraction)} .$$

The difference between the contraction of the bolt and the reduction in the thickness of the flange is:

$$\Delta L_T = \Delta L_B - \Delta L_F ,$$

$$= -0.00135 - (-0.00188) ,$$

$$\Delta L_T = .00053 \text{ in.}$$

Therefore the apparent strain reduction due to thermal contraction is calculated as follows:

$$\epsilon_{\Delta T} = -\Delta L_T / L_1 ,$$

$$\epsilon_{\Delta T} = -.00030 \text{ in./in.}$$

Since this strain reduction at minus 40 degrees Fahrenheit appears to be slightly larger then the strain originally induced by torquing the bolts to 30 foot-pounds at 70 degrees Fahrenheit, the pre-load condition would tend to be relieved during transport at minus 40 degrees Fahrenheit.

The entire pre-load is not relieved as there will always exist a pre-load (strain) in the bolts due to compression of the O-rings until ΔL_T exceeds the following:

$$\Delta L_T \geq X + \epsilon_T (L_1) - \alpha_R (0.135) \Delta T ,$$

where:

α_R =0.000062 in./in./°F (O-ring coefficient of thermal expansion)

X = original compression in O-rings = X = .028 in. minimum,

$\epsilon_T(L_1)$ = original preload stretch in bolt due to torque ,

$\sigma_R(0.135)\Delta T$ = contraction of O-ring diameter at -40°F ,

Uncompressed O-ring thickness is 0.135 in .

Thus,

$$\Delta L_T \geq 0.0268 \text{ in.}$$

Since $\Delta L_T \ll 0.0274$ inch, the preload (strain) due to gasket loading is not reduced significantly; but the containment vessel lid will separate from the container body flange by the amount as shown below:

$$B_S = \text{Body and lid separation distance} \,,$$

$$B_S = \Delta L_T - \epsilon_F(L_1) \,,$$

$$B_S = 0.00053 - 0.000142(1.747) \,,$$

$$B_S = 0.00028 \text{ in.}$$

Even with this separation of lid and flange, there still exists sufficient O-ring compression for containment as calculated below:

$$\phi = \text{O-ring thickness under compression} \,,$$

$$\phi = 0.028 - B_S + \alpha_R(\Delta T)(0.135) \,,$$

$$\phi = 0.028 - 0.00028 + 0.000062(-110)(0.135) \,,$$

$$\phi = 0.0268 \text{ in.}$$

Therefore, the percent compression of the O-ring at minus 40 degrees Fahrenheit is,

$$0.0268/0.135 \quad = 19.85\% \,.$$

Since the normal design compression of the O-rings ranges from 21 to 29 percent, a compression of 19.85 percent is considered more than adequate to maintain containment.

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APPENDIX F

FATIGUE LOADING OF CONTAINMENT VESSEL BOLTING

The following calculations investigate the fatigue loading on the containment vessel bolting during trailer shipment of a shipping package. A number of assumptions are made to establish the maximum possible loading on the containment vessel bolts caused by vibration during shipment of a shipping package under Normal Conditions of Transport.

- The package is assembled at 70 degrees Fahrenheit.
- The package is shipped at 100 degrees Fahrenheit with insolation and 5 Watts internal heat load.
- The maximum vibration level is 1.2 g.
- The containment vessel is loaded with the maximum weight contents of 30.6 pounds without any internal support.
- The bolts are torqued to the maximum specified (30 +2/-0 foot-pounds).
- The containment vessel lid and the contents are synchronous and both are at the maximum up position due to vibration.
- The endurance limit is 40 to 60 percent of the tensile strength for infinite life defined as greater than 10^6 cycles (p. 5-10[11]).

Calculations to determine the maximum stress in the bolts for this projected worse condition of normal transport are given below:

Stress in bolt at maximum torque at 70 degrees Fahrenheit,

$$T = 32 \text{ ft-lb} = 384 \text{ in.-lb},$$

$$\text{Load/bolt} = F = 384/(0.2 \times 0.75) = 2{,}560 \text{ lb/bolt} .$$

where:

0.75 in. = bolt shank diameter

Then, the bolt stress at the threads is equal to:

$$S_{BT} = F/A_s ,$$

$$S_{BT} = 2{,}560/0.334 = 7{,}665 \text{ psi} .$$

where:

A_s = 0.334 in.^{2} (stress area of 0.75 in. bolt),

Stress in the bolts due to a temperature differential of 130 degrees Fahrenheit (maximum potential temperature is about 200 degrees Fahrenheit for normal or accident conditions).

$$S_{B\Delta T} = E(\alpha_F - \alpha_B)\,\Delta T\ ,$$

$$S_{B\Delta T} = (30 \times 10^6)(9.8 \times 10^{-6} - 7.0 \times 10^{-6})(130)\ ,$$

$$S_{B\Delta T} = 10{,}920 \text{ psi}\ .$$

where:

E = 30×10^6 pounds per square inch (used from Table A6 for both flange and bolts as conservative)

Coefficients of thermal expansion:

α_F = 9.8×10^{-6} in./in./°F (For flange, Table A4) ,

α_B = 7.0×10^{-6} in./in./°F (For bolt, Table A6) .

The stress in the bolts due to increased pressure in containment vessel at 200 degrees Fahrenheit ($S_{B\Delta P}$) is thus calculated as follows:

The maximum calculated internal pressure at 200 degrees Fahrenheit is 18.31 pounds per square inch is,

Pressure Differential = Internal pressure-ambient pressure

= (18.31 - 14.7) psi

= 3.61 psig

Force upward due to internal pressure F_P is equal to:

$$F_P = \Delta P(\text{Area}) ,$$

$$F_P = (3.61)(\pi)(6.342)^2/4 ,$$

$$F_P = 114 \text{ lb} .$$

where:

Mean diameter of O-rings in 6.342 inches.

$$\text{Load per bolt} = F_P/8 = 14 \text{ lb} ,$$

$$S_{B\Delta P} = 14/0.334 = 42 \text{ psi} .$$

Stress in bolts due to 1.2 g's on contents - S_{VC}

$$F_{VC} = (1.2)(30.6)/8 ,$$

$$= 4.6 \text{ lb/bolt} ,$$

$$S_{VC} = F_{VC}/A_S ,$$

$$= 4.6/0.334 ,$$

$$= 14 \text{ psi} .$$

where:

Weight of contents = 30.6 lb.

Stress in bolts due to 1.2 g's on the lid of containment vessel,

$$F_{VT} = (1.2)(20)/8 ,$$

$$= 3 \text{ lb/bolt} ,$$

$$S_{VT} = F_{VT}/A_S ,$$

$$= 3/0.334 ,$$

$$= 9 \text{ lb} .$$

where:

Weight of containment vessel lid is 20 pounds

Thus, the maximum stress in the bolt threads at 1.2 g's vibration is then totaled as follows:

$$S_{BV} = S_{BT} + S_{B\Delta T} + S_{B\Delta P} + S_{VC} + S_{VT} ,$$

$$= 7{,}665 + 10{,}920 + 42 + 14 + 9 ,$$

$$S_{BV} = 18{,}650 \text{ psi} .$$

Using 40 percent of the tensile strength given in Table A6 as the endurance limit (yield) for the bolts, a value of 30,000 pounds per square inch is established. This value for yield is applicable to 200 degrees Fahrenheit. When comparing this value to the calculated maximum stress value of 18,650 pounds per square inch, the design is adequate. The actual comparative value would be less if S_{mean} [equal to $(S_{max} + S_{min})/2$] were used as is normally done in fatigue analysis.

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APPENDIX G

DESIGN AND ANALYSIS OF CLOSURE BOLTS

NOTE: This appendix is taken from UCRL-ID-110989 Draft Section 5.2.1.3 with minor editorial revisions. Although this appendix is written from the spent fuel casks perspective, it contains useful information for the DOE drum-type container design.

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APPENDIX G

DESIGN AND ANALYSIS OF CLOSURE BOLTS

1.0 INTRODUCTION

Much of the information in this appendix is from *Design and Analysis of the Closure Bolts of Shipping Casks*[21]

2.0 FUNCTION

Figure G1 shows schematic diagrams of typical closure systems for shipping casks and the closures can be either exposed or protected. A protected closure is one that is protected by a cask wall such that no transverse component of the impact force can be transmitted from the impact limiter to the edge of the lid. Additionally, the puncture pin is not able to touch the edge of the lid to induce a transverse shear force on the bolts. On the other hand, the lid or the bolts of an exposed closure system do not have the protection of the cask wall, as they would in the protected closures. The use of exposed systems is discouraged.

A closure can also be either a single-lid or a double-lid closure system. In the case of a double-lid closure system, the shielding and leak-proof functions are separated or shared between the two closures. For shipping casks without impact limiters to absorb the impact energy and to prevent large local plastic deformation, a double-lid closure system with an inner-lid providing a leak-proof function is recommended because local plastic deformation of the outer closure may be severe enough to damage the seals.

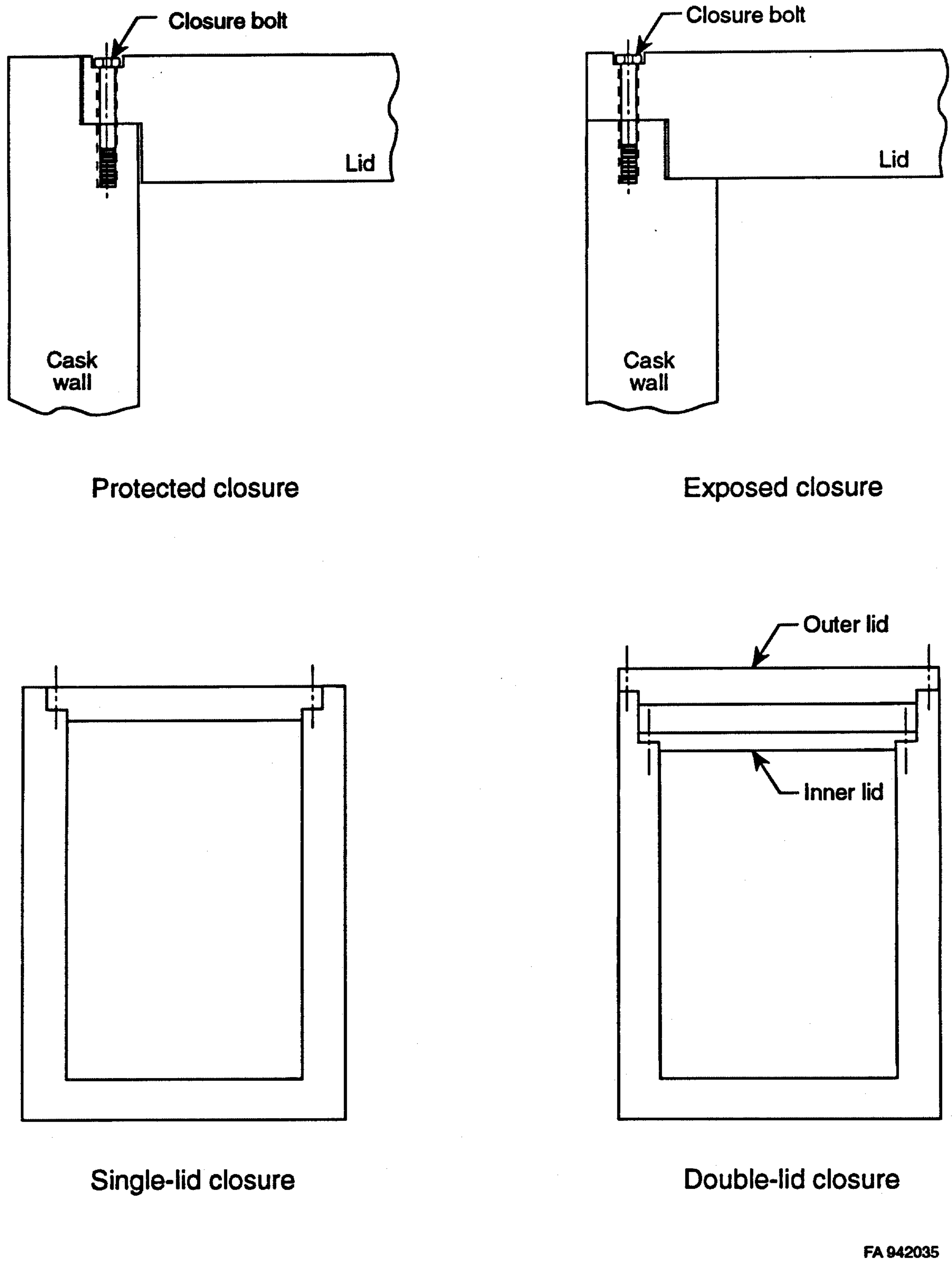


Fig G1. Typical shipping cask closure system.

Seals or gaskets prevents leaks. The two types of seal arrangements considered with and without metal-to-metal contacts between the lid and the cask wall. In the case of a closure system without metal-to-metal contact, gaskets fit between the lid and the cask wall. The lid and the cask wall have no direct contact except through the closure bolts and the seals. The metal-to-metal contact type of closure systems, used often in shipping casks, consist of elastomer or metal O-ring seals, or seals of a self-energized type, in grooves, which are machined into the lid.

Two types of bolt connections are considered depending on whether or not a nut is used and these are shown in Fig. G2. When the bolt is under tension, the real stress distribution along its axis and on its cross-sectional area is rather complicated. However, it is convenient and sufficient to assume a uniform stress distribution on its cross-sectional area (neglecting stress concentration effects) and to assume a stress distribution along its length as shown in Fig. G3. The effective length of the bolt in carrying load is defined in this figure. The effective length of a bolt includes several regions with different cross-sectional areas. The assumed effective lengths in the head, in the nut, or in the threaded region are approximate values.[-22] [23] However, these assumed effective lengths are believed to be sufficient for shipping cask design because they usually are significantly shorter than the grip length of the bolts.

