SAFETY ANALYSIS REPORT
FOR
CHEM-NUCLEAR SYSTEMS, INC.
MODEL CNS 3-55
TYPE B RADWASTE SHIPPING CASK

REVISION 0 - FEBRUARY 1994

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SAFETY ANALYSIS REPORT
FOR THE
CNS-3-55 SHIPPING CASK

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1.0 GENERAL INFORMATION

1.1 INTRODUCTION

This document presents a safeguard evaluation of the design and use of the Model No. CNS 3-55 Shipping Cask (also known as the Vandenburgh Cask), designed for motor transit

shipment of miscellaneous non-fissile exempt irradiated components packaged in secondary containers and by-product material in the form of solids and solidified waste contained within secondary containers.

The cask is designed to meet the Nuclear Regulatory Commission (NRC) requirements of 10 CFR 71[1] and the Department of Transportation (DOT) requirements of 49 CFR 171-179[2].

All shipments will be made according to NRC and DOT regulations for transporting large quantities of non-fissile radioactive materials by motor vehicle assigned for sole use.

Summary

A safety evaluation has been made of the CNSI 3-55 shipping cask in accordance with NRC and DOT regulations. The results of the evaluation indicate that:

(a) The shipping cask satisfies the standards specified in Subpart C of 10 CFR 71.
(b) Radiation dose rates are within the limits specified in 49 CFR 173.393 for transport in a motor vehicle assigned for sole use.

1.2 PACKAGE DESCRIPTION

1.2.1 Packaging

The shipping cask arrangement is shown in Drawing MOD-100, Revision 11 in Section 10.0 and sketch 1.2.1 below.

FIGURE WITHHELD UNDER 10 CFR 2.390

CNS 3-55 CASK

5KETCH NO. 1.2.1

A waste container or basket is positioned on the cask base closure assembly during loading. The cask upper section, shaped like a bell, is lowered down over the container or basket and bolted to the closure assembly using a socket wrench on a long extension. The cask is then lifted out of the pool, prepared for shipment and then positioned horizontally on the cradle skid mounted on a flat bed trailer. The shipment is made with the cask in the horizontal position.

Cask lifting yokes and slings are provided for loading and unloading.

The package is a steel-encased, lead-shielding cask with crushable impact limiters. The basic cask body is a steel cylinder 133-3/4 inches long by 50-1/2 inches in diameter outside dimensions. The maximum cavity dimensions are 111.125 inches long by 36 inches in diameter.

The outside steel encasement consists of two 1/2-inch thick plates on the sides and three plates totaling 2-5/8 inches thick on the closed end. The outside plates are stainless steel. The outer surface of the cask may be optionally coated with a nuclear grade epoxy paint.

The containment vessel (cavity liner) consists of a 1/4-inch thick cylinder plate clad with stainless steel on the cavity side and welded to a 1/2-inch thick stainless plate at the closed end of the cavity.

The cavity liner and the outer shells, at the open end (closure end) of the cask body, are welded to a 1-1/4-inch thick stainless steel ring plate.

Cask body shielding is provided by 6-inch thick lead on the sides and 5-1/4-inch thick lead at the closed end, poured to fill the annular and end spaces.

The removable, flanged and recessed base plate closure assembly consists of an outer surface 3/8 inch thick stainless steel plate covering an inner plate 1-1/4 inch thick. The inside cavity plate is 5/8 inch thick. The space between the plates is lead filled to a maximum of 6 inches thick.

A lead filled ring is provided at the bottom of the cask to provide shielding where lead slumping could occur in the cask body. This ring is attached to the base plate.

The base plate closure assembly is secured to the cask body by twelve 1-1/2 inch diameter high strength bolts and nuts and sealed with two silicone O-ring seals.

The cask cavity containment is penetrated by a vent port at the closed end and a drain line through the base plate closure assembly.

Two 8 inch diameter trunnions are provided for cask handling. Two 4 inch diameter trunnions are designed to be bolted to the lower portion of the cask for tilting the cask. The two 4 inch trunnions are removed during shipping and bolted to the cradle assembly.

The cask will have a shock absorber structure attached to both ends to cover the corners. Both are similar in design and are built to encircle and protect the cask during normal and accident conditions. The shock absorbers are constructed of 1/2 inch thick mild steel plates. Lengths of 6 inch Schedule 80 pipe are on the shock absorbers on each end of the cask to cushion impact in normal or accident conditions.

These pipe sections are closed to add strength. They vary somewhat in length to provide step-wise contact as deformation occurs with impact. Projecting radially from the side of the shock absorbers are lengths of 6-inch Schedule 80 pipe. Each shock absorber has a total of six 6-1/8-inch long pipes to protect the trunnions in a side drop. The shock absorber for the closure end also has six 2-inch long pipes to protect the cask closure bolts in a side drop. In additional to the shock absorbers on the ends of the cask are two trunnion shock absorbers which are gusseted structures of 1-inch thick mild steel plates.

As an option, a sunshade may cover the cask during transport. The shade is shown in Drawing C-110-D-5001, Revision 1. The structure is rigid and designed to withstand all reasonable conditions in transport.

Cask Transport Skid

The cask is mounted horizontally and handled during transport in a structural steel skid assembly that his designed to spread the load on the flatbed transport trailer. (Reference Drawing Number MOD-124, Rev. D.) The cask body rests in two steel beam cradles. Two 1.0-inch thick steel bands strap the cask down at the cradle locations, and are bolted to the structural frame by four 1.25-inch diameter bolts for each end of each strap. The skid assembly has kick (stop) plates at each end of its frame to restrain the cask from forward or aft movement. The skid assembly is then bolted to the transport trailer by twenty-eight 1.25-inch diameter and four 1.50-inch diameter bolts.

1.2.2 Operational Features

All items in this section are covered in other sections. There are no complex operating features related to this package.

1.2.3 Contents of Packaging

Type, form, and maximum quantity of material per package:

(a) Greater than Type A quantities of radioactive material as neutron activated metal or metal oxide in solid form.

(b) Greater than Type A quantities of by-product material in a solid or solidified form.

(c) Both (a) and (b) above will be contained in secondary containers.

(d) That portion of that cask contents that, due to chemical or physical form, can be termed as available for leakage shall be limited to the specific quantities given in Section 4 of this SAR and will comply with NRC Regulatory Guide 7.4.

(e) That quantity of radioactive material that does not generate spontaneously more than 250 thermal watts of decay heat.

(f) The maximum weight of the contents in the cask cavity, including any secondary container(s) and shoring, is 9434 pounds.

(g) Depleted neutron startup sources.

1.3 Appendix

1.3.1 References

(1) "Packaging of Radioactive Material for Transport", Code of Federal Regulations, Title 10, Part 71 (December 31, 1968).

(2) "Radioactive Materials and Other Miscellaneous Amendments", Code of Federal Regulations, Title 49, Parts 171-179 (October 4, 1968).

1.3.2 Detailed Package Description page 1-8

1.3.2 Detailed Package Description

The cylindrical cask body is constructed of a 0.25 inch, A-285, Grade C steel cavity liner clad with 20% T-304 stainless steel on the cavity side, a maximum of 6 inches of lead; and a 1.0-inch outer shell fabricated from 0.50-inch, A-285, Grade C steel (inner) and 0.50-inch, 304 stainless steel (outer).

The cask outer surface may be optionally painted with a nuclear grade epoxy coating. This coating provides excellent wear resistance, and provides a more easily decontaminated surface for handling operations.

The cylindrical cavity liner, at the closed end of the cask, is welded to a 0.50-inch thick, A-285, Grade C circular steel plate clad with 20% T-304 stainless steel on the cavity side.

The cylindrical outer shell, at the closed end of the cask, is welded to a 0.50-inch rind plate, 50.50 inches O.D. by 36.25 inches I.D., A-285, Grade C steel. This ring plate I.D. is then closed by a 0.50-inch circular plate, 36.25 inches O.D., A-285, Grade C steel, clad with 20% T-304 stainless steel on the outer surface. A second layer of steel, on the closed end of the cask, is made up of a 0.625-inch circular plate, T-304 stainless steel, 22.00 inches O.D.; a 0.625-inch plate ring, T-304 stainless steel, 33.00 inches O.D.; and a 0.625-inch plate ring, T-304 stainless steel, 50.50 inches O.D. by 34.00 inches I.D. These three plates are welded, full penetration, to each other as well as to the first steel plate layer.

A third layer of steel, on the closed end of the cask, is made up of a 1.50 inch plate ring, T-304 stainless steel, 50.50 inches O.D. by 7.62 inches I.D., welded to the second layer at the O.D. and I.D.

A vent hole made up of a 1.25 inch, Schedule 40, stainless steel pipe, penetrates the center of the closed end of the cask. The pipe inner end is welded to the 0.50-inch cavity liner and a 0.75-inch plate back-up ring. The pipe outer end is welded to the two inner layers of 0.50-inch and 0.625-inch steel.