The proposed analysis criteria for the closure bolts of shipping casks follow Regulatory Guides 7.6 and 7.8 of the NRC and the applicable paragraphs of Subsection NB of the ASME Code, Section III. The proposed criteria are intended for the closure bolts of circular, cylindrical shipping casks only.

Subsubarticle NB-3230 of the ASME Code delineates stress limits for bolts used for Class 1 components designed in accordance with the Code. Figure G4 is a diagram showing the requirements

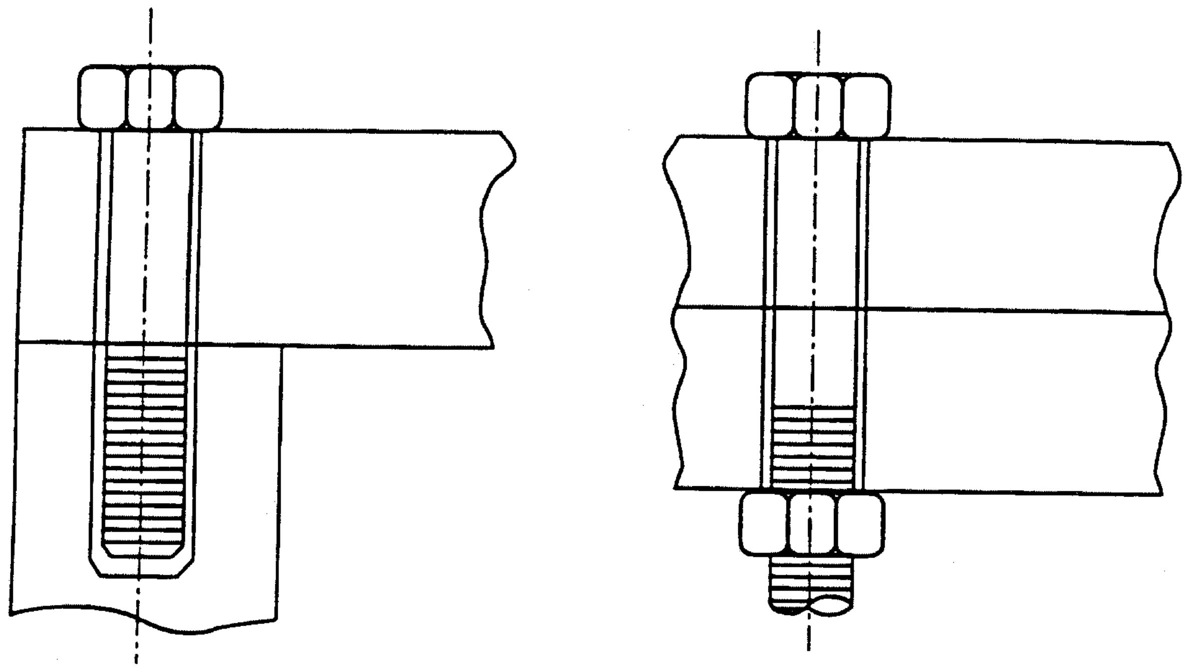


Fig. G2. Two types of bolted connections.

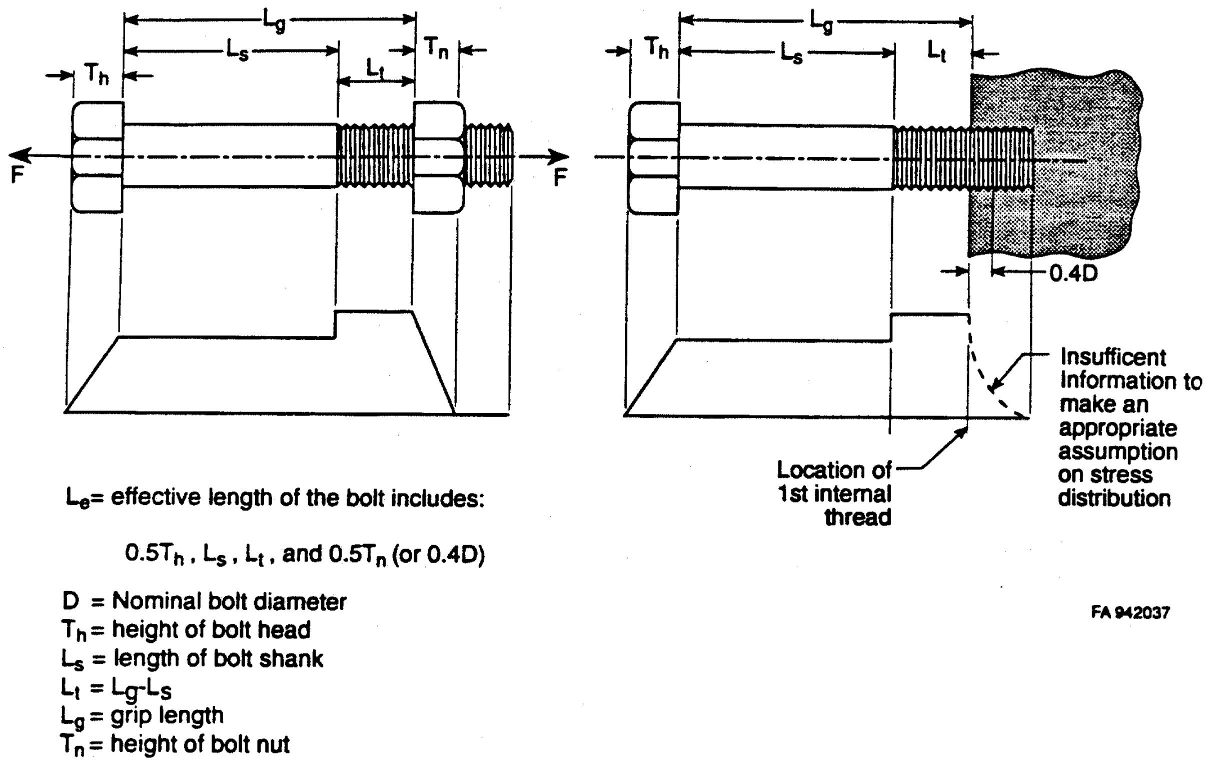


Fig. G3. Assumed stress distribution and effective length of a bolt.

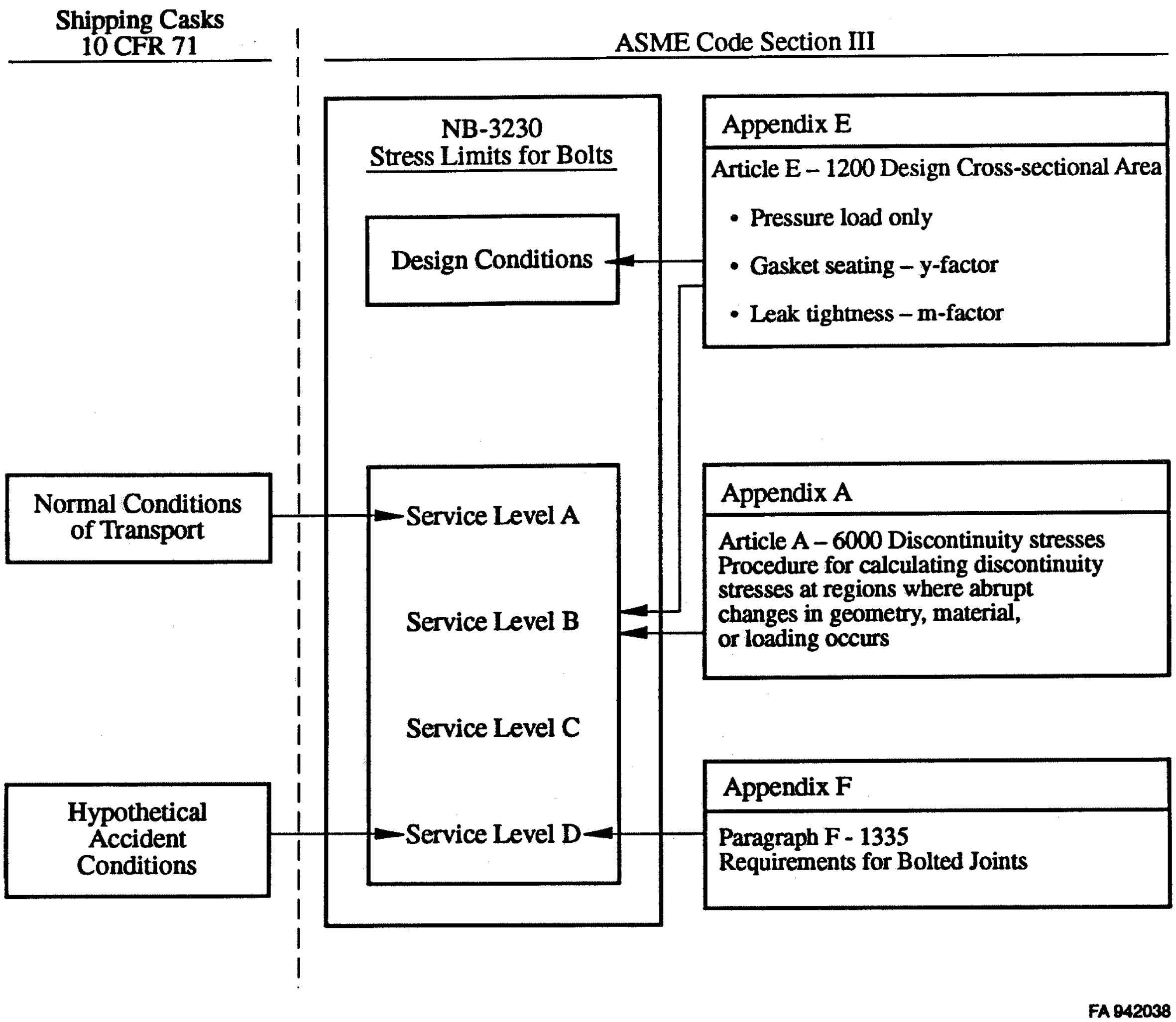


Fig. G4. Loading conditions of shipping casks and the corresponding ASME Cod requirement for Class 1 components.

for design conditions and four levels of service conditions. Also shown on this figure for comparison purposes are the corresponding test conditions under 10 CFR 71.

For a pressure load specified under ASME Code design conditions, the bolt cross-sectional area can be determined in accordance with Subarticle E-1200 of Appendix E of the Code. For service conditions, in addition to Subarticle E-1200, the bolts should also be designed for other loads. Discontinuity stresses at regions of abrupt changing geometry, material, or loading conditions (e.g., at the lid/cask wall interface in the case of shipping casks) should be considered in accordance with the principles described in Article A-6000, Appendix A of the Code. This article describes an acceptable analysis method based on the continuity requirements of displacement and angular rotation at regions of discontinuity.

The ASME stress limits of interest to shipping cask designers are included in NB-3232 for Level A service conditions (corresponding to the Normal Conditions of Transport for shipping casks) and F-1335 of Appendix F for Level D service conditions (corresponding to the Hypothetical Accident Conditions for shipping casks).

The proposed analysis criteria are also based on current knowledge of pressure vessels and structures.[24][25][26][27] This appendix on closure bolts reflects commonly accepted approaches and does not preclude cask designers from using other methods if they are based on reasonable assumptions and sound technical basis.

Figure G5 is a flow chart showing the steps involved in the design of closure bolts of a shipping cask. A sample problem following the steps described in the flow chart can be found in Appendix A of *Design and Analysis of the Closure Bolts of Shipping Casks*.[21]

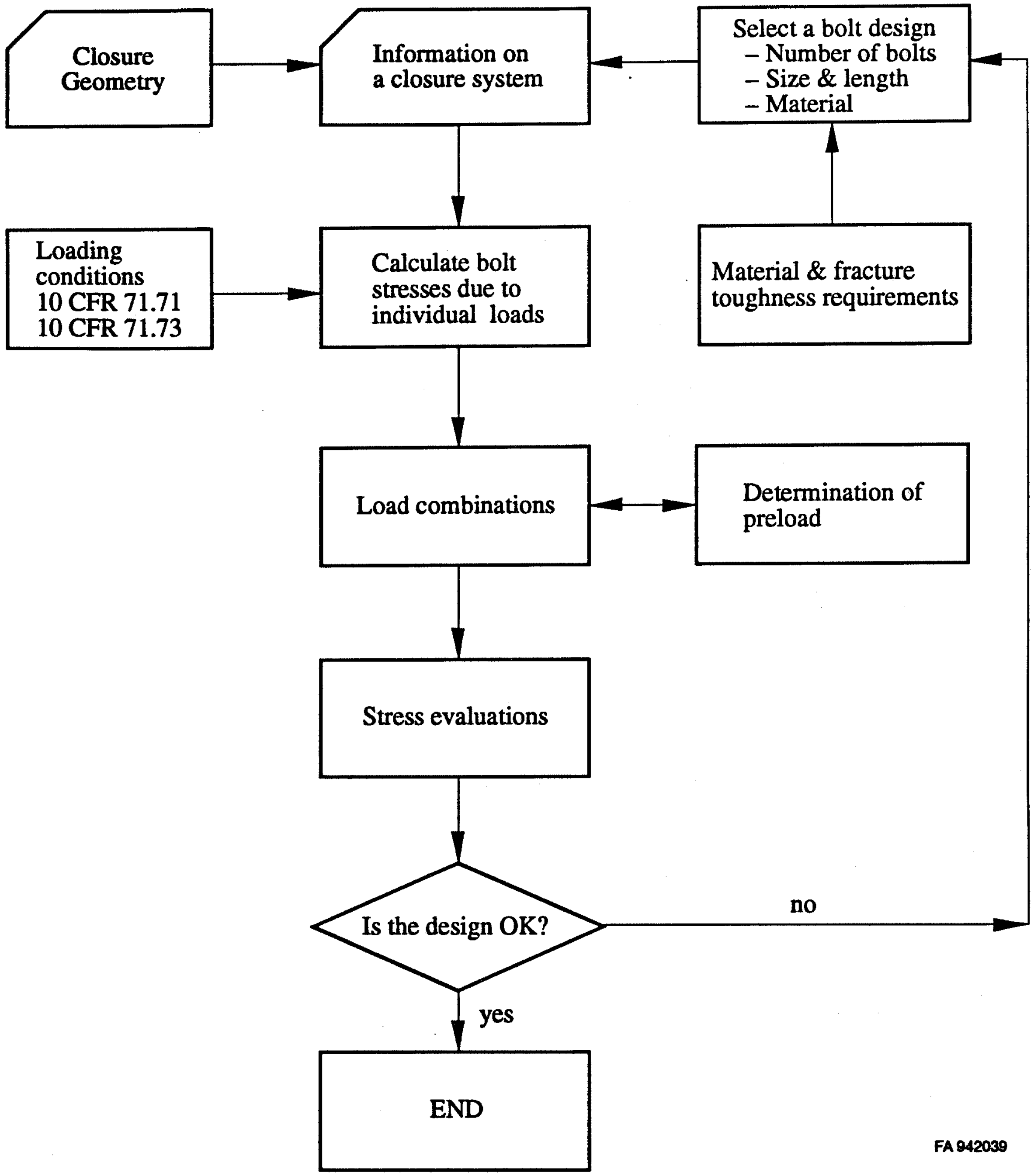


Fig. G5. Design procedures for the closure bolts of a shipping cask.