The vent hole in the closed end of the cask is plugged by a 6.00 inch long, lead-filled 1.00 inch Schedule 40 stainless steel pipe, welded to a circular cover cap plate, 1.50 inch thick by 7.375 inches O.D., sealed with a 1/8 inch thick gasket of sheet packing and a 3/8" silicone O-ring. The assembly is fastened with four 0.50 inch by 2.00 inch bolts. This vent plug assembly also contains a recessed plug valve stem sealed with a 1/16 inch by 3/16 inch silicone O-ring. The stem is threaded 7/16-14 NC and can be opened to a diagonal 1/4 -18 NPT threaded sampling port for test pressure gauge or vent sampling. The sample port is plugged with a 1/4 inch NPT male pipe plug when not in use. The valve stem is closed by turning in to full stop. The radial O-ring on the end of the stem in conjunction with the other seals and gaskets shown on Drawing No. 0999-C-08. seal the containment. A 1-1/2 inch by 3 inch by 1/8 inch stainless cover plate with a 1/16 inch neoprene gasket retained by (6) 8-32 by 1/4 inch stainless steel bolts covers the recessed valve stem hole and the test port hole when not in use or in transport.

Total material shield thickness of closed end of the cask is 3.125 inches of steel and 5.25 inches of lead.

The cylindrical cavity liner and the cylindrical outer shell, at the open end of the cask, are welded to a 1.25 inch ring plate, 50.50 inches O.D. by 36.625 inches I.D., T-304 stainless steel.

The base plate closure retaining flange, at the open end of the cask, is made up of two plate rings of 0.50 inch, A-285, Grade C steel. The upper ring is 58 inches O.D. by 49.625 inches I.D., and the lower ring is 57 inches O.D. by 49.625 inches I.D. The lower ring is located 3 inches from the cask open end closure ring plate, and the upper ring is located 6 inches from the cask open end closure ring plate. The two flange rings are welded to the cask outer shell, and are

-Continued on Following Page-

joined and reinforced by spaced gusset plates as well as twelve equally spaced pipes providing the bolt holes for the base plate closure bolts.

The cask base plate closure (lid) assembly is made in a stepped configuration so that the closure penetrates 2.625 inches into the cavity of the cask open end. The cavity side of the closure is a 0.625 inch, T-304 stainless steel circular plate 34.75 inches O.D. Its circumference is welded to a 0.50 inch, T-304 stainless steel ring 35.5 inches O.D. by 3.0 inches high. A 1.0 inch thick, T-304 stainless steel flange plate ring, 58.0 inches O.D. by 35.5 inches I.D. is welded to the bottom circumference of the ring. A second 0.50 inch ring of A-285, Grade C steel, 49.5 inches O.D. by 3.0 inches high is welded to the underside of the 1.0 inch flange plate ring. Twenty 0.50 inch by 3.0 inch by 4.25 inch gussets of A-285, Grade C steel are spaced around the circumference of the 0.50 inch ring, reinforcing the outer edge of the 1.0 inch flange plate ring. The bottom of the cask base plate closure assembly is made up of a 1.25 inch A-285, Grade C steel circular plate, 50.5 inches O.D. welded to the 0.50 inch ring. The 1.25 inch bottom plate is then covered with a 0.375 inch plate ring 50.0 inches O.D. by 34.0 inches I.D. and a 0.375 inch circular plate, 33.5 inches O.D., both of T-304 stainless steel, welded, full penetration, to each other, as well as to the 1.25 inch bottom plate.

The 1.0 inch flange plate of the base plate closure assembly has two O-ring grooves at 43.0 inches O.D. and 39.0 inches I.D. to contain two 0.375 inch diameter silicone O-rings. The O-rings seat and seal on the flat surface of the 1.25 inch ring plate at the open end of the cask.

The cask base plate closure assembly has a 6 inch diameter drain sump, 1.625 inches deep, lined with 0.25 inch, T-304 stainless steel. The bottom of the drain sump is connected to a 0.75 inch, Schedule 40, T-304 stainless steel pipe leading out through the side of the lower outside ring of the assembly, below the closure flange. This drain pipe is closed by a 0.75 inch threaded plug relief valve. Total material shield thickness of the base plate closure assembly is 2.25 inches of steel and 6 inches of lead.

An internal lead-filled shield ring assembly (Reference Drawing No. C-111-D-0001) is mounted on and bolted to the 0.625 inch plate that forms the cavity side of the closure assembly. The base of the ring is a ring plate 26 inches I.D. by 33 inches O.D. by 0.75 inch thick 304 stainless steel. The inside of the ring assembly is a rolled ring of 0.75 inch 304 stainless steel, 5.25 inches high. The outside of the ring assembly is a rolled ring of 0.75 inch 304 stainless steel, 5.25 inches high. The ring assembly is closed by a top ring plate 24.50 inches I.D. by 34.50 inches O.D. by 0.75 inch thick 304 stainless steel. The inside of the ring assembly is filled with lead 3.50 inches thick. All plate welds are full penetration fillet welds. The inside of the ring assembly has eight (8) bolt lug flanges 2.50 inch by 2.50 inch by 0.750 inch thick 304 stainless steel with 0.875 inch diameter bolt holes, each welded to the inside surface of the ring with 3/8 inch fillet welds. Each bolting flange bolts to a 2.50 inch by 2.50 inch by 0.750 inch thick 304 stainless steel bolting boss block that is welded to the closure assembly top cavity-side plate with 3/8 inch fillet welds. Each boss block is drilled and tapped 3/4-10 UNC-2B. The shield ring is secured to the closure assembly by eight (8) 3/4-10 UNC by 1-1/4 A-193, 304 stainless steel bolts.

The cask base plate closure assembly with shield ring is attached to the cask during shipment by twelve 1.50 inch diameter bolts, equally spaced, between the base plate closure assembly flange and the cask retaining flange. There are three guide pins on the flange plate face of the closure assembly that penetrate into alignment holes in the cask open end face plate.

The 8.0 inch diameter by 5.0 inches long lifting trunnions are welded to the outer shell of the cask, 42-5/8 in. from the closed end of the cask. Two flanged 4.0 inch diameter turning trunnions are bolted to plate trunnion bosses 18.75 inches from the open end of the cask. These turning trunnions are removed during cask shipment.

An impact limiter is attached to each end of the cask prior to lifting the cask into the cask transport skid assembly mounted on the flat-bed trailer (Reference Drawing No. 0999-D-07, Rev. G).

The limiter on the base plate closure end of the cask closely fits the entire closure flange and is made up of a 0.50 inch, A-36 steel plate ring 59.56 inches O.D. by 34.94 inches I.D. with a 0.50 inch, A-36 steel ring skirt 59.75 inches O.D. by 13.0 inches wide welded to its circumference. The plate ring has sixteen 6 inch diameter, Schedule 80 pipes, 6 inches long, end-welded to it, equally spaced, on a 43.0 inch diameter centerline circle. The ring skirt has twelve 6 inch diameter, Schedule 80 pipes, 6 inches long, end-welded to it, 9.37 inches from the plate ring joint.

The pipes are arranged so that a 6 inch long pipe on each side is on the trunnion centerline with an additional 6 inch long pipe at 20 degrees on each side of it. Six pipes 2 inches long are arranged similarly at 90 degrees from the trunnion centerline on each side of the limiter skirt, with a 22.5 degree spacing. This limiter assembly is fastened to the cask by six 1.0 inch diameter by 16.25 inches long bolts from the plate ring through the closure assembly flange and then through the two cask flange rings, with a reinforcing spacer block between the two flange rings at each bolt hole.

The limiter on the closed end of the cask is constructed similarly with a 51.5 inches O.D. by 35.5 inches I.D. plate ring and a 51.5 inches O.D. by 10.0 inches wide ring skirt, all of A-36 steel. The plate ring has sixteen 6.0 inch diameter, Schedule 80 pipes, 6 inches long, equally spaced, end-welded to it on a 43.0 inch diameter centerline 3.0 inches from the plate ring and skirt joint. These are arranged so that one pipe on each side is on the trunnion centerline with an additional pipe at 22.5 degrees space on each side of it.

This limiter closely fits the O.D. of the cask and is fastened to the cask by six equally spaced 0.75 inch by 1.25 inch bolts through 0.50 inch by 5.0 inch by 2.5 inch attachment bar plates welded to the skirt and into six 0.50 inch by 2.0 inch by 2.0 inch threaded boss plates all of T-304 stainless steel. The boss plates are welded to the outer shell of the cask.

All fifty 6.0 inch pipes on both limiter assemblies are closed by a 0.50 inch thick, 6.0 inch diameter pipe cap plate welded to the pipe end.