A typical bolt design starts with an initial selection of bolting material, bolt size, and number of bolts. Using the geometry of the closure, the bolt stresses are then calculated and combined based on the loading conditions specified in 10 CFR 71.71 and 71.73. The required preload can be determined by the maximum calculated bolt stresses and the sealing requirements of the shipping cask. Total stresses in bolts (including the preload) is then evaluated against the stress limits. The stress evaluation provides concrete information about the suitability of the initial design. Adjustments can be made if needed, and a new design can be evaluated. In most situations, one or two iterations is sufficient to obtain a good closure bolt design.

3.0 LOAD CONDITIONS

The functions of the closure bolts in shipping casks are to fasten the lid to the cask body and to maintain the effectiveness of the seals under both normal and accident conditions. The stresses of these bolts are influenced by the nature of the sealing system and the geometry of the closure joint. The bolt analysis will consider two types of seals. One is a relatively soft gasket whose sealing effectiveness is based on maintaining a minimum interface pressure between the flange and seal. The second is a self energizing seal (e.g., an elastomeric "O" ring) that depends upon internal pressure and the initial clamping pressure to force the seal against both flanges. For effective performance, the self-energizing seal must maintain metal-to-metal flange contact or limit the separation to a minimal amount.

For the relatively soft gasket seals, the preload must be large enough to accommodate additional loadings (which cause the bolts to stretch) without reducing the compression on the gasket below its minimum requirements. This is illustrated by a diagram that plots the load deflection characteristics of the bolt as it is loaded in tension (Fig. G6) and the members of the joint loaded in compression (Fig. G7).

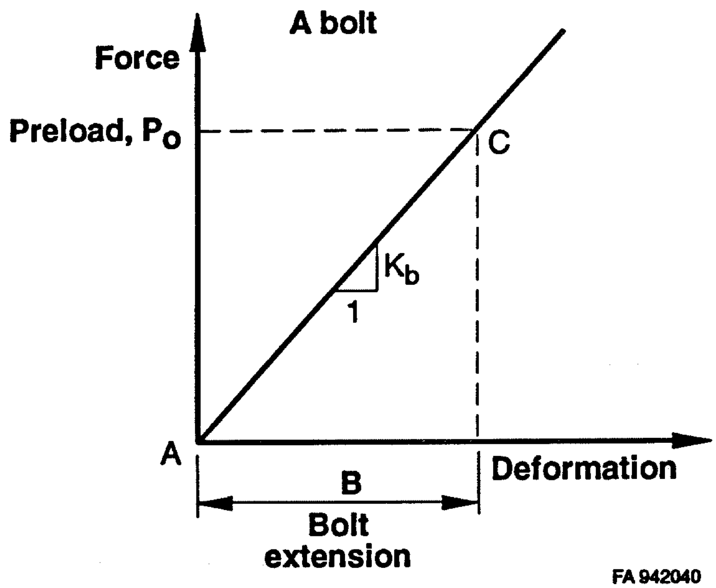


Fig. G6. Force-deformation curve of a bolt.

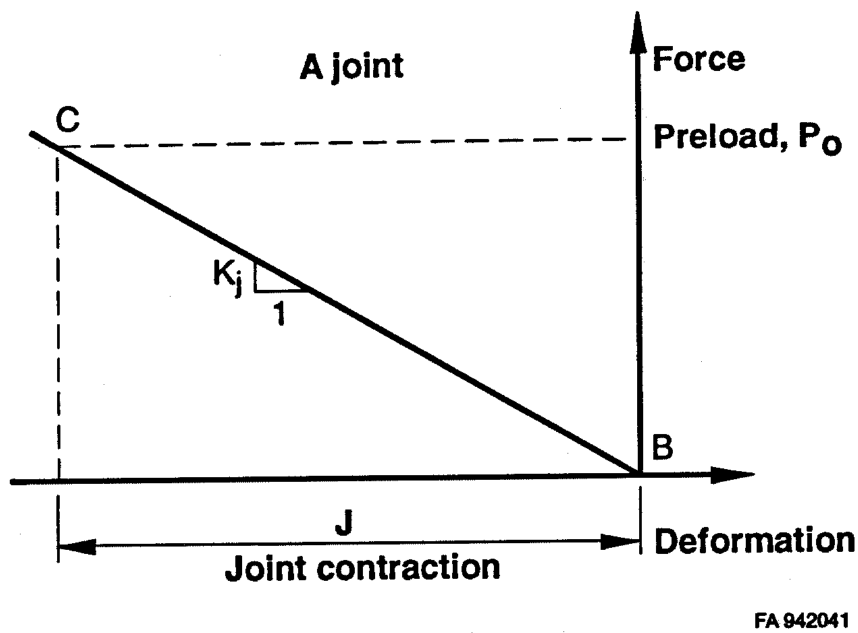


Fig. G7. Force-deformation curve of a joint.

This diagram showing the load-deflection behavior of the bolt and the joint is shown in Fig. G8, which is a combination of Figs. G6 and G7.

Because the stiffness of the metallic joint components is so much greater than that of the gasket, these components may be considered absolutely incompressible. Starting from zero bolt stretch at A and zero gasket compression at B (Fig. G8), the preload torque is applied until the required preload (P_0) at point C is achieved. The increase in bolt length is dB, and the gasket decrease in thickness is dJ. Note that the slopes of the load-deflection curves represent the spring constants of the bolt (K_b) and the gasket (K_j) respectively. The bolt and gasket deflections are

$dB = P_o/K_b$, and (G1)

$dJ = P_o/K_{j.}$

If an external load, F, is applied to the closure, which may be due to internal pressure or inertia loads, the bolt increases in tension from C to D. At the same time, the compression on the gasket decreases from C to E. The amount of additional bolt stretching (dB') is equal to the amount of relaxation in the gasket (or the joint) (dJ') that is,

$dB' = dJ' = dP_b/K_b = dP_j/K_j = F/(K_b+Kj).$ (G-2)

This external force is represented on the diagram (Fig. G8) by line DE. The design must be such that under the maximum anticipated force (F_{max} the residual gasket compression represented by point X is sufficient to maintain the seal. The residual compression in the gasket is

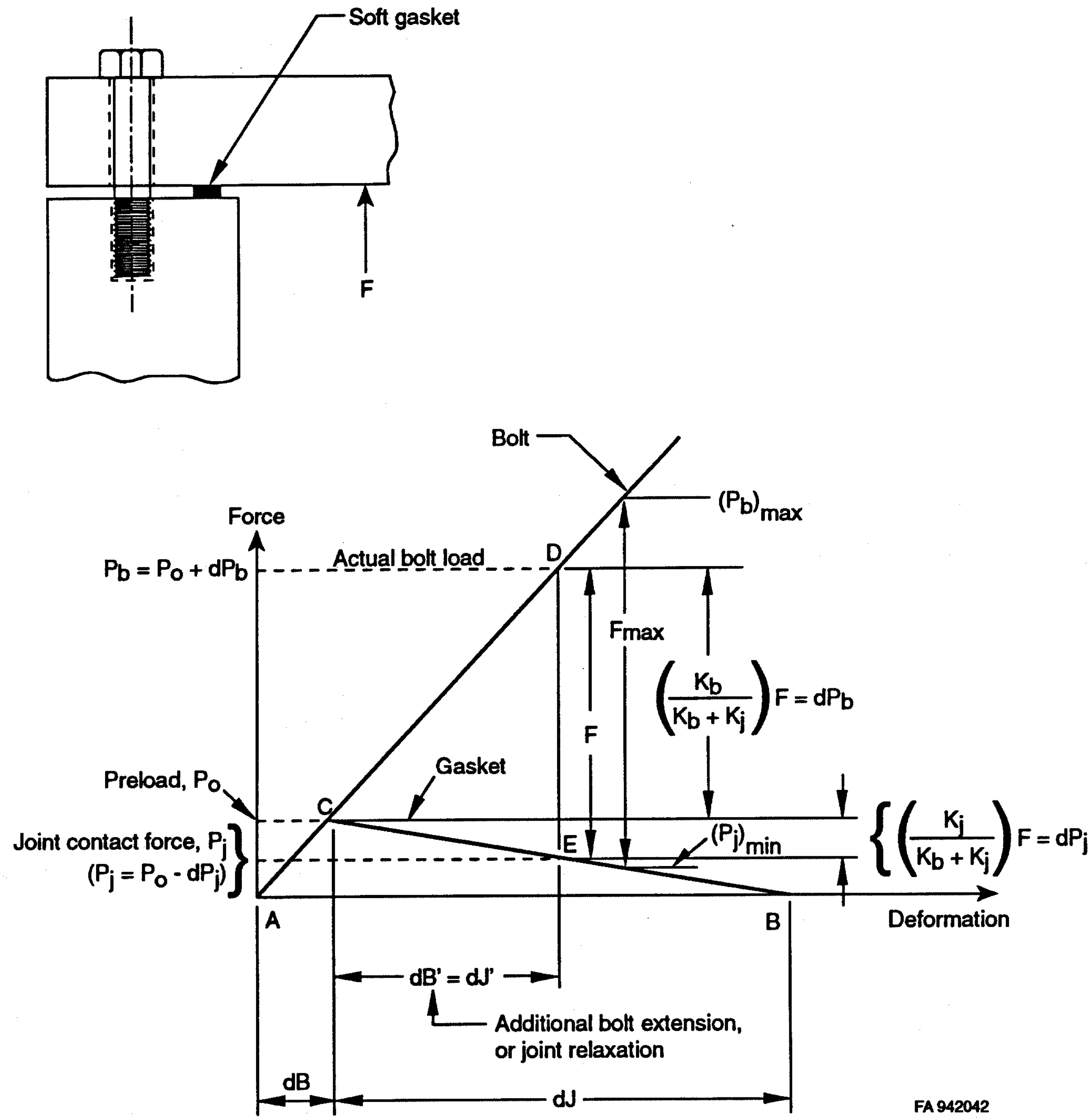


Fig. G8. Force-deformation characteristics of a joint with soft gasket seals.

$$P_j = P_o - dP_j, \text{ and} \tag{G-3}$$

$$(P_j)_{min} = P_o - (dP_j)max$$

where:

$$dP_j = F[K_j/(K_b+K_j)], \text{ and} \tag{G-4}$$

$$(dP_j)_{max} = F_{max}[K_j/(K_b+K_j)].$$

Rules for determining the minimum required gasket compression are given later. Free body diagrams of the joint are shown in Fig. G9 to illustrate the external load and the interface forces.