A trunnion impact limiter is provided for each lifting trunnion. They consist of a 10.0 inch diameter, Schedule 160 pipe, 6.0 inches long, welded to a 1.0 inch by 24.0 inch by 24.0 inch, A-36 steel plate with an 8.5 inch diameter hole in its center. The plate is rolled to a 25.25 inch radius to closely fit the O.D. of the cask outer shell. The sides of the pipe are gusseted to the plate with four 1.0 inch by 6.0 inch by 6.125 inch, A-36 steel plates. Each trunnion impact limiter assembly is fastened to the cask with four 0.75 inch by 1.875 inch long bolts through four, corner mounted, 1.0 inch by 2.0 inch by 5.0 inch attachment bar plates welded to the limiter base plate, and into four 1.0 inch by 2.0 inch by 2.0 inch threaded T-304 stainless steel boss plates welded to the cask outer shell. These trunnion impact limiters provide 2.0 inches clearance from the end of the trunnion to the end of the 10.0-inch diameter pipe.

Cask Transport Skid

The cask is mounted horizontally and handled during transport in a structural steel skid assembly that is designed to spread the load on the flatbed transport trailer. (Reference Drawing Number MOD-124, Rev. D.) The cask body rests in two steel beam cradles. Two 1.0-inch thick steel bands strap the cask down at the cradle locations, and are bolted to the structural frame by four 1.25-inch diameter bolts for each end of the strap. The skid assembly has kick (stop) plates at each end of its frame to restrain the cask from forward or aft movement. The skid assembly is then bolted to the transport trailer by twenty-eight 1.25-inch diameter and four 1.50-inch diameter bolts.

Cask Sunshade Assembly

As an option, the entire cask and skid assembly may then be covered during transport by a sunshade (Reference Drawing Number C-110-D-5001, Rev. 1) bolted to the trailer bed.

STRUCTURAL EVALUATION

August 11, 1975

Rev. 1 - September 29, 1975

Performed by: Tri Nuclear Corporation
P. O. Box 178
Ballston Lake, New York 12019

Prepared by: James Warden

2.0 STRUCTURAL EVALUATION

2.1 Structural Design

2.1.1 Discussion

As shown on the drawings, the cask assembly consists of the main body and the bottom cover closure assembly with shield ring. The containment vessel is considered to be the cask inner liner.

The cask closures include the main bottom cover, which is secured with 12 1-1/2 inch bolts and sealed with two concentric silicone O-rings, the drain line which is fitted during shipment with a vent relief valve and a vent plug assembly in the closed end of the cask.

There are several impact limiters, which are secured to the cask only during shipment. Each end of the cask is fitted with an end collar, having (16) 6 inch short pipe stubs welded in a circle axially with respect to the cask. There also are (3) 6 inch short pipe stubs extending radially from each side of the two end impact limiters in line with the trunnions. The two trunnions are each fitted with an impact limiter to protect them from a side drop accident. Sketches showing these impact limiters are included in the appendix to this section. The impact limiter on the closure end of the cask also has three (3) 2 inch pipe stubs on each side between the 6 inch pipe studs.

2.1.2 Design Criteria

The design loads considered for the analyses include all of those specified in 10CFR71, including the Standards for Hypothetical Accident Conditions of Transport.

The general standards for all packaging and the standards for normal conditions of transport are met by limiting stresses on any part of the cask due to their loads to less than the yield strength of the material, except for the following condition. In the case of the normal condition free fall one foot drop tests, the impact limiters will be subject to local deformation beyond the ultimate strength of the material at the impact areas. The cask itself, however, will not be stressed above the yield strength of the material at any area which is vital to the safe operation of the cask.

For the hypothetical accident conditions, the stresses can exceed the yield strength and result in localized material deformation. Leak tightness is not required under these tests, since the potential radioactive material available for leakage is below the amount which could be tolerated by the standards for the hypothetical accident condition. Gross leakage or rupture of the cask, which could result in material release, is not acceptable, nor is tearing of the outer shell which could result in subsequent melting of lead in the fire test and excessive loss of shielding material.

For impact loading, both the "g" force and the total available energy (in. lbs.) are considered in the analysis. For the special case of impacting on a softer surface than an "unyielding" surface (cookie cutter problem), a special factor of force times available energy (pounds/square inch x inch pounds) = (pounds squared/inch) was used to determine the conditions for the maximum potential damage to the cask.

Standard "strength of material" formulas are used in the analyses, plus special formulas from R.J. Roark's 4th Edition, "Formulas for Stress and Strain".

2.2 Weights and Centers of Gravity

2.2.2 Weight Analysis

o Vandenburgh Cask

		Pounds
1. Shell	51,947	
2. Base Plate (Cover)	4,965	
		56,912

o Internals

1. Lead Shielding Ring (overcome slump)	1,307	
2. Cask Contents, including container and shoring	9,434	
		10,741

o Impact Limiters (on cask during shipment)

1. For cover end of cask	1,100	
2. For solid end of cask	700	
3. For trunnions (2) x (250)	500	
		2,300
TOTAL WEIGHT OF CASK AND CONTENTS		69,953 lbs.

2.2.2 Center of Gravity

Corner drop such that the cask center of gravity is directly over the point of impact will give an angle to the vertical of 21-1/2^{o}F.

2.3 Mechanical Properties of Materials

The following is a list of pertinent material data for the materials used in the construction of the cask. Properties at the various significant temperatures are given. The reference for the lead data is the National Lead Company and MIL-HDBK No. 5 is used for the other materials.

See Appendix 2.10.1 and 2.10.2 to this section for additional test data results applied to the cask analyses.

2.3 Mechanical Properties of Materials (continued)

2.3.1 Lead (chemical)

- (γ) density - 0.41 lbs/in^3
- (F_{tu}) ultimate tensile strength - 2600 psi
- (F_{ty}) yield strength - 860 psi
- (E) Young's Modulus - 2.5 x 10^6 psi
- % of Elongation - 53
- (F_{cu}) compressive strength - 2900 psi
- (α) coefficient of linear thermal expansion
  50 F to 212F 16.40×10^{-6} in/in/^oF
  solidification shrinkage - 3.85%

2.3.2 Stainless Steel

AISI 301, 302, 303, 304, 321, and 347 (annealed)

- (γ) density - 0.286 lbs/in^3

% elongation	thickness
40	≤ 0.015
45	0.016 to 0.030
50	≥ 0.031

- (α) coefficient of linear thermal expansion to
  620 F 9.70×10^{-6} in/in/^oF

			Room Temperature (ksi)	130 F (ksi)
o	(F_{tu})	ultimate tensile strength	75	71.2
o	(F_{ty})	tensile yield strength	30	28.8
o	(F_{cy})	compressive yield strength	35	34.3
o	(F_{su})	ultimate shear strength	40	37.6
o	(F_{bru})	ultimate bearing strength (e/D = 2.0)	150	141
o	(F_{bry})	bearing yield strength (e/D = 2.0)	50	49.5
o	(E)	tensile modulus of elasticity	29,000	28,700
o	(E_c)	compressive modulus of elasticity	28,000	27,700
o	(G)	shear modulus	12,500	

2.3 Mechanical Properties of Materials (continued)

2.3.3 Carbon Steel

AISI 1025 (Room Temperature Properties)

- o (γ) density - 0.284 lbs/in^3
- o (α) coefficient of linear thermal expansion
  - 68 F to 212 F 6.78×10^{-6} in/in/^oF
  - 68 F to 392 F 7.00×10^{-6} in/in/^oF
- o (F_{tu}) ultimate tensile strength - 55,000 psi
- o (F_{ty}) tensile yield strength - 36,000 psi
- o (F_{cy}) compressive yield strength - 36,000 psi
- o (F_{su}) ultimate shear strength - 35,000 psi
- o (F_{bru}) ultimate bearing strength - 90,000 psi
  - (e/D = 2.0) % elongation - 10 to 22 minimum
  - (tubing) - 8 to 13
- o (E) modulus of elasticity - 29×10^6 psi
- o (G) shear modulus - 11×10^6 psi

2.3.4 Welds

carbon and alloy steels

- o (F_{tu}) ultimate tensile strength - 51,000 psi
- o (F_{su}) ultimate shear strength - 32,000 psi

2.4 General Standards for All Packages

2.4.1 Chemical and Galvanic Reactions

No chemical or galvanic reaction is anticipated for the intended service. Materials within the cask consist of carbon steel (painted), stainless steel, aluminum, silicone rubber O-ring seal and residual water vapor of neutral pH.

2.4.2 Positive Closure

The open-end closure consists of two concentric silicone rubber O-rings, as shown on the drawing, and a protected vent valve spring loaded to relieve excessive over-pressure within the cask, if necessary, during accident conditions. Pressure relief is not expected, even with hypothetical accident

2.4.2 Positive Closure (continued)

conditions since only residual water vapor will remain within the cask during shipment.