The load on the bolt is:

$$P_b = P_o + dP_b, \text{ and} \tag{G-5}$$

$$(P_b)_{max} = P_o + (dP_b)_{max}$$

where:

$$dP_b = F[K_b/(K_b+K_j)] \text{ , and} \tag{G-6}$$

$$(dP_b)_{max} = F_{max}[K_b/(K_b+K_j)].$$

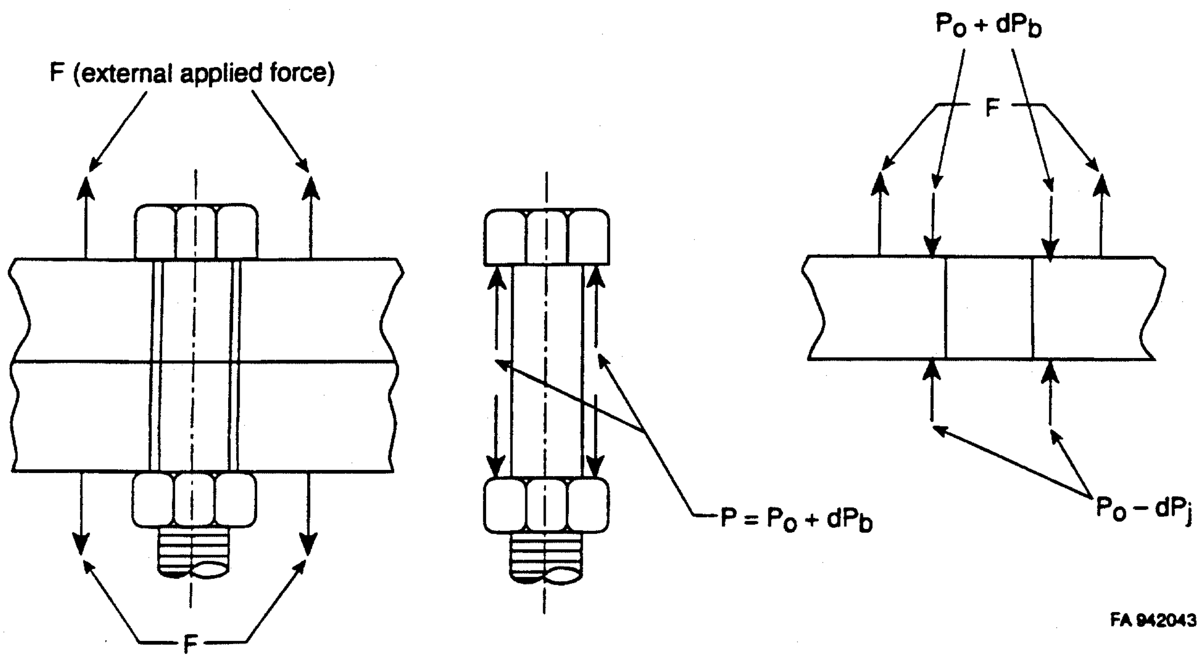


Fig. G9. Free-body diagrams of the bolt and the joint.

Equation (G-6) can be used to compute the maximum uniform tensile stress. Note that, from Equations (G-3) and (G4), the preload must be at least

$$P_o > (P_j)_{min}, + F_{max}[K_j/(K_b + K_j)]. \qquad \text{(G-7)}$$

For joints with a self-energizing seal, the preload must be such that, for the maximum load, F_{max}, the metallic surfaces will still be in contact. This is illustrated in the load-deflection diagram shown in Fig. G10

As it does with the gasket joint, preload stretches the bolt from A to C and compresses the metallic joint components from B to C. Note that the stiffness of the metallic joint component is about the same order of magnitude as the stiffness of the bolts. Upon application of external forces, the bolt stretches further to D, while the flange interface pressure relaxes toward E. The maximum external force that may be applied is that which results in zero flange interface pressure. Any additional load beyond this point will cause separation of the flanges that could compromise the seal. For zero interface compression,

$$0 = P_o - F_{max}[K_j/(K_b+K_j)].$$

The minimum preload required to maintain a seal is therefore determined by

$$(P_o)_{min} = F_{max}[K_j/(K_b + K_j)]. \qquad \text{(G-8)}$$

The preload is equivalent to the maximum external load if the joint components are infinitely stiff relative to the bolt. If this is not so, the preload must be made less than the maximum external force if the bolt is not to be overstressed when the maximum external load is applied.

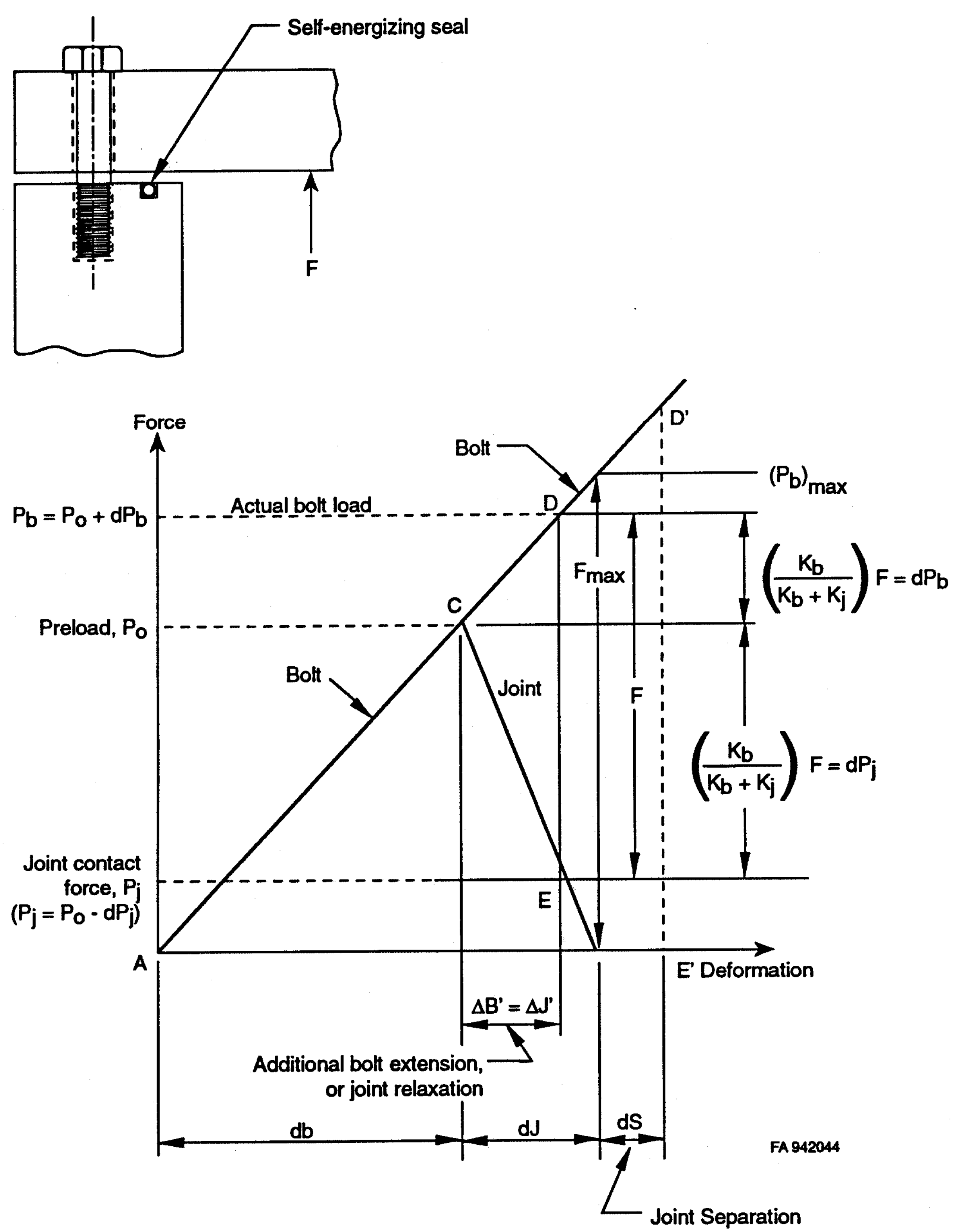


Fig. G10. Force-deformation characteristics of a joint with self-energizing seals.

The load-deflection diagram described in previous sections is commonly called the "joint diagram."[22,24,26] Based on the joint diagram, the bolt load (P) can be calculated for a given preload (P_o) and a given external load or force (H).

Without separation of contact surfaces ($H < P_o[K_b+K_j]/K_j$)

$$P = [K_b/(K_b+K_j)]H + P_o. \tag{G-9}$$

With separation of contact surfaces ($H > P_o[K_b+K_j]/K_j$)

$$P = H \tag{G-10}$$

and

$$d_S = [H-P_o(K_b+K_j)/K_j]K_b \quad \text{and} \tag{G-11}$$

$$= \text{the amount of joint separation.}$$

The spring constant (K_b) can be calculated easily by considering the bolt as a one-dimensional element with several sections of different cross-sectional area in series,

$$K_b = 1/(L_i/E_bA_i)\ , \tag{G-12}$$

where E_b is the Young's modulus of the bolt material, A_i is the cross-sectional area, L_i is the length, and the subscript *i* represents various regions of a bolt as described earlier and Fig. G3.

Several formulas for calculating K_j are available and they give a wide range of results. However, the following simple formula has been verified both in laboratory experiments and by the finite-element method of analysis.[28][29][30] It is appropriate for our purpose of applying simple methods to analyze the closure systems of shipping casks shown in Fig. G1.

$$K_j = E_j A_c / L_j \tag{G-13}$$

where:

E_j = Young's modulus of the joint material,

A_c = equivalent cross-sectional area of the joint,

= $(\pi/4)\ [(D_c + L_j/10)^2 - D_h^2]$

L_j = length of the joint

D_c = diameter of the contact area between the bolt head and the joint,

D_h = diameter of the bolt hole.

The above formula along with the method of calculating bolt load presented earlier was also used by the Society of German Engineers in a design handbook.[31] Motosh proposed an even more accurate, but more lengthy, formula for calculating the equivalent cross-sectional area.[29]

For joints in which a soft gasket separates two metal parts (or without metal-to-metal contact), the joint stiffness K_j is small and can be assumed to be zero. According to Equation (G-9) and Fig. G8, any applied external force will be carried almost completely by the bolts.

10 CFR 71.71 and 10 CFR 71.73 specify all the loading conditions that a shipping cask should be designed for under the Normal Conditions of Transport and the Hypothetical Accident Conditions. In stress calculations, the loads or stresses on a shipping cask can generally be grouped into the following categories in accordance with the source of these loads:

- preload tension and torsion,
- pressure loads,
- thermal loads,
- impact loads,
- vibration and shock loads,
- puncture loads and,
- fabrication stresses.

The major functions of the preload in closure bolts are to join the contact surfaces and to provide an environment in which the gaskets will function properly so that cask containment or dispersal requirements of 10 CFR 71.51 are met. Preload is also helpful in preventing vibration loosening, fatigue failure, and joint slippage.

The bolts can be tightened using tools such as torque wrenches, bolt tensioners, or bolt heaters. Depending on the tools used to tighten the bolts, there are several control methods for achieving the desired amount of preload. These control methods are torque control, turn control, control by a combination of torque and turn, and stretch control. To ensure that the minimum preload required to provide leak-tight closure is achieved, it is important to consider uncertainties associated with all factors affecting the actual preload that may exist in the bolt.

The most common method of controlling preload is torque control, in which a torque wrench is used to tighten a bolt. The input torque required to tighten a bolt to a specified preload is dependent on the friction between the internal and external threads, the slope of the inclined plane of the threads, and the friction between surfaces of the bolt and the joint (head, nut, washer, and joint). The input torque can be derived mathematically in a long-form formula from the pitch of the threads and the coefficients of friction between various contact surfaces. However, it is convenient to use the following short-form formula to estimate the preload.

$$T = kDP_o, \tag{G-14}$$

where:

T = input torque,

D = nominal diameter of the bolt,

P_o = preload, and

k = the "torque coefficient" or the 'nut factor".

Using an average coefficient of friction of 0.15 between all surfaces for standard screw threads, the torque coefficient (k) was found to be approximately equal to 0.2, based on mathematical derivations.[24,26]

The torque coefficient k can also be determined experimentally, and it can be found in many reference books. Bickford and Looram compiled a table based on the types of bolt lubrication.[25] In their table, the value of k varies from 0.095 to 0.3. Bickford also constructed a histogram of reported k-values in the literature and he found that the k-value of as-received steel fasteners has a mean value of

0.199 and a three-standard-deviation of 0.05[22] The mean value is very close to the mathematically derived value of 0.2 described above. With plus or minus three standard deviations, k varies from 0.15 to 0.25, a scatter of around plus and minus 25 percent.

When a torque wrench is used to tighten a bolt, a residual torsional stress exists in the bolt after the tightening is completed. This residual torsional stress is due to the frictional residence between the threads of the bolt and the joint, and between the bolt head and the associated surface of the joint. Because of joint relaxation, this residual torsional stress drops significantly after the bolt is tightened, and the stress is usually not a concern unless the frictional coefficients are high. An upper bound value of this residual torsional stress can be estimated based on the frictional coefficients of the contact surfaces described above and the pitch of the threads. If it is suspected that the frictional coefficients between the contact surfaces are high, the torsional stress in the bolts should be calculated and evaluated in accordance with the stress limits presented later.

Casks experience pressure loads due to the differential pressure between the inside and outside of the cask containment. Initial internal pressure can be caused by prepressurization, cask temperature increase due to the decay heat of contents, and any gas leakage from contents. Reduced and increased external pressures are part of the loads specified for the Normal Conditions of Transport.

For the circular, cylindrical shipping casks considered in this study, as shown in Fig. G1, the closure bolts are evenly spaced along a circular circumference. Under the differential pressure load, the interface pressure at the mating surfaces of the lid and the cask wall is not uniform in the radial direction of the cask. This nonuniform distribution is caused by the rotation of the joint due to the bending of the lid and the cask wall under differential pressure.

Bolt tension due to the interface bending moment at the mating surfaces between lid and cask wall is usually not significant. Tension due to the uniform portion of the interface force can be calculated as follows:

$$P = pA/N, \qquad \text{(G-15)}$$

where:

p = differential pressure inside and outside the containment, (only pressure differential causing bolt tension is considered),

A = area inside the gasket circle, and

N = number of bolts.

The pressure load is an external load on the bolts. The actual increase in bolt tension due to the differential pressure load can be calculated using Equation (G-6).

For closures without metal-to-metal contact between the lid and the cask wall, the only interface force is the pressure force on the gaskets. There is no bolt tension attributable to bending because there is no interface bending moment between the lid and the cask wall. Equation (G-15) represents the total bolt tension under the differential pressure load in this case.

Thermal analysis consists of heat transfer analysis and thermal expansion stress analysis. Two thermal stress conditions exist. One is caused by the differential thermal expansion due to the hypothetical fire condition in which a radiation environment of 1475 degrees Fahrenheit is assumed. The other is caused by the steady state temperature distributions in the cask under the Normal Conditions of

Transport. If differential thermal expansion exists between the lid and the cask body in the radial direction of the cask, shear load on the bolts may occur.

There are two sources of thermal stress. One is the axial stress in the bolt due to differential thermal expansion of the bolt and the lid and cask wall in the axial direction of the bolt. The other source is the shear and tensile loads on bolts due to differential thermal expansion of the lid and the cask wall in the radial direction of the cask.

The axial bolt load in a bolt due to temperature change in the bolt and its surrounding environment can be calculated as follows[21]:

$$P = K_b K_j (a_j - a_b) L d_T / (K_b + K_j), \qquad \text{(G- 16a)}$$

where:

P = axial bolt load due to differential thermal expansion of the bolt and the lid and cask wall in the axial direction of the cask,

L = Lg (approximate),

a_j = Coefficient of thermal expansion of the joint,

a_b = Coefficient of thermal expansion of the bolt,

K_j = Stiffness of the joint,

K_b = Stiffness of the bolt, and

d_T = Change in temperature from initial temperature.

The bolt load calculated with Equation (G-16a) can be added directly to other loads in the load combinations to calculate the total bolt load without using Equation (G-6) because the local stiffness of the bolt and the lid and cask wall have already been taken into consideration.

A simple and conservative formula can be obtained from Equation (G-16a) by assuming that the joint is rigid, i.e.,

$$P = (a_j - a_b) d_T E_b A_b, \qquad \text{(G- 16b)}$$

where A_b and E_b are the cross-sectional area and the modulus of elasticity of the bolt, respectively. Note that, in Equation (G-16b), there is no need to calculate K_b and K_j.

Without finite-element analyses, it is not feasible to calculate accurately the bolt stresses due to free drops of a shipping cask onto an unyielding surface. However, simplified approaches are possible if conservative assumptions are made.

In an end drop, the lid experiences the deceleration force of the lid and the cask contents. The lid also experiences an external pressure load due to the interface force between the lid and the impact limiter. The amount of tension force on the bolts can be calculated by considering all the pressure loads on the lid, as shown in Fig. G11.

Several assumptions are made:

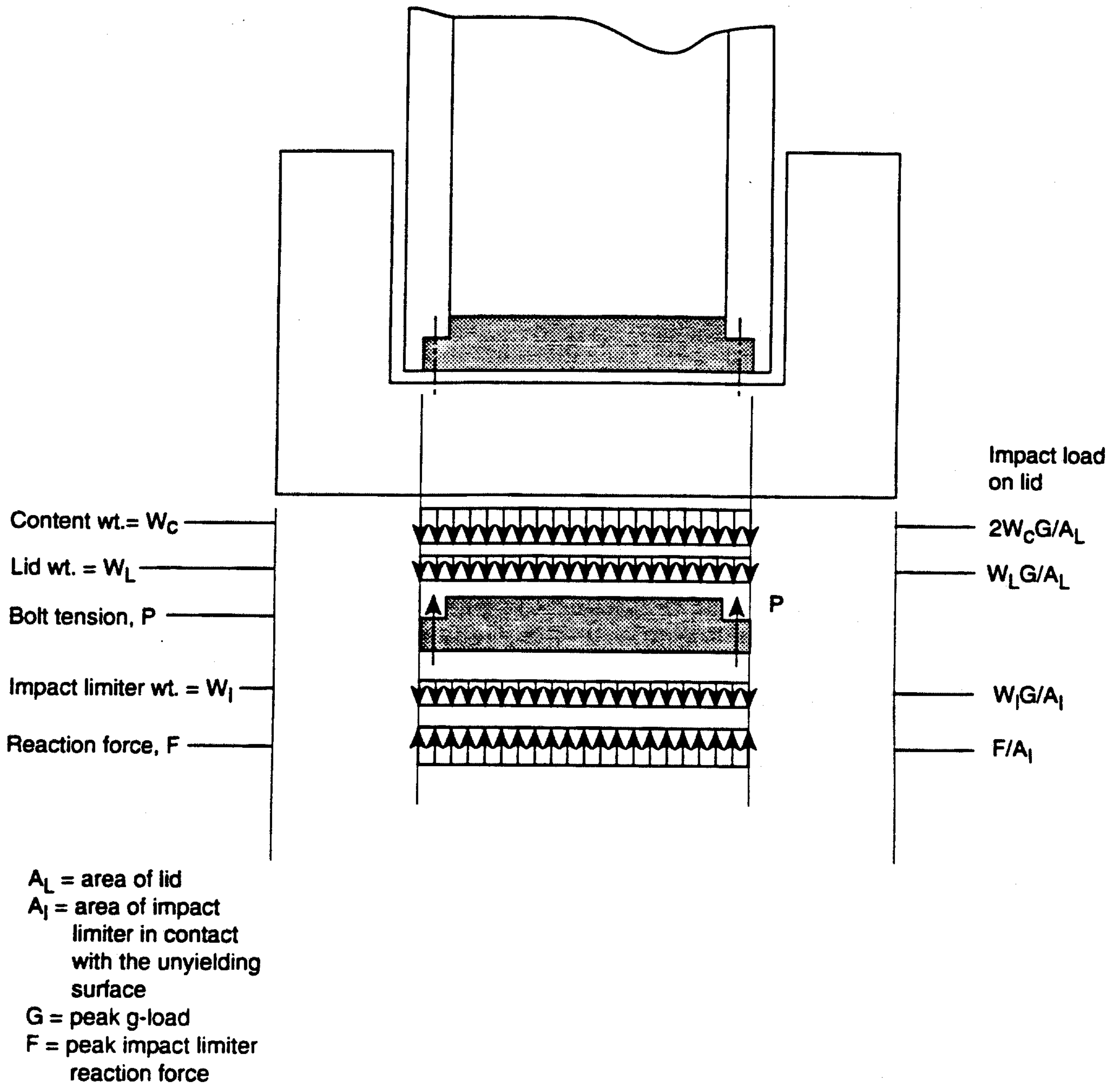


Fig. G11. Calculation of peak bolt tension during end drop.

- It is assumed that all loads on the lid are uniformly distributed. This includes the interface forces between the cask and the impact limiter and between the impact limiter and the unyielding surface. The assumption of uniform interface pressure between cask and impact limiter is reasonable because the impact limiter experiences uniform axial deformation in this region.

  The interface pressure between the impact limiter and the unyielding surface will be larger in the central region inside the cask diameter than in the region outside the cask diameter. The assumption of uniform interface pressure between the impact limiter and the unyielding surface results in less external pressure being applied to the lid in the central region. This external pressure load is in the opposite direction to the deceleration loads of the contents and the lid, which cause tension in the bolts. By assuming a smaller interface pressure between the impact limiter and the unyielding surface in the central region, a higher, or conservative, bolt tensile stress will be obtained.

- The peak impact limiter reaction force is assumed to occur at the same time as the peak deceleration force of the cask.

- The cask contents (including the basket) are assumed to have a dynamic amplification factor of 2 - the maximum amplification of a single degree of freedom system—because no dynamic analysis of the basket and cask contents is made.[32]

With the above assumptions, the pressure loads on the lid due to the impact can be determined using the results of a finite-element impact analysis—the peak deceleration loads of the cask and its contents and the peak impact limiter reaction force. The maximum axial stress in the bolts can be calculated following the same approach described earlier for the pressure load.

Note that the bolts do not experience a transverse shear force during the cask end drop.

During a cask side drop, the bolts do not experience axial tension. However, due to the deceleration of the lid and the lateral component of the impact limiter reaction force, a shear load on the bolts may exist. Figure G12 shows the assumed interface force and pressure distributions between the impact limiter and the unyielding surface and between the cask and the impact limiter. Again, conservative assumptions are used to calculate the total shear force on the bolts.

For a protected type of closure (see Fig. G1), the impact limiter reaction force is totally carried by the cask wall. The only load on the bolts is the deceleration of the lid. It is assumed that the shear force is evenly distributed among the bolts because the enplane stiffness of the lid is very large and the bolts are experiencing almost the same amount of shear deformation regardless of their distance from the point of impact.

For the exposed closure, two loading conditions are considered, and the higher bolt stress from one of these two cases is used for design. The first case assumes that the impact limiter force is completely carried by the cask wall. This results is the same loading condition that occurs for the protected closure (Fig. G12). The only load on the bolts is again the deceleration of the lid. The second case (Fig. G12) assumes that the impact limiter force outside the interface plane between the lid and the cask wall is transmitted to the lid and, therefore, to the bolts. Again, the shear force is assumed to be evenly distributed to all the bolts due to the large in-plane stiffness of the lid.

For oblique impacts in free drops, the closure bolts experience both axial and transverse shear forces. Only axial force exists in end drops, and only shear force exists in side drops. The method for calculating tension in the bolts due to the axial component of the deceleration force for the circular,