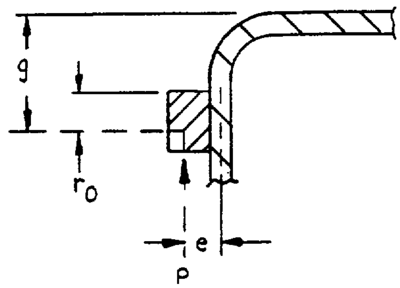
Positive closure of the cask is accomplished, during normal shipping conditions, by 12 1-1/2 inch diameter bolts having 160,000 pounds ultimate tensile strength. The closure with respect to accident conditions is analyzed in a subsequent section.

The closure is inaccessible during transportation, since the vent relief valve and the cask closure bolts are covered by the pipe buffer shock absorber assembly bolted and locked into place by the cask skid cradle during transportation. The cask must be removed from its skid cradle before the shock absorber assembly can be removed from the cask, thereby making the closure devices accessible.

2.4.3 Lifting Devices

The only lifting devices on the cask are the two main trunnions located near the solid end of the cask. The trunnions are designed to support three times the weight of the loaded cask without generating stress in any material of the cask in excess of its yield strength.

A conservative assumption for the maximum stress in the outer shell due to the trunnion is to assume the outer shell near the trunnion is equivalent to a simple supported circular plate with the following dimensions.



r_o = 4 in.
g = 41.75
e = 3.25

The maximum stress in the shell is:
(Reference Roark - "Formulas for Stress and Strain" - page)

$$\sigma_{max} = \frac{3\,Pe}{4\pi t_{o.s.}{}^2 r_o}\left\{1 + 1.3\ \ln\ \frac{2(g - r_o)(r_o + .7g)^2}{.49g^3}\right\}$$

$$= \frac{3(3.25)(99{,}969)}{4\pi(1.5)^2(4.0)}\left\{1 + 1.3\ \ln\ \frac{2(37.75)(33.2)^2}{.49(41.75)^3}\right\}$$

$$= 18{,}100 \text{ psi}$$

$$\text{M.S.} = \frac{28{,}800}{18{,}100} - 1 = \underline{.59}$$

The maximum bending stress on the trunnion is:

$$\sigma_{max} = \frac{4\,Pe}{\pi r_o^3} = \frac{4(99{,}969)(3.25)}{\pi(4.0)^3} = 6{,}464 \text{ psi}$$

$$\text{M.S.} = \frac{28{,}800}{6{,}464} - 1 = \underline{\text{large}}$$

The maximum shear stress on the trunnion is:

$$\sigma_{s\ max} = \frac{P}{\pi r_o^2}$$

$$= \frac{(99{,}969)}{\pi(4.0)^2}$$

$$= \underline{1{,}988 \text{ psi}}$$

$$\text{M.S.} = \frac{37{,}000}{1.5(1988)} - 1 = \underline{\text{large}}$$

2.4.3 Lifting Devices (continued)

The maximum bending stress on the trunnion weld is:

$$\sigma_{max} = \frac{P_e}{\pi r_o^2 \; t_{tw}}$$

$$= \frac{99{,}969\ (3.25)}{\pi(4.0)^2(1.0)}$$

$$= 6{,}464 \text{ psi}$$

$$M.S. = \frac{51{,}000}{1.5(6464)} - 1 = \text{large}$$

The maximum shear stress on the trunnion weld is:

$$\sigma_s \max = \frac{P}{2\pi r_o \; t_{tw}}$$

$$= \frac{99{,}969}{2\pi(4.0)(1.0)}$$

$$= 3{,}978$$

$$M.S. = \frac{32{,}000}{1.5(3978)} - 1 = \underline{\text{large}}$$

2.4.4 <u>Tie-down Devices</u>

As shown on the drawing, (Reference Drawing MOD-124), the cask is transported in a skid cradle bolted to the truck trailer frame. The vertical and horizontal transverse restraint is through the cradle and the two heavy hold down straps; the horizontal axial restraint (direction in which vehicle travels) is through the kick plates in the skid cradle through the cask closure end pipe buffer shock absorber assembly to the cask itself. There is an upending moment on the hold-down straps in the axial direction to resist the 10g axial load (direction in which vehicle travels).

2.4.4 Tie-down Devices (continued)

The maximum load on a strap with the three combined loads is:

$$P_{max} = \frac{10W(\frac{D}{2})}{2L} + \frac{2W}{4} + \frac{5(\frac{D}{2})W}{2D}$$

where: D = outside dia. of cask = 50.5 inches
L = distance between hold-down straps = 79 inches
W = 70,000 pounds

$$P_{max} = W\left(\frac{10x\ \frac{50.5}{2}}{2\ x\ 79} + \frac{2}{4} + \frac{5x\ \frac{50.5}{2}}{2\ x\ 50.5}\right)$$

$$P_{max} = W\ (1.6 + .5 + 1.25)$$

$$P_{max} = 70,000\ x\ 3.35 = 234,500 \text{ pounds}$$

Tensile stress in the 8 in. x 1 in. strap is:

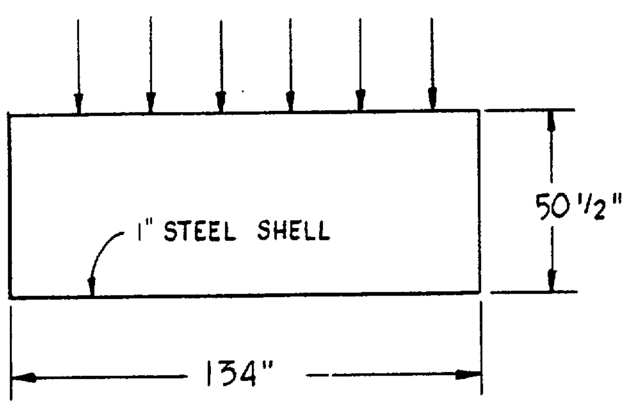
$$\sigma = \frac{P}{A} = \frac{234,500}{8\ x\ 1} = 29,312 \text{ psi}$$

$$M.S. = \frac{36,000}{29,312} - 1 = \underline{0.23\ M.S.}$$

2.5 Standards for Type B - Large Quantity Packaging

2.5.1 Load Resistance

W = 70,000 pounds
5W = 350,000 pounds



2.5.1 Load Resistance (continued)

MC

whre I M ~ x0O~ 3

5 x 70,000 x .134 5.86x106 in.-lbs.

C - 50.5 / 2 = 25.25

1 = ff25.254 " 24.254 = 47,100

$\frac{5.86\times10^{6} \times 25.25}{47,100}$ = 3142 psi

M.S. = $\frac{36,000}{3,142}$ _ I = 10.5 M.S.

2.5.2 External Pressure

The requirement for external pressure is that the cask must be able to withstand an external pressure of 25 psig without loss of contents (Ref. 10CFR71.32(b). Assume the outer shell is supported by the lead in resisting the external pressure.

FIGURE WITHHELD UNDER 10 CFR 2.390

Closed end:

Steel plates: t = 2.625 in,

Lead: t = 5.250 in.

V = steel .30

lead .45

D = 49.5 in. mean diameter

P = 25 psig

at 32 (D/t)^2 P (3 + V)

(From Roark, R.J. Formulas for Stress and Strain, 5th Edition, Case lOa, Page 363.)

steel: S3f 3 (49.5 / 2.625)^2 (25)(3 + .30)

= 2750 psi

lead: f32 3 (49.5 / 5.25) (25)(3 +.45)

= 718 psi (ayield = 1180 psi for lead)

2.5.2 External Pressure (continued)

Lid Closure End:

Steel plates: t = 1.625 in.

Lead: t = 6.0 in.

$$\text{steel: } \sigma_f = \frac{3}{32}\left(\frac{49.5}{1.625}\right)^2 (25)(3 + .30)$$

$$= 7176 \text{ psi}$$

$$\text{lead: } \sigma_f = \frac{3}{32}\left(\frac{49.5}{6.0}\right)^2 (25)(3 + .30)$$

$$= 550 \text{ psi}$$

For the closed end and the closure end of the cask, the entire pressure can be taken by the steel outer plates and the lead backing the plates with a large safety margin.

Cylindrical Cask Shell:

D = 49.5 in.

t = 1.0 in.

E = Modulus of Elasticity = 28×10^{-6} psi

σ_y = Yield Strength = 30,000

$$P_c = \frac{2t}{D} \frac{\sigma_y}{1 + \frac{\sigma_y}{E} \frac{D^2}{t}}$$

(Roark, R.J., Formulas for Stress and Strain, 4th Edition, Chapter 12, page 298, Case 1)

$$P_c = \frac{2(1.0)}{49.5} \frac{30,000}{1 + \frac{30,000}{28 \times 10^6}\left(\frac{49.5}{1}\right)^2}$$

$$P_c = 338 \text{ psi}$$

$$\text{M.S.} = \frac{338}{25} - 1 = \underline{12.5 \text{ M.S.}}$$

2.6 Normal Conditions of Transport

2.6.1 Heat

The thermal evaluation for the heat test is reported in Section 3.4.