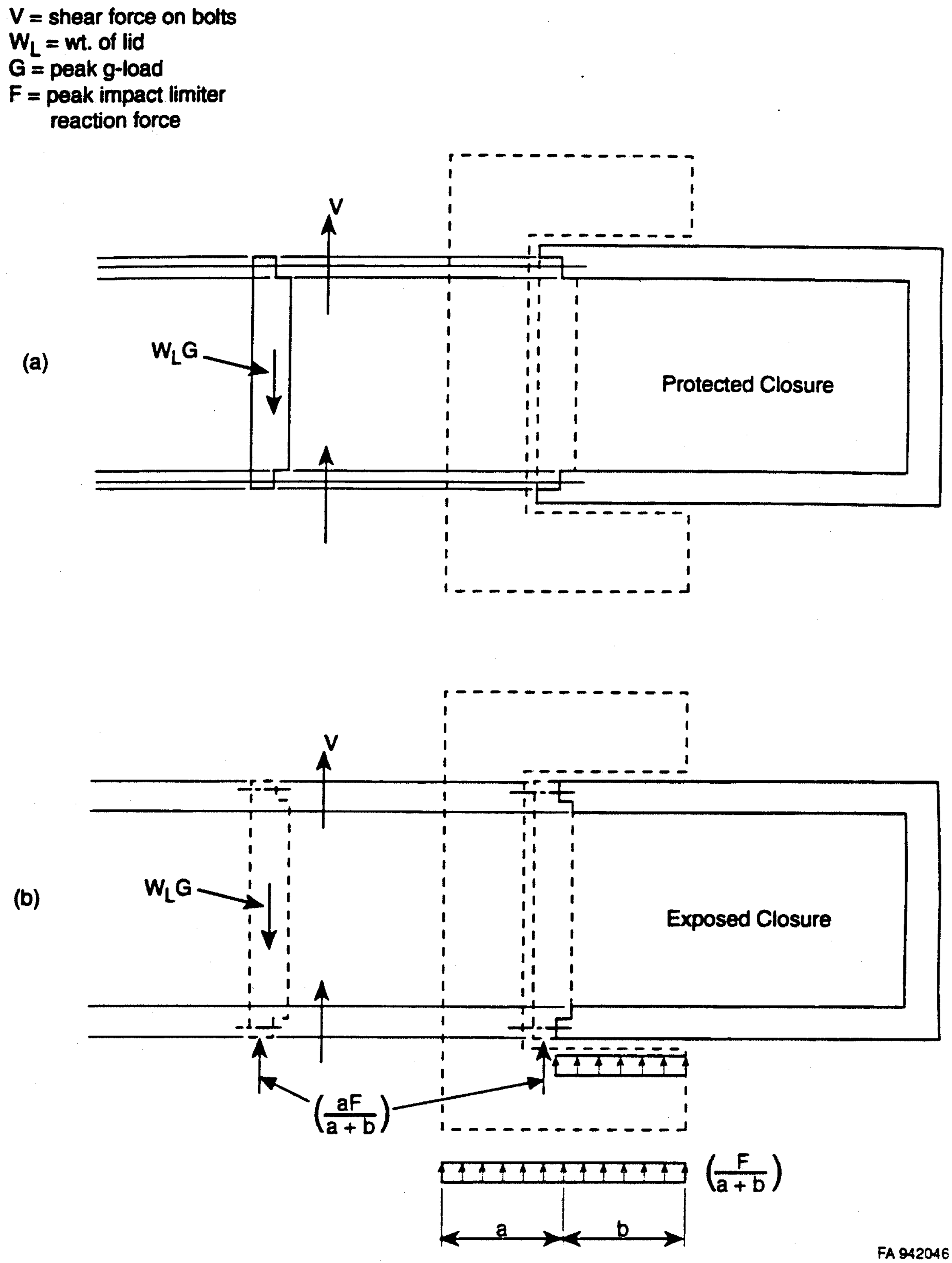


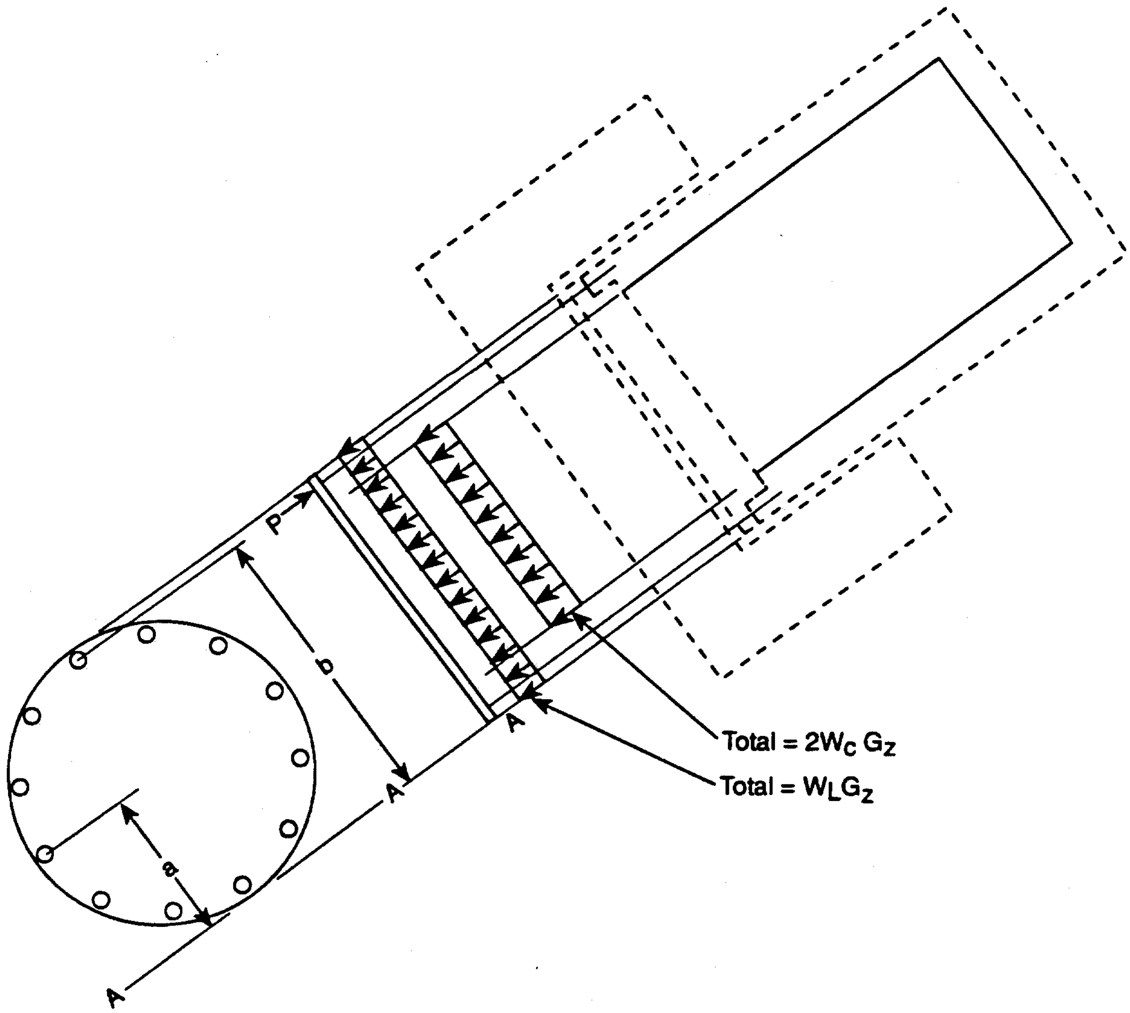
Fig. G12. Calculation of peak bolt shear force during side drop.

cylindrical shipping casks is developed for SCANS.[33] This method of calculating the bolt tension is summarized in Fig. G13. The method is based on the assumption that the lid and the cask wall are rigid and the lid rotates about a pinned boundary at the edge at the lowest point of the lid. The bolt tensile stress, S, can be calculated based on the area moment of inertia of the bolt pattern, I, and the distance, c, of the bolt of interest to the pinned boundary using the formula $S = Mc/I$. The moment (M) is due to the deceleration loads of the lid and cask contents about the pinned boundary. The bolt load (P) due to impact is simply equal to the product of the stress (S) and the bolt area (A_b). The peak deceleration load of the cask obtained from an impact analysis should be used in calculating bolt stresses. Also, the basket and the cask content should be assumed to have an amplification factor of 2 if their stiffness are not considered in the dynamic analysis of the cask.

The transverse component of the impact limiter force is assumed to be transmitted to the lid and to produce a transverse shear force on the bolts, as shown in Fig. G14. Again, two loading cases are considered in the same manner as for the side drop with an exposed closure. The transverse shear force is also assumed to be evenly distributed to all the closure bolts regardless of their relative position on the closure.

Based on the approach described above, the bolts experience the same amount of transverse shear, and the bolt with the largest distance from the point of impact receives the maximum tensile force. Therefore, the stresses on the bolt farthest from the point of impact should be used in the design of closure bolts.

It is unlikely that the puncture test condition described in 10 CFR 71.73 will cause any significant bolt tension and shear. Figure G15 shows the puncture condition of a shipping cask with an exposed lid. The worst condition is that the puncture pin penetrates the impact limiter and is compressed far



$$P = \left[\frac{(2W_C + W_L) G\ a\ b}{I_A} \right] A = \text{peak bolt tension}$$

I_A = moment of inertia about A-A
A_b = bolt area (least cross-section)
W_C = wt. of content
W_L = wt. of lid
G_Z = axial component of deceleration g-load

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Fig. G13. Calculation of peak bolt tension during oblique drop.

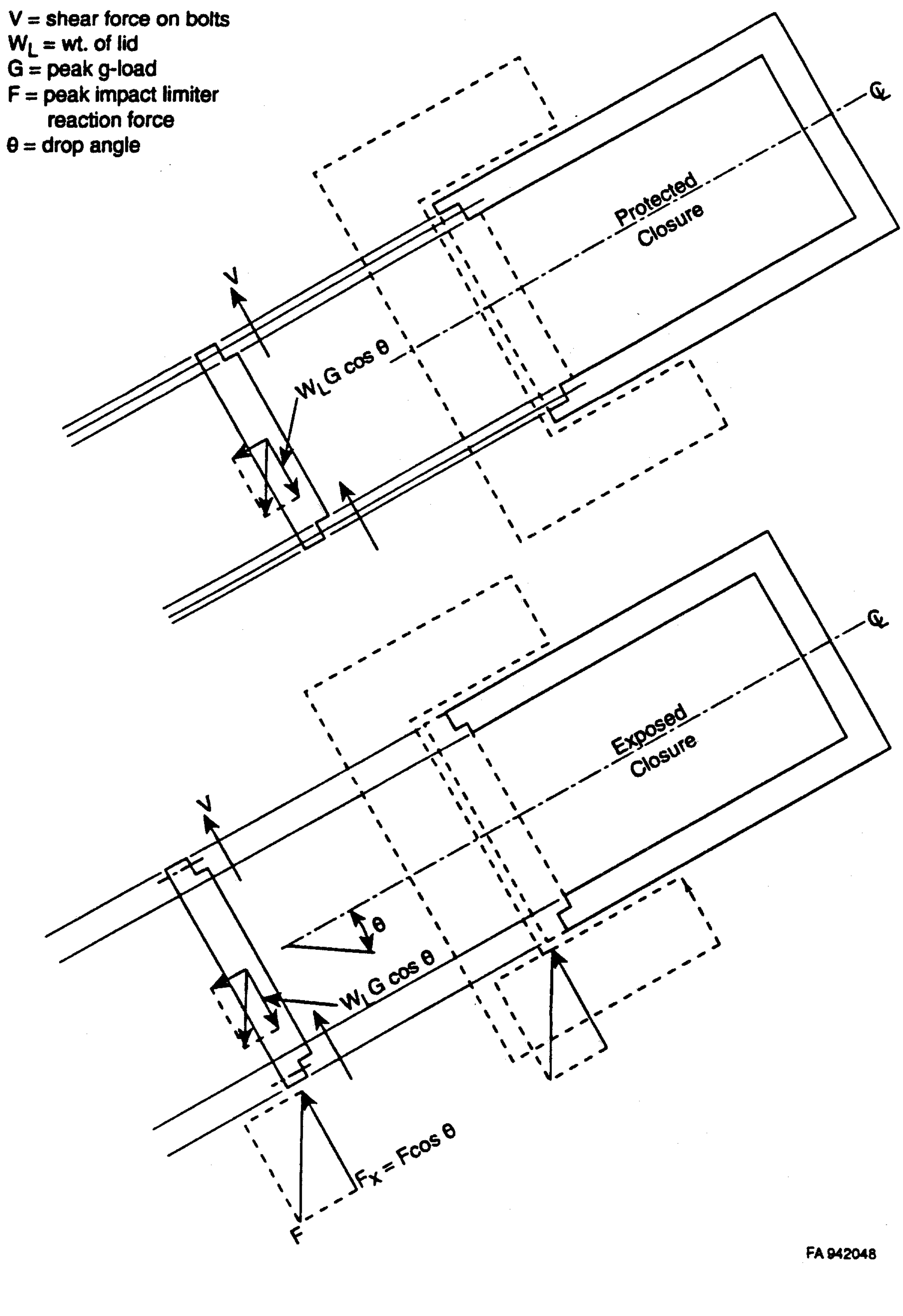

Fig. G14. Calculation of peak bolt shear force during oblique drop.

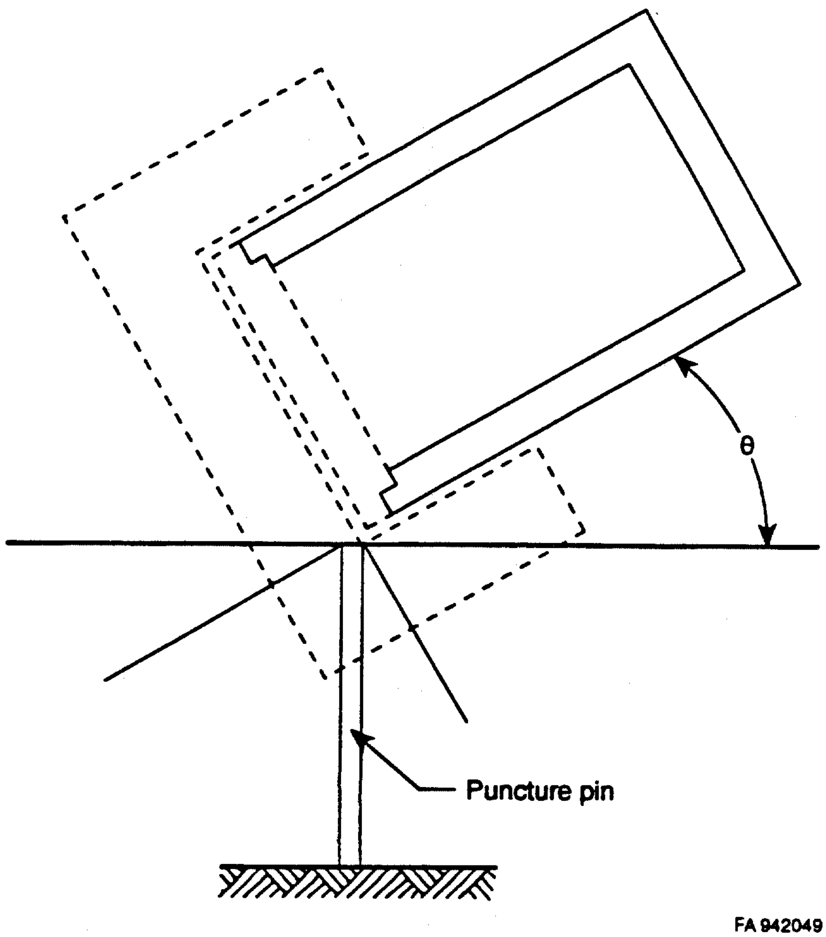

Fig. G15. Shear load on the bolts of an exposed closure under puncture condition.

beyond the yield strength of the material without buckling. The transverse shear stress (S_b) in a bolt for this worst condition can be calculated as follows:

$$A_p S_p \cos\Theta = N A_b S_b, \tag{G-18}$$

where:

A_p = cross-sectional area of a 6-in.-diam puncture pin

= $\pi \times (3)^2 = 28.3$ in.^{2},

S_p = maximum stress in a mild steel puncture pin that can be reached without buckling of the pin, = orientation of the cask with respect to a horizontal surface,

N = number of bolts,

A_b = area of a bolt, and

S_b = transverse shear stress in a bolt.

For a typical mild steel, the tensile strength is usually on the order of 60 kilo-pounds per square inch. Assuming that S_p reaches the tensile strength without causing buckling in the puncture pin and assuming also that 30 bolts of 1.25-inch-diameter are used, one obtains the following:

$$S_b = 46.1 \times \cos\Theta \text{ ksi}. \tag{G-19}$$

The yield strength of bolts is typically over 100 kilo-pounds per square inch. Assuming allowable shear strength of $0.6S_y$, the above calculated transverse shear stress in a bolt is still within the allowable stress of 60 kilo-pounds per square inch (0.6 x 100) even if $\cos\Theta = 1$ is assumed. Bolt stresses due to pressure and thermal conditions during puncture test condition are usually small, and they are therefore neglected in the above calculation.

It is unlikely that the closure bolts of shipping casks will experience any significant vibration and shock loads under Normal Conditions of Transport. Preload is useful for the bolts to resist any vibration loads.

Table G1 summarizes the load combinations applicable to the design and analysis of the closure bolts of a shipping cask. This table, which follows Regulatory Guide7.8,[34] lists only those loads considered to be of significance in the design of closure bolts. The bolt load associated with minimum internal pressure is not considered in all load combinations because these pressure loads will not induce tension in the closure bolts.