2.6.1.1 Summary of Pressures and Temperatures (Normal Conditions)

Maximum pressure - 3 psig (See 3.4.4)

Maximum temperatures -

Ambient air	- 130°F
Cask outside surface	- 160°F
Cask inside surface	- 160°F

2.6.1.2 Differential Thermal Expansion (Normal Conditions)

α= Coefficient of linear expansion 9.7×10^{-6} in/in/°F

D = outside diameter of cask shell = 50.5 in.

L = length of cask shell = 133 in.

Radial thermal expansion of outer shell relative to inner shell:

$Rx \propto x \Delta T = 25.25 \times 9.7\times10^{-6} \times 1 = \underline{0.0002 \text{ in.}}$

Axial thermal expansion of outer shell relative to inner shell:

$Lx \propto x \Delta T = 133 \times 9.7\times10^{-6} \times 1 = \underline{0.0012 \text{ in.}}$

The axial and radial differential thermal expansion of the lead relative to the inner and outer steel shells will be calculated for the most severe conditions (cold 2.6.2) and (fire 2.7.3.2). The normal conditions ($\Delta T = 7°F$) will not present any problem for either axial or radial expansion.

2.6.1.3 Stress Calculations

These thermal conditions are well below the heat stresses the cask was subjected to during initial pouring of the lead shield, and the above differential thermal expansions indicate no problem would be encountered.

2.6.1.4 Comparison with Allowable Stresses

Under normal conditions the temperature encountered will not result in any stresses above the yield point.

The inner shell is considered the containment vessel and can withstand the "normal condition" maximum internal pressure of 3 psig without exceeding the yield point. (See Section 2.5.2 and 2.6.3.)

2.6.2 Cold

The $-40^{o}F$ environment in still air will not adversely affect the normal operation of the cask. The temperature differentials across the various steel plates will be limited to several degrees F. and would not introduce any thermal stresses above the yield point of the material.

The outer cask shell is made of 304 L stainless steel, which has excellent impact and strength characteristics at $-40^{o}F$.

The three O-rings in the main cask closure and cask top vent plug, and the vent relief valve seat are all suitable for operation at $-40^{o}F$.

The cask is drained free of water before shipping, therefore, freezing of liquids will not be a problem.

The following analysis will evaluate the axial and radial differential thermal expansion of the load relative to the inner and outer steel shells to show the $-40^{o}F$ will not present a problem for the inner steel shell.

The axial thermal expansion will not present a problem since the colder temperature will increase the gap at the top of the lead shield, which can not exert any pressure on the steel shells in the axial direction. This gap would be approximately 1/2 inch, which would not present any shielding problem, since the lead shielding ring installed on the underside of the cask cover for lead slump would compensate for this small gap.

The radial thermal expansion will not affect the outer steel shell, since the lead to steel gap would increase on lower temperature.

2.6.2 Cold (continued)

The radial thermal expansion on the inner steel shell could present a problem and will be analyzed as follows:

α_s = coefficient of linear expansion of steel
9.7×10^{-6} in/in/^{o}F

α_l = coefficient of linear expansion of lead
16.4×10^{-6} in/in/^{o}F

diameter of inner steel shell = 36 inches
thickness of lead = 6 inches
thickness of inner steel shell = 1/4 inch
Poisson's ratio: steel 1/3; lead 0.45
tensile yield strength, lead = 800 psi
compressive yield strength, steel = 36,000 psi
E for steel = 29×10^6; E for lead = 2×10^6

Assume when cask was originally made the lead solidified at 620^{o}F and the lead and steel shells cooled together to ambient temperature.

Find radial interference from shrinkage of lead onto inner steel shell when lead and steel are cooled from 620^{o}F to -40^{o}F.

Radial shrinkage = $R \alpha \Delta T$
R.S. of lead = $18 \times 16.4 \times 10^{-6} \times 660$ = 0.195 in.
R.S. of steel = $18 \times 9.7 \times 10^{-6} \times 660$ = 0.115 in.
Radial interference = 0.080 in.

Assume all radial interference is taken in lead strain.

$$\frac{0.080}{18} \times 100 = 0.44\%$$

From WADC-TR57-695-Figure 12, at a lead strain of 0.44%, and a mean temperature of 290^{o}F $\left(\frac{620}{2} - \frac{40}{2} = 290\right)$, a radial lead stress would be developed equal to 250 psi.

2.6.2 Cold (continued)

An article appearing in the AIAA Journal, February 1965, entitled, "Influence of Cushion Stiffness on the Stability of Cushion Loaded Cylindrical Shells" by D.O. Brush and E.V. Pittner, indicates that with a lead outer cushion around the inner steel shell, the following minimum buckling pressure would be developed:

Equation 19 from above article:

$$\frac{P_o a}{E\,h} = (q^2 - 1)\left(\frac{(h/a)^2}{12(1 - v^2)}\right) + \frac{K_s\left(\frac{Es}{E}\right)\left(\frac{a}{h}\right)}{q^2 - 1}$$

$$P_o = \frac{Eh}{a}\left[(q^2 - 1) \times \frac{(h/a)^2}{12(1 - v^2)} + \frac{K_s\left(\frac{Es}{E}\right)\left(\frac{a}{h}\right)}{q^2 - 1}\right]$$

$$P_o = \frac{29 \times 10^6 \times 0.25}{18}\left[(q^2 - 1) \times \frac{(0.25/18)^2}{12(1 - 0.3^2)} + \frac{K_s\left(\frac{Es}{E}\right)\left(\frac{18}{0.25}\right)}{q^2 - 1}\right]$$

$$P_o = 400{,}000\left[(q^2 - 1)(1.8 \times 10^{-5}) + \frac{72\,K_s\left(\frac{Es}{E}\right)}{q^2 - 1}\right]$$

From WADC-TR57-695, Figure 11, Young's Modulus at 250 psi stress, 0.44% strain, and a mean temperature of 290°F is approximately 12,200.

In above equation 19, the factor Es/E = 12,200/29x10^6 = 0.00042.

$$P_o = 400{,}000\left[(q^2 - 1)(1.8 \times 10^{-5}) + 0.03\,\frac{K_s}{(q^2 - 1)}\right]$$

2.6.2 Cold (continued)

From Figure 7 of above AIAA Journal article, with h_s/a = 6/18 = 0.33, then the relationship of Ks to q is as follows:

K_s	q
3.5	7
3.75	7.5
4	8

Substitue K_s and q into above equation, and by trial and error determine smallest value of P_o, which will be minimum buckling pressure of inner steel shell by external compression of lead shielding.

	P_o	Ks	q
	1219	3.5	7
(min) =	1210	3.75	7.5
	1213	4	8

Since the smallest value of P_o = 1210 psi, and since this is above the radial stress on the steel of 250 psi, the steel shell will not buckle.

Find the radial stress on steel shell which will produce a fiber stress equal to compressive yield strength of steel shell (36,000 psi).

$$s = \frac{PR}{t} \; ; \quad P = \frac{st}{R}$$

$$P = \frac{36{,}000 \times 0.25}{18} = 500 \text{ psi}$$

Since the 500 psi is below the 1210 psi minimum buckling stress, the shell will start to yield in compression before it will buckle; but since the 500 psi is above the 250 psi shrinkage radial stress, the steel shell will not be stressed beyond its buckling point or beyond its compressive yield strength.

2.6.3 Pressure

Under Section 2.5.2 External Pressure, the cask was analyzed for 25 psi external pressure and results indicated all stresses were below the yield stress. Since for 0.5 atmospheric external pressure the cask inner shell will be completely backed up by lead, even lower stresses would be encountered. In addition, the O-ring seals will perform properly in either direction.

The external cask shell is constructed of heavy steel, 1 inch minimum, and would not be affected by such a low pressure.

It is concluded that the cask would not be affected adversely by a 0.5 atmospheric external pressure.

2.6.4 Vibration

The cask and contents are strongly designed and would not be affected by vibrations normally incident to transportation.

2.6.5 Water Spray

The outside of the cask is all steel and would not be affected by a water spray. This test is not required for an all steel exterior.

2.6.6 Free Drop (One Foot)

2.6.6.1 Drop Onto Solid End - One Foot

Available energy = WH = 70,000 x 12 = 0.84×10^6 in.-lbs.

Three of the 16 end buffer pipes extend 1/2 inch beyond the others in order to give buffer cushioning with a one foot drop.

For 3 pipe buffers on end of cask, each pipe would absorb:

$$\frac{0.84 \times 10^6}{3} = 280,000 \text{ in.-lbs.}$$

2.6.6 Free Drop (continued)

From Curve 2 of Appendix 2.10.1, this would result in a deflection of 0.4 inches.

Peak load for one buffer pipe at 0.4 in. deflection is 550,000 pounds. Therefore, deceleration factor is:

$$\frac{F}{W} = \frac{550,000 \times 3}{70,000} = 23\text{-}1/2 \text{ g}$$

,-SHEAR POINT **"d"**

W_1 = 6,000 lbs.w/ring

W_2 = 10,000 lbs.

W_3 = 5,000 lbs.

FIGURE WITHHELD UNDER 10 CFR 2.390

SHE4AR POINT **"b"**

Shear at Point "a"

Dynamic load = $(W_1 + W_2)$xg = 16,000 x 23.5 = 376,000 pounds.

Shear stress = $\frac{376,000}{\pi \times 36 \times 0.5}$ - 6649 psi

Steel ultimate shear strength : 35,000 psi

M.S. = 35,000 – 1 = 4.2 M.S.

Shear at Point "b"

Dynamic load = $(W_1 + W_2 + W_3)$xg = 21,000 x 23.5 = 493,500 pounds

Shear stress = $\frac{493,500}{\pi \times 36 \times 2.625}$ = 1,662 psi

M.S. $\frac{35,000}{1,662}$ - 1 = 20 M.S.

Bending of Solid End Outside Plate (Point "b")

Curve 2 of Appendix 2.10.2 shows an energy absorption curve for a 2-5/8 inch plate with a 37 inch span, which was scaled up from a scale model test described in Appendix 2.10.2.

Total energy = $(W_1+W_2+W_3) \times H = 21{,}000 \times 12 =$ 252,000 in.-lbs.

From Curve 2 this would give a deflection of 0.3 inches, which from the model test showed no bending failure.

Lead Slump

The ratio of available energy for the one foot end drop vs. the 30 foot end drop = $\dfrac{0.28\text{x}10^6}{25.2\text{x}10^6} = 0.01.$

The lead slump for the 30 foot drop was 4.1 inches. For the one foot drop the slump would be 0.01 x 4.1 = 0.04 inch, which would not present any problem in shielding.

2.6.6.2 Drop Onto Closure End - One Foot

$W1 = 5,000$ pounds
$W2 = 10,000$ pounds

$W3 = 6,000$ pounds

FIGURE WITHHELD UNDER 10 CFR 2.390

SHEAR POINT "b"

Shear at Point "a"

This end also has 3 pipe buffers extending 1/2 inch beyond the others for cushioning.

The shear stress in the inside plate of the top solid end would be similar to the stress calculated for the one foot drop onto the solid end, since the plate is also 1/2 inch, and the dynamic loading is similar. This gives a M.S. = 4.6.

Shear at Point "b"

Dynamic load $= (W_1+W_2+W_3)xg = \mathbf{21,000} \times 23.5 =$ 493,500 pounds

$$\text{Shear Stress} = \frac{493,500}{\pi \times 36 \times 1.5} = 2909 \text{ psi}$$

$$\text{M.S.} = \frac{35,000}{2,909} - 1 = 11 \text{ M.S.}$$

2.6.6.2 (continued)

Bending of Closure End Outside Plate (Point "b")
Curve 4 of Appendix 2.10.2 shows an energy absorption curve for a 1-1/2 inch plate with a 37 inch span, which was scaled up from a scale model test described in Appendix 2.10.2.

Total energy = $(W_1+W_2+W_3)xH$ = 21,000 x 12 = 252,000 inch pounds.

From Curve 4 this would give a deflection of 0.9 inches, which from the model test showed no bending failure.

Lead Slump
The lead slump, as shown on the one foot drop onto the solid end, would be 0.04 inch, which would not present any problem in shielding.

2.6.6.3 One Foot Side Drop - NOT on Trunnion

As shown on Dwg. 0999-D-07, Rev.G, the pipe shock absorber cover on the closure end has 2 inch long pipes on the sides of the cover (3 top and 3 bottom) between the longer 6-5/8 inch long pipes in line with the trunnions. These shorter pipes will absorb the energy from the one foot side drop and protect the cask closure cover flanges and bolts.

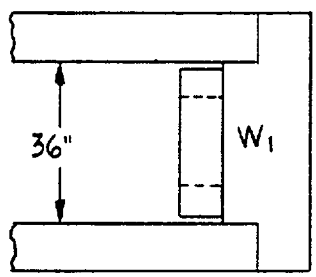
The available total energy on a one foot side drop = 70,000 x 12 = 0.84×10^6 inch pounds; one half of this (420,000 inch pounds) will be absorbed on these short pipe buffers. For a direct zero angle impact a single short pipe will absorb 420,000 inch pounds of energy with a deflection of 0.7 inches (see Curve 2 of Appendix 2.10.1); for a 22-1/2 degree angle impact a single short pipe will absorb 420,000 inch

2.6.6.3 (continued)

pounds of energy with a deflection of $\sim$ 1.8 inches (see Curve 4 of Appendix 2.10.1).

The solid end of the cask will absorb the energy by local steel deformation of the shock absorber cover and steel shell, which will be minor and not impair the safety of the cask in any way.

Although the cask closure flanges and bolts are protected from direct impact damage, a check is necessary to evaluate the dynamic reaction forces of the cover on the closure mechanism.



W_1 = 6,000 pounds with ring

Determine maximum "g" value by finding force for deflecting one short pipe with zero impact angle a distance of 0.7 inches. From Table 1 of Appendix 2.10.1:

Scale up factor for deflection is 2.8, therefore from scale model test data, a deflection on the full scale pipe of 0.7 inches would correspond to a scale model deflection of $\frac{0.7}{2.8}$ = 0.25 inches. This deflection would result from a load of 56,000 pounds on the scale model. The scale up factor for load is 10.2, therefore, the load to deflect the full scale pipe 0.7 inches would be = 56,000 x 10.2 = 570,000 pounds.

2.6.6.3 (continued)

The maximum "g" value would be for this end of the cask.

$$\frac{570,000}{1/2 \times 70,000} = 16 \text{ g.}$$

The average unit loading on the cask closure 3 inches deep step would be:

6000 x 16 = 96,000 pounds

From an inspection of Figure 10, the shielding lead could not be deformed until the heavy steel plates were deformed. Calculate shear stresses at cask closure step steel skirt as shown on Figure 10.

$1/2 \times \pi \times 36$ = length of steel in shear

0.5 = thickness of steel in shear

96,000 = total load on steel in shear

$$s = \frac{96,000}{1/2 \times \pi \times 36 \times 0.5} = 3395 \text{ psi shear}$$

$$\text{M.S.} = \frac{35,000}{3,395} - 1 = \underline{9.3 \text{ M.S.}}$$

As indicated on the cask drawing, the gap between the cask cover step O.D. and the cask I.D. wall is 0.125 inches, which is less than the cask cover closure bolt hole clearance gap of 0.38 inches. With the above stress, there would be no shearing or bending reaction on the cask closure bolts.

2.6.6.4 <u>One Foot Side Drop Onto Trunnion</u>

As shown on Figure 2, the side drop onto the trunnion will impact at the two ends of the cask and be taken up by the pipe side buffers; the trunnion area will not impact.

2.6.6.4 (continued)

Each end will impact on a single pipe with absorption of energy at each end:

$1/2 \times 70{,}000 \times 12 = 420{,}000$ inch pounds

From Curve 2 of Appendix 2.10.1 this will give a deformation of the pipe buffer of 0.7 inches.

FIGURE WITHHELD UNDER 10 CFR 2.390

Find the cask shell bending stress at the center.

As shown under paragraph 2.6.6.3, a deflection of a pipe buffer of 0.7 inches results from a force of 570,000 pounds, which is equal to "F" in sketch.

F = 570,000 pounds

s = maximum fiber stress at mid-point of shell

c = 50.5/2 = 25.25 inches.

$$M = \frac{Wl}{8} = \frac{2Fl}{8} = \frac{2 \times 570{,}000 \times 127}{8} = 18\times10^6 \text{ in.-lbs.}$$

$$\frac{\pi(d_1^4 - d_2^4)}{64} \quad \frac{\pi(50.5^4 - 48.5^4)}{64} \quad 47{,}600$$

$$\frac{MC}{I} \quad \frac{18\times10^6 \times 25.25}{47{,}600} = 9500 \text{ psi}$$

tensile yield strength of steel = 36,000 psi

$$\text{M.S.} = \frac{36{,}000}{9{,}500} - 1 = 2.8 \text{ M.S.}$$

The dynamic reaction force of the cask closure for this same loading was calculated under paragraph 2.6.6.3 and indicated the cask closure mechanism would not be impaired in any way.