4.0 DESIGN VALIDATION METHODS

Leakage requirements for shipping casks are described in 10 CFR Part 71. Leak-proof containment is defined in this report as leakage within the limits specified in 10 CFR 71.51. Leak-proof containment is required at all times. There should be no loss or dispersal of radioactive contents, as demonstrated to a sensitivity of $10^{-6}A_2$/hours under Normal Conditions of Transport. Under the Hypothetical Accident Conditions, there should be no escape of krypton-85 exceeding 10,000 Ci in one week, and no escape of other radioactive material exceeding a total amount A_2 in one week.

The bolt preload (P_o) is an essential element in making shipping cask seals effective. Of course, in situations where the cask content is nondispersible and leak-proof packaging is not required, little preload is required.

In casks without metal-to-metal contact, the seals or the gaskets usually have low stiffness because gasket material is soft compared to the metal parts, or the gasket surface is small, or both.

Table G1. Summary of load combinations for the Normal Conditions of Transport and the Hypothetical Accident Conditions

	Preload and fabrication stress	Initial temperature Max.	Initial temperature Min.	Internal pressure Max.	Internal pressure Min.
Normal Conditions of Transport					
Hot environment – 100°F ambient temperature	x			x	
Cold environment – −40°F ambient temperature	x				
Reduced external pressure – 0.25 atm	x	x		x	
Free drop – 1 ft					
End drop	x	x		x	
End drop	x		x		
Oblique – max. tension	x	x		x	
Oblique – max. tension	x		x		
Oblique – max. shear	x	x		x	
Oblique – max. shear	x		x		
Side drop	x	x		x	
Side drop	x		x		
Hypothetical Accident Conditions					
Free drop – 30 ft					
End drop	x	x		x	
End drop	x		x		
Oblique – max. tension	x	x		x	
Oblique – max. tension	x		x		
Oblique – max. shear	x	x		x	
Oblique – max. shear	x		x		
Side drop	x	x		x	
Side drop	x		x		
Thermal – fire accident	x			x	

Therefore, the joint stiffness is usually much smaller than that of the bolt. In this case, Equation (G-9) can be conservatively approximated by

$$P = [K_b/(K_b + 0)]H + P_o$$

or

$$P = H + P_o. \tag{G-20}$$

Equation (G-20) is used in Subarticle E-1200 of ASME Code, Section VIII Appendix E for determining bolt cross-sectional area for leak tightness. P is the minimum required design bolt load. Applying E-1200 criteria to shipping casks, H is the maximum total external load on a bolt for all cases of load combinations (excluding, of course, the preload) described earlier. P_o is the minimum preload required for proper sealing of shipping casks.

$$Po = A_{gst}mp \tag{G-21}$$

where:

A_{gst} = 6.28bG,

p = $H/(0.785G^2)$

p = the maximum equivalent differential outward pressure of the cask for all load combination cases,

b = effective gasket or joint contact surface seating width, in.,

G = diameter (in.) at the location of gasket load reaction, and

m = ratio of the gasket pressure needed to ensure leak tightness and the maximum equivalent differential pressure p.

For a given gasket, a minimum initial bolt load W is needed to seat the gasket or to contact joint surfaces properly.

$$W_m = A_{gst}\ y/2 \qquad \text{(G-22)}$$

where y is the minimum design seating stress. The values of y and m can be found in Appendix E of the ASME Code, Section VIII for many materials and types of gasket.

The preceding paragraph describes gasketed closure in which the lid and the cask wall have no direct contact (or closures without metal-to-metal contact). For the metal-to-metal contact type of closures, seals are placed in grooves, which are machined into the lid. The lid and the cask wall are in direct contact with each other. The dimensions of the grooves are specified by the seal manufacturers to provide predetermined seal deformation to achieve proper seal compression. The dimensions of the grooves in which the seals are placed should follow the manufacturer's specifications. The dimensional specifications of these grooves can be considered as the gasket seating requirement for closures with metal-to-metal contact between the lid and the cask wall.

It is possible that seals may still be able to maintain leak tightness if a small amount of separation between the lid and cask wall occurs. Equation (G-11), which considers the lid-cask wall separation, can be rewritten as follows.

$$P_o = (H-K_b d_s)[K_j/(K_b+K_j)]. \qquad \text{(G-23)}$$

In Equation (G-23), P_o is the required preload and d_s is the maximum separation between the lid and the cask wall without a leakage through the seal. A great deal of effort may be needed to determine the allowable value of d_s for a given seal so that the dispersal limits of 10 CFR 71.51 can be met. For convenience and public safety, it is reasonable and conservative to assume that d_s equals zero. That is, metal-to-metal contact must be maintained at all times for this type of seal unless it can be demonstrated with considerable certainty that separation will not result in the dispersal of radioactivity exceeding 10 CFR 71.51 limits. Without establishing the allowable separation between the lid and the cask wall, metal-to-metal contact ($d_s = 0$) can be used as the criterion for determining preload in accordance with Equation (G-23). Not following this criterion does not imply a poor package design because this criterion is conservative as stated above.

Another reason for using metal-to-metal contact as the criterion for preload determination is that uncertainties exist in calculating bolt load due to various loading conditions and in determining and applying preload. By adopting a conservative criterion for calculating the required preload there is less concern with these uncertainties.

Since the container must not leak after an accident, the bolt stresses which are either average or at the extreme fibre, should be kept within elastic range at all times (including Hypothetical Accident Conditions) to ensure that preload is not lost as a result of permanent plastic deformation in bolts.

Using the joint diagram and the condition that no loss of metal-to-metal contact occurs, the minimum required preload can be calculated as shown in Fig. G16 where the external load H (vertical dashed line D'E') is at the right corner of the joint diagram (AB'C'). At this location, the amount of contact force and the amount of separation are both equal to zero. The minimum preload ($(P_o)_{min}$) required for no loss of lid/cask wall contact at the location of the bolts is equal to $H[K_j/(k_j+K_b)]$.

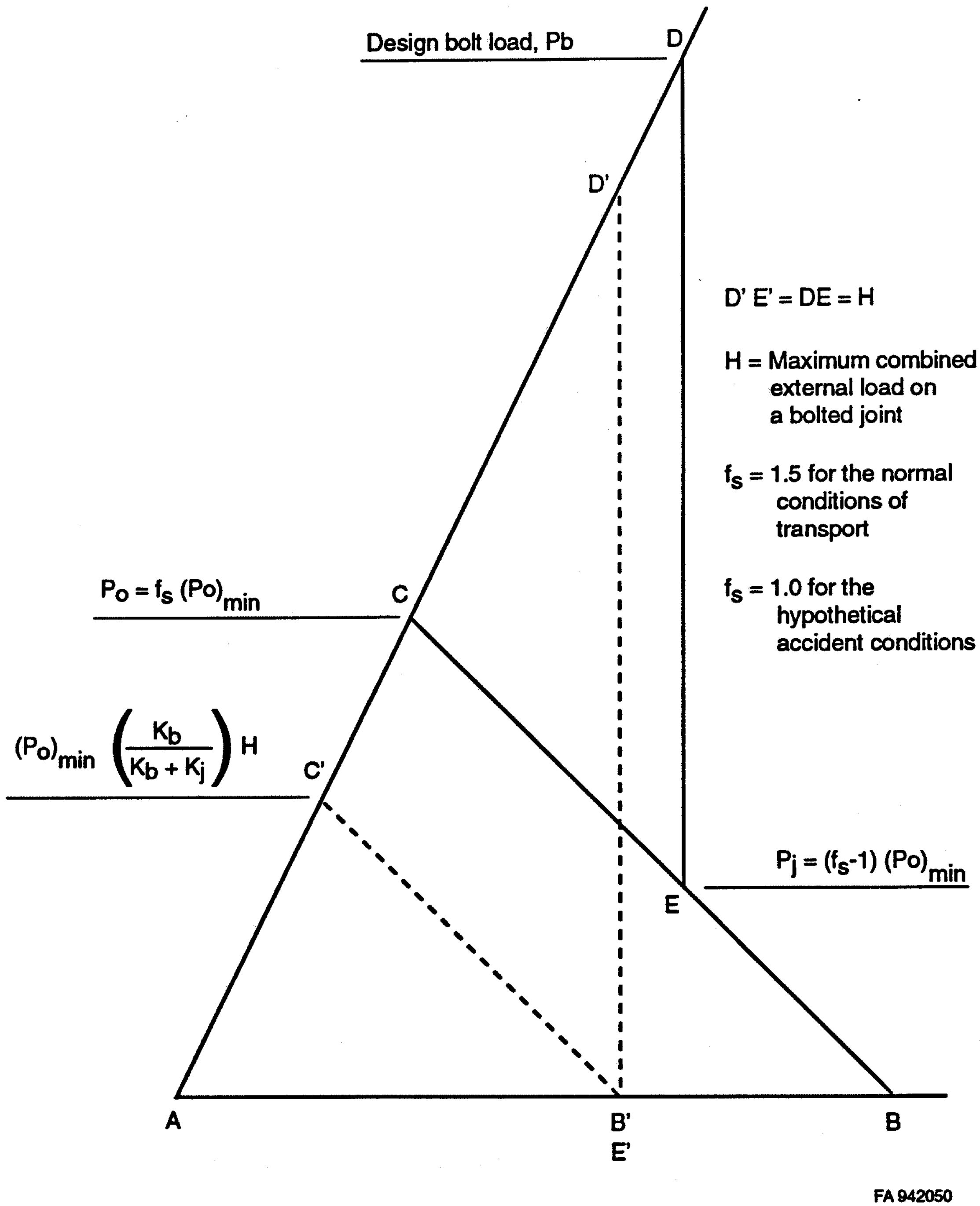

Fig. G16. Determination of preload.

For the containment vessel, a factor of safety (f.) can be applied to the minimum preload $(P_o)_{min}$ to yield the design preload (P_o).

$$P_o = f_s(P_0)_{min} = f_s.H[K_j/(k_j + K_b)]. \qquad \text{(G-24)}$$

In Equation (G-24), the maximum combined external load H on a bolt should include the impact load.

Table G2 shows P_o $(P_o)_{min}$, the design bolt load (P), and the residual contact force for a given maximum combined external load (H) and various safety factors (f_s) and k_b/K_j ratios. The f_s values of 1.5 and 1.0 are recommended for the Normal Conditions of Transport and the Hypothetical Accident Conditions, respectively. The applied preload should be the greater of the values calculated based on these two transport conditions. The use of the 1.0 factor of safety for the Hypothetical Accident Conditions is again justified because of the use of conservative criterion for preload determination.

ASME Code Section, III, Subsection NB-3232 provides allowable stress limits for Level A service conditions, which, in accordance with Regulatory Guide 7.6,[35] corresponds to the Normal Conditions of Transport for shipping casks. These stress limits can be adopted for use in the design of shipping cask closure bolts. The ASME Code does not have provisions for shear stress in bolts due to transverse loads like those resulting from an oblique impact of a shipping cask against an unyielding horizontal surface. A set of criteria for transverse shear stress is proposed.

The allowable average tensile stress, (F_t), in a bolt neglecting stress concentration is equal to two times the design stress intensity, (S_m), or

$$F_t = 2S_m. \qquad \text{(G25)}$$

Table G2. Design preload, design bolt load, and residual contact force between lid and cask wall

	K_b/K_j	Minimum preload $(P_o)_{min}$	Design preload (P_o)	Design bolt load (P)	Residual contact force
(f_s = 1.50)	.25	4H/5	6H/5	7H/5	2H/5
	.33	3H/4	9H/8	11H/8	3H/8
	.5	2H/3	H	4H/3	H/3
	1	H/2	3H/4	5H/4	H/4
	2	H/3	H/2	7H/6	H/6
(f_s = 1.34)	.25	4H/5	16H/15	19H/15	4H/15
	.33	3H/4	H	5H/4	H/4
	.5	2H/3	8H/9	11H/9	2H/9
	1	H/2	6H/9	7H/6	H/6
	2	H/3	4H/9	10H/9	H/9
(f_s = 1.00)	.25	4H/5	4H/5	H	0
	.33	3H/4	3H/4	H	0
	.5	2H/3	2H/3	H	0
	1	H/2	H/2	H	0
	2	H/3	H/3	H	0

H = maximum combined external bolt load including impact.

The stress is calculated based on the available bolt cross-sectional area, which is equal to either the nominal cross-sectional area of the bolt in the shank region or the stress area in the threaded region, whichever is smaller. The stress area of a bolt with standard threads is equal to the mean of pitch and root diameters,

$$\text{Stress area} = \pi/4\,(D - 0.9743/n)^2, \tag{G-26}$$

where D is the nominal diameter and n is the number of threads per inch of bolt length. The stress area of the bolt in the threaded region is equal to approximately 70 to 75 percent of the nominal cross-sectional area of the bolt. If the above formula for stress area is not used, it is sufficient to use the factor 0.7 on the nominal cross-sectional area.

The value of S_m can be calculated according to Subarticle 2120 of the ASME Code Section III. The S_m values for materials manufactured in accordance with the specifications of the ASME Code Section II can be found in Table I-1.3 of the Code. These S_m values are the smaller of one-third of the minimum specified yield strength at room temperature or one-third of the yield strength at the operating temperature. Thus, the yield strength of the bolting material corresponds to $3S_m$.

In shipping casks, there is usually no significant source of loading that would create bending stresses in the closure bolts. The residual torsion in bolts due to torquing in preload application is also usually small. However, in cases where significant bending and torsion exist, the stress limits included in this section should be used.

In cases where the bolts are under both tension and bending, the allowable tensile stress at the extreme fibre is limited to three times S_m, according to AS ME Code, Section III, Subsection NB-3232, or,

Allowable tensile stress at the extreme fibre = $3S_m$. (G27)

(bolts under both tension and bending)

Again, the stress is calculated based upon the available bolt cross-sectional area as described previously. One exception to Equation (G-27) is that, for high-strength alloy steel bolting, the limit is $2.7S_m$ instead of $3S_m$ if the higher of the two fatigue curves in the Code is used for fatigue evaluation in accordance with NB-3222.4(e).