CASK COVER END SHOCK ABSORBER RING

FIGURE WITHHELD UNDER 10 CFR 2.390

FIG. i

FIGURE WITHHELD UNDER 10 CFR 2.390

SIDE DROP ONTO TRUNNION

FIG. 2

2.6.6.5 Corner Drop Onto Solid End - One Foot Drop

From Figure 3 the corner drop onto the solid end will crush some of the buffer end pipes:
Available energy = 0.84×10^6 inch pounds.

Assume 2 inch deflection of first buffer pipe and 1 inch deflection of next two buffer pipes, as shown on Figure 3.

From Curve 4 of Appendix 2.10.1 the energy absorbed by the buffer pipes would be:

one buffer pipe 2 inch deflection	=	500,000 in.-lbs.
two buffer pipes 1 inch deflection	=	400,000 in.-lbs.
TOTAL	=	900,000 in.-lbs.

$$\text{M.S.} = \frac{900{,}000}{840{,}000} - 1 = \underline{0.07\ \text{M.S.}}$$

The three buffer pipes would absorb all of the available energy with no damage or deformation of the cask.

2.6.6.6 Corner Drop Onto Closure End - One Foot Drop

On the closure end one foot corner drop the buffer pipes would also absorb all of the available energy, similar to the solid end corner drop.

Analyze the closure bolts for stress beyond the yield point (if any).

2.6.6.6 (continued)

Determine deceleration in one foot corner drop:

V_{max} = 8 ft./sec.; V_{aw} = 4 ft./sec.

$S = V_{av}t$; $t = \frac{2/12}{4} = 0.042$ sec.

$V_{max} = at$; $a = \frac{8}{.042} = 190$ ft./sec.^{2}

Assume peak deceleration = 2 x average.

Peak deceleration = $\frac{2 \times 190}{32.2}$ = $\underline{11.8\ g}$

Load on closure bolts (hinge line rotation):

Weight of internals on cover = 15,400 pounds

12 bolts - 1 1/2 inch diameter

Total load on bolts = W x sin 21-1/2^{o}F x g

F = 15,400 x 0.93 x 11.8 = 169,000 pounds

INSIDE CASK CLOSED END

FIGURE WITHHELD UNDER 10 CFR 2.390

CORNER DROP

FIG. 3

CORNER DROP

2.6.6.6 (continued)

From calculations on 30 foot corner drop for hinge type rotation:

$169{,}000 \times 27 = 2P \times 5095 + 2916P$

$13{,}100P = 4.56 \times 10^6$

$\underline{P = 348}$

Maximum load on bolt #1 = 348 x 54 = 18,800 pounds

Area of bolt = 1.404 square inches

Maximum stress on bolt = $\frac{18{,}800}{1{,}404}$ = $\underline{13{,}400 \text{ psi}}$

This is well below the yield point of bolt.

Shear across bolts from one foot corner drop:

Total load = $W \times \sin 21\text{-}1/2^{o}F \times g$

$= 70{,}000 \times 0.3665 \times 11.8 = 303{,}000$ pounds

Area of 12 1-1/2 inch diameter bolts = 1.404 x 16 =

= 22.5 square inches

Maximum shear stress = $\frac{303{,}000}{22.5}$ = $\underline{13{,}460 \text{ psi}}$

This is well below the yield shear stress of bolts.

2.6.7 Corner Drop

Not applicable to this cask.

2.6.8 13 Pound Penetration Test - 40 Inch Drop

The only potentially vulnerable area on the cask which could be damaged by this test is the vent relief valve. This is located at the closure end of the cask near the corner, and will be covered by the impact limiter support skirt (1/2 inch steel plate); therefore, this is adequately protected and would not be damaged.

2.6.9 Compression

Not applicable to this cask.

2.7 Hypothetical Accident Conditions

2.7.1 Free Drop - 30 Feet

2.7.1.1 End Drop

2.7.1.1.1 Drop 30 Feet onto Solid End of Cask

Available energy = WH = 70,000 x 30 x 12 = $\underline{25.2 \times 10^6 \text{ in.-lbs.}}$

For 16 pipe buffers on end of cask, each pipe would absorb:

$$\frac{25.2 \times 10^6}{16} = 1.575 \times 10^6 \text{ in.-lbs.}$$

From Curve 2 of Appendix 2.10.1, this would result in a deflection of 3.8 inches on each pipe buffer.

Determine maximum "g" value by finding force for deflecting pipe buffers 3.8 inches.

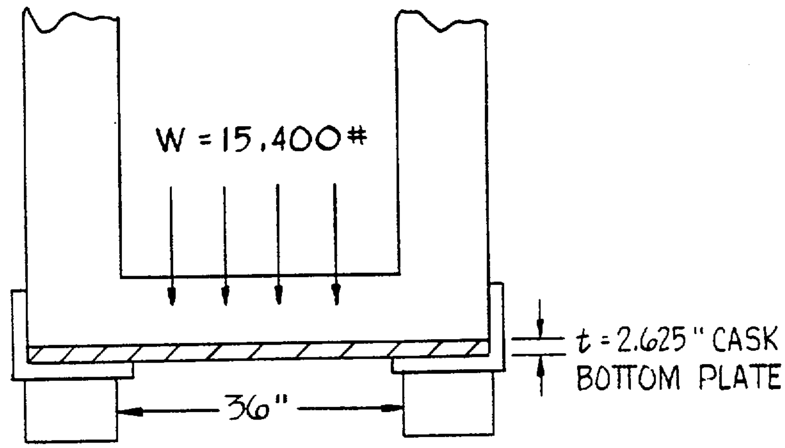
From Table 1 of Appendix 2.10.1:
Scale up factor for deflection is 2.8, therefore from scale model test data a deflection on the full scale pipe of 3.8 inches would correspond to a scale model deflection of $\frac{3.8}{2.8}$ = 1.36 inches.

This deflection would have received a maximum loading of 56,000 pounds on the scale model. The scale up factor for load is 10.2, therefore, the maximum loading during the deflection of the full size pipe to 3.8 inches would be = 56,000 x 10.2 = 570,000 pounds.

The maximum "g" value would be:

$$\frac{570{,}000 \times 16 \text{(pipes)}}{70{,}000} = \underline{130 \text{ g}}$$

Analyze strength of bottom plate:



Total load = W x g = 15,400 x 130 = 2×10^6 pounds

$$\text{Shear stress} = \frac{2\times10^6}{\pi \times 36 \times 2.625} = 6700 \text{ psi}$$

Ultimate shear strength = 35,000 psi

$$\text{M.S.} = \frac{35,000}{6,700} - 1 = \underline{4.2 \text{ M.S.}}$$

Curve 2 of Appendix 2.10.2 shows an energy absorption curve for a 2-5/8 inch plate - 37 inch span, which was scaled up from a scale model test described in Appendix 2.10.2. For a total absorbed energy of 5.5×10^6 in./lbs. (15,400 lbs. x 360 in. drop) the curve indicates a total deflection of 3.3 inches. With the bottom pipe buffers crushing 3.8 inches and the bottom plate of the cask deflecting 3.3 inches, the bottom of the cask in the middle will just about touch the ground. The bottom plate will not rupture under this loading.

The 4.1 inch lead slump would present a serious radiation streaming problem. To overcome this problem, Figure 4 shows a lead annulus ring which will be installed under the cask cover adjacent to the lead slump area. The lead ring, steel covered, will be bolted to the inside of the cask cover.

Lead Slump (End Drop Solid End)

Shield deformation from a 30 foot free drop on the flat end is determined from the analysis which follows based on impact causing the lead to settle. Based on L.B. Shappert, Cask Designer's Guide, ORNL-NSIC-68, February, 1970, page 63, the lead slumping would be as follows:

ΔH = lead slumping in inches

R = outer radius of lead in inches

r = inner radius of lead in inches

W = weight of cask in pounds

H = drop height in inches

t_s = outer shell thickness in inches

S_s = dynamic flow stress in steel in psi

S_{pb} = dynamic flow stress in lead in psi

$$\Delta H = \frac{RWH}{\pi(R^2 - r^2)(t_s S_s + R\ S_{pb})}$$

R = 24.25 inches

W = 70,000 pounds

H = 360 inches

r = 18.25 inches

FIGURE WITHHELD UNDER 10 CFR 2.390

RING WT. ~ 1000# D

LEAD SHIELDING RING
TO OVERCOME LEAD SLUMP

FIG. 4

2.7.1.1.1 (continued)

t_s = 1 inch (0.5 in. stainless, 0.5 in. carbon steel)