For bolts having residual torsional stress due to the application of a preload using a torque wrench, the stress intensity, instead of the maximum stress, is limited to $3S_m$, or

Allowable stress intensity at the extreme fibre = $3S_m$, (G-28)

(bolts under tension, bending, and torsion)

When applying the above limits, the stresses in bolts can be calculated without considering the effects of stress concentration.

Under Level A Service Conditions (or Normal Conditions of Transport for shipping casks), the ASME Code does not have provisions for shear stress occurring as a result of a transverse load, such as the impact load in a shipping cask. It is recommended that the average shear stress across the available cross sectional area be limited to that allowed under Hypothetical Accident Conditions, but reduced by a factor of 1.5. That is the allowable average shear stress, (F_v), should be less than the smaller of $0.40S_y$ and $0.28S_u$,

Allowable average shear stress,

$F_v < 0.60S_y/1.5 = 0.40S_y$, and (G-29)

(available cross-sectional area)

$F_v < 0.42S_u/1.5 = 0.28S_u$.

S_y and S_u are the yield and tensile strengths, respectively, of the bolt material at operating temperature. The 1.5 factor is adopted from the relative margin of safety between some stress intensity limits of level A and level D service conditions of the ASME Code, Section IV (NB-3133 vs F-1331.5b, NF-3221.2 vs F-1332.1, and Section 1400a vs Section 1400c of Code Case N-284).

When both tension and transverse shear exist in a bolt, the following equation should be satisfied.

$$R_t^2 + R_v^2 < 1.0, \tag{G-30}$$

where:

$R_t = f_t/F_t$,

$R_v = f_v/F_v$,

f_t = computed tensile stress, and

f_v = computed average shear stress

Equation (G-30) follows also Appendix F of the ASME Code Section III. The allowable tensile and shear stresses, Ft and Fv, were calculated earlier respectively.

Appendix F of the ASME Code, Section III provides stress criteria for bolts under Level D Service Conditions, which, in accordance with Regulatory Guide 7.6,[35] corresponds to the Hypothetical Accident Conditions for shipping casks.

The average tensile stress, F_t, across the available cross-sectional area of a bolt is limited to the smaller of $0.70S_u$ and $1.0S_y$.

All load sources should be considered in calculating the bolt stresses including prying of the connected parts. However, in shipping casks, it is required that there be no leakage after impact.

This leakage requirement essentially eliminates plastic deformation in bolts. Therefore, the tensile stress at the extreme fiber of the bolts is limited to the yield strength of the material.

Under the ASME Code, the allowable extreme fiber tensile stress resulting from tension and bending, excluding stress concentration, is limited to S_u for high-strength bolts having a tensile strength of 100 kilo-pounds per square inch or greater at the operating temperature. The use of the S_u limit for high-strength bolts is not recommended because permanent plastic deformation is not allowed in closure bolt design. The stress limits are also applicable to the Hypothetical Accident Conditions if bending or torsional stresses are significant under Hypothetical Accident Conditions.

The allowable average shear stress (F_v) calculated based upon the available cross-sectional area, shall be the smaller of:

Allowable average shear stress, $F_v < 0.60S_y$, and (G-31)

(available cross sectional area) $F_v < 0.42S_u$

For bolts under the action of both tension and transverse shear.

$$R_t^2 + R_v^2 < 1.0. \qquad \text{(G-32)}$$

where:

$R_t = f_t/F_t$,

$R_v = f_v/F_v$,

f_t = computed tensile stress, and

f_v = computed shear stress.

Stresses caused by thermal expansion are considered as secondary under ASME Code because of their self-limiting nature. That is, local yielding and deformation is sufficient to provide relief for these stresses. The ASME Code stress limits involving secondary stresses are significantly higher than those for the load-controlled stresses. Appendix F of the Code does not place any limits on thermal expansion stress. It is clear that the stress limits proposed in the previous sections are too conservative if they are used for the accidental fire conditions under 10 CFR 71 requirements. The following allowable stresses are proposed for shipping cask design until better criteria becomes available.

Allowable tensile stress at the extreme fiber, or (G-33)

Allowable average tensile stress, $F_t = S_u$

and

Allowable average shear stress, $F_v = 0.60S_u$. (G-34)

For combined tension and shear, Equation (G-32) is still applicable. The allowables proposed above are conservative because the thermal load is self-limiting.

It is important to guard against possible thread stripping in bolt design. Thread stripping can occur during preload or during normal and accident conditions. Inadvertent overtightening of bolts during preload is dangerous because stripping of a fraction of the threads without breaking the bolt may go undetected and be the prelude to undesirable consequences. To avoid thread stripping, the engagement length (L_e) of the bolt should be calculated based on the concept that the shear strength of threads should be greater than the tensile strength of the bolt (T_t).

Shear strength of threads of the bolt, $T_{sb} = 0.5S_{tb}A_{sb}$. (G-35)

Shear strength of the internal threads, $T_{si} = 0.5S_{ti}A_{si}$ (G-36)

and

$T_{sb} > T_t$, (G-37)

$T_{si} > T_t$. (G-38)

where:

T_t = tensile strength of a closure bolt = $S_{tb}A_s$,

S_{tb} = tensile strength of bolt material,

S_{ti} = tensile strength of internal thread material,

A_s = tensile stress area of the bolt,

= $0.7854(D - 0.9743/n)^2$,

A_{si} = minimum thread shear area of internal threads

= $\pi n L_e D_{smin}[1/2n + 0.577(D_{smi} - E_{nmax})]$,

A_{sb} = minimum thread shear area of the bolt

= $\pi n L_e K_{nmax}[1/2n + 0.577(E_{smi} - K_{nmax})]$,

L_e = engagement length,

n = number of threads per inch,

D_{smin} = minimum of major diameter of bolt thread,

E_{smin} = minimum of pitch diameter of bolt thread,

K_{smax} = maximum minor diameter of internal thread, and

E_{smax} = maximum pitch diameter of internal thread.

If nuts are used, the above calculation method for engagement length can be used to establish the effective height of the nut, except that the shear strength of threads should be reduced by a factor of 0.75. This factor is used to cover the effects of nut dilation (an increase in nut diameter due to radial force on the nut as a result of wedging action of the threads under load). The minimum wall thickness or width across flats of the nut is also limited to 70 percent of the nominal diameter of the bolt.[36][37]

It is a good design practice to include recessed keys between the closure lid and the cask wall as shown in the typical closure geometries in Fig. G17. However, in calculating bolt stresses due to thermal expansion and impact conditions, justification is needed if credit is to be taken for the effects of the keys. Also, the gap between the lid and the cask wall should be considered in this situation.

Protective coating on the contact surfaces of a connection is usually applied to prevent corrosion, to improve slip resistance, and sometimes to prevent galling. However, under Regulatory Guide 1 .65[38] for reactor vessels, metal-plating (a protective coating) in the closure bolting is not allowed unless the plating will not degrade the quality of the material in any significant way or reduce the quality of results attainable by various required inspection procedures. The objection was based on the fact that plating can be more detrimental than helpful in terms of the fracture of the plating in the root of the threads, cracking and hydrogen embrittlement in the base material, and seizing of metallic coatings between the bolts and the nuts.

Lubrication is permissible in the cask closure bolts to prevent galling and to increase preload accuracy if a torque or turn method of preload control is used. However, lubricants should be stable at operating temperatures, and they should be compatible with the bolting and cask materials and the surrounding environment.[38]

Bolts should be tightened in a star or cross pattern for the preload and to prevent damage to gaskets during assembly. Also, tightening the bolts in several passes may be needed to increase accuracy in applying the preload.

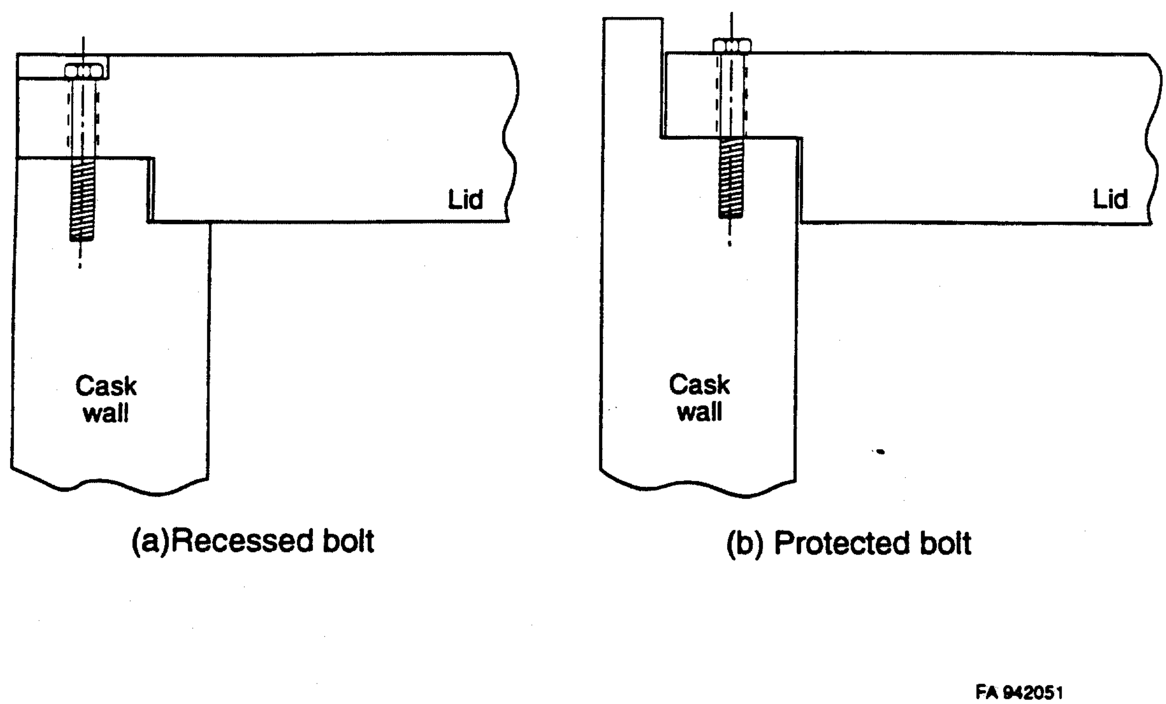

Fig. G17. Recessed and protected bolts.

It is a good design practice to provide recessed holes or protective rings on either the lid or the cask wall as those shown in Fig. G17 to prevent direct exposure of the bolt heads or bolt nuts to a lateral force from outside impact sources, such as a puncture pin.

It is also a good practice to design bolts with impact energy-absorbing capability. Figure G18 shows a bolt with reduced shank diameter, which is smaller than the minor diameter of its threads. For these kinds of bolts, the failure mode will not be overstressing at the thread root, but it will be overstressing at the shank where some impact energy can be absorbed. In Fig. G17, the shank is enlarged at its two ends to avoid a large gap between the bolt and the bolt hole, which may induce a large bending moment in the bolt due to relative lateral movement of the lid and cask wall. The number of bolts that should be used can be determined by stress limits. Trial and error, or iteration, on the number of bolts and the size of bolts may be required in design. If the stresses turn out to be very low, only a minimum number of bolts need to be provided. There is no rigid rule on the minimum number of bolts that should be used for a shipping cask. Several considerations are important in determining the minimum number of bolts to use in a shipping cask.

1. Thickness of lid. The thicker the lid the less likely that the lid will deform significantly under preload or internal pressure to cause uneven pressure on the gasket or the seal, which provide leak-tight containment for the radioactive material.

2. Radius of bolt circle. The spacing between bolts varies with the radius of the shipping cask. The larger the radius the larger the spacing and the more likely the lid will deform under preload and internal pressure. It is necessary that the spacing be small enough to maintain uniform pressure on the gasket between bolt holes in accordance with manufacturer’s specifications.

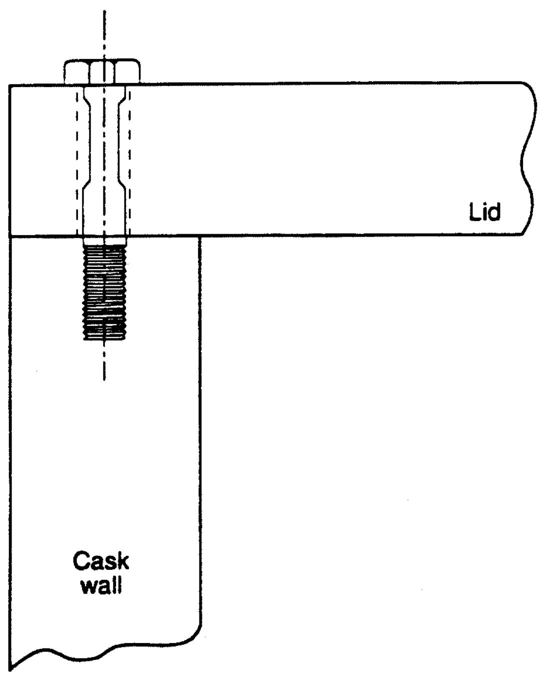

Fig. G18. Energy-absorbing bolt.

3. Common sense. It is clear that more than one closure bolt is required simply because one bolt does not provide a stable lid attachment to the cask body. It seems that a minimum of 8 bolts may be required for shipping casks. Also, it is more convenient if the number of bolts is equal to multiples of 4—the number of quadrants in a circle.

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