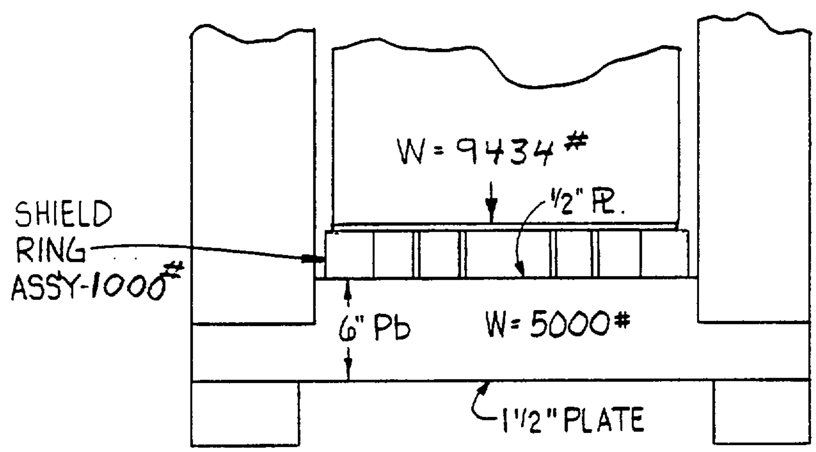
S_s = ultimate tensile S.S. = ultimate tensile C.S. = $\frac{71,200 + 55,000}{2} = 63,000$ psi

S_{pb} = 5000 psi

$$\Delta H = \frac{24.25 \times 70,000 \times 360}{\pi(24.25^2 - 18.25^2)(1 \times 63,000 + 24.25 \times 5000)}$$

ΔH = 4.1 inches lead slump

2.7.1.1.2 <u>Drop 30 Feet Onto Closure End of Cask</u>



B

Analyze strength of bottom plate.

The 15,400 load of cask internals and top cover is transmitted through the shield ring to the bottom cover. From paragraph 2.7.1.1.1 the "g" force is 130 g. The total dynamic load on shield ring is 15,400 x 130 = 2×10^6 pounds.

B

2.7.1.1.2 (continued)

The shield ring has an inner and an outer vertical 3/4 inch steel plate, the mean circumference is 30 inches. Total area of steel = x 30 x 2 x 0.75 = 141 sq. in.

$$\text{Stress in compression} = \frac{2 \times 106}{141} = 14{,}200 \text{ psi}$$

Steel has a compressive yield strength of 36,000 psi.

$$\text{M.S.} = \frac{36{,}000}{14{,}200} - 1 = \underline{1.5 \text{ M.S.}}$$

Shear area through two cover plates and lead with a circumference of 35 inches =

π x 35 x (1/2 in. + 1-1/2 in. steel) =
220 sq. in. steel area

π x 35 x (6 in. lead) = 660 sq. in lead area

Steel has a shear strength of 35,000 psi
Lead has a shear strength of 1825 psi

Steel strength	= 220 x 35,000 =	7.7×10^6	lbs.
Lead strength	= 660 x 1825 =	$\underline{1.2 \times 10^6}$	lbs.
	TOTAL	8.9×10^6	lbs.

$$\text{M.S.} = \frac{8.9 \times 10^6}{2 \times 10^6} - 1 = \underline{3.4 \text{ M.S.}}$$

2.7.1.1.2 (continued)

Bending of bottom cover plate from lead weight:

Curve 4 of Appendix 2.10.2 shows an energy absorption curve for a 1-1/2 inch plate - 37 inch span, which was scaled up from a scale model test described in Appendix 2.10.2. The available energy is 5000 pounds x 360 in. drop = 1.8×10^6 in.-lbs. The curve indicates an energy absorption of 1.7×10^6 in./lbs. with a deflection of the bottom plate of 3.3 inches (point at which middle of plate hits ground, see 1.7.1.1.1 for 3.8 inch deflection of pipe buffers). This is a close balance of energy, the small amount of excess energy will be absorbed by local lead deformation where bottom plate hits ground. The cask cover bottom plate will not rupture under this loading.

Lead slump (end drop onto cover end) see Figure 5.

Evaluate the type of lead movement involved at the solid end of the cask when dropped onto the closure end:

<u>Shear Break Along A-A</u> (Figure 5)
Shear strength of Chem. lead = 1825 psi
For a 1 inch segment at line A-A
Shear resistance = 1 x 5.25 x 1825 =
9580 pounds

LEAD SLUMP SOLID END

FIGURE WITHHELD UNDER 10 CFR 2.390

2.7.1.1.2 (continued)

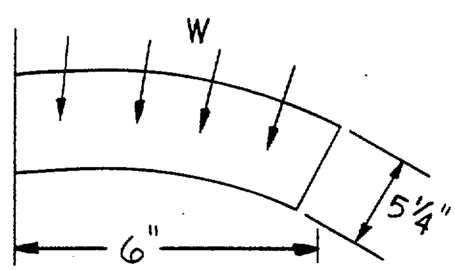
Tensile break along B-B (Figure 5)

Tensile strength of Chem. Lead = 1800 psi

For a 1 inch segment at line B-B

Tensile resistance = 1 x 6 x 1800 = 10,800 pounds

Bending Failure along C-C (Figure 5)



$W = wl$

Calculate resistance to bending for 1 inch segment in pounds.

$$M_o = \frac{(wl)\ l}{2} = \frac{Wl}{2}$$

$$S = \frac{M_o}{I}$$

$$c = \frac{5.25}{2} = 2.625$$

s = 800 psi (yield strength of lead)

$$I = \frac{bh^3}{12} = \frac{1 \times 5.25^3}{12} = 12$$

$$s = \frac{\frac{Wl}{2}(c)}{I}$$

$$W = \frac{sI \times 2}{lc} = \frac{800 \times 12 \times 2}{6 \times \frac{5.25}{2}} = 1220 \text{ lbs.}$$

Bending yield = 1220 pounds

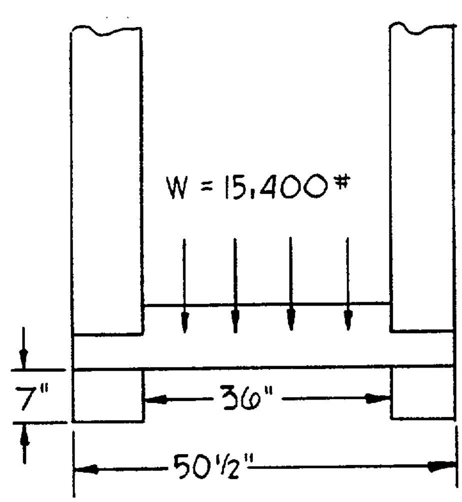
2.7.1.1.2 (continued)

The least strength is in bending, therefore, the lead would deform along a line such as C-C , Figure 5.

As shown on Figure 5, the minimum lead remaining by equating the volume of lead lost to a typical deformation pattern would be $\sim$ 3-1/2 inches, which is in excess of the minimum thickness of 3 inches required after the hypothetical accident conditions.

2.7.1.1.3 <u>"Cookie Cutter" Problem (End Drop)</u>

Evaluate maximum stresses and deformation on a 30 foot end drop when the impact area is not unyielding, but has a finite resistance.



U = total available energy 25.2×10^6 in.-lbs.

Total area of end of cask = 2002 sq. in.

16 pipes with cover plates with a total area of 552 sq. in.

σ = dynamic flow stress of earth, psi

X = inches penetration in earth of cask

X+7 = inches penetration of pipes

2.7.1.1.3 (continued)

$$U_{pipes} = 552 \times \sigma \times (7+X)$$
$$U_{bottom} = 1450 \times \sigma \times X$$
$$U_{pipes} + U_{bottom} = 552\,\sigma X + 3864\,\sigma + 1450\,\sigma X$$

$$U_{pipes} + U_{bottom} = 25.2\text{x}10^6 \text{ in.-lbs.}$$

$$25.2\text{x}10^6 = \sigma\,(3864 + 2002X)$$

$$\sigma = \frac{25.2\text{x}10^6}{3864 + 2002X}$$

Cookie Cutter Data

X = penetration of cask bottom into earth (in.)

σ = dynamic flow stress of earth (psi)

U = absorbed energy by cask bottom from earth (in.-lbs.)

σ x U = potential damage to cask from earth impact

$$\sigma = \frac{25.2\text{x}10^6}{3864 + 2002X}$$

U = 1450 UX

X(in.)	σ(psix103)	U(in.-lbs.x106)	σU(lbs2/inx109)
0	6.5	0	0
1	4.3	6.2	26.7
1.2	4.0	7.0	28.1
1.4	3.8	7.7	29.1
1.6	3.6	8.3	29.6
1.8	3.4	8.8	29.7
2.0	3.2	9.3	29.8
2.2	3.1	9.7	29.6
2.4	2.9	10.1	29.4
2.6	2.8	10.5	29.2
2.8	2.7	10.8	28.7
3.0	2.6	11.1	28.3
4.0	2.1	12.3	26.1
5.0	1.8	13.2	24.0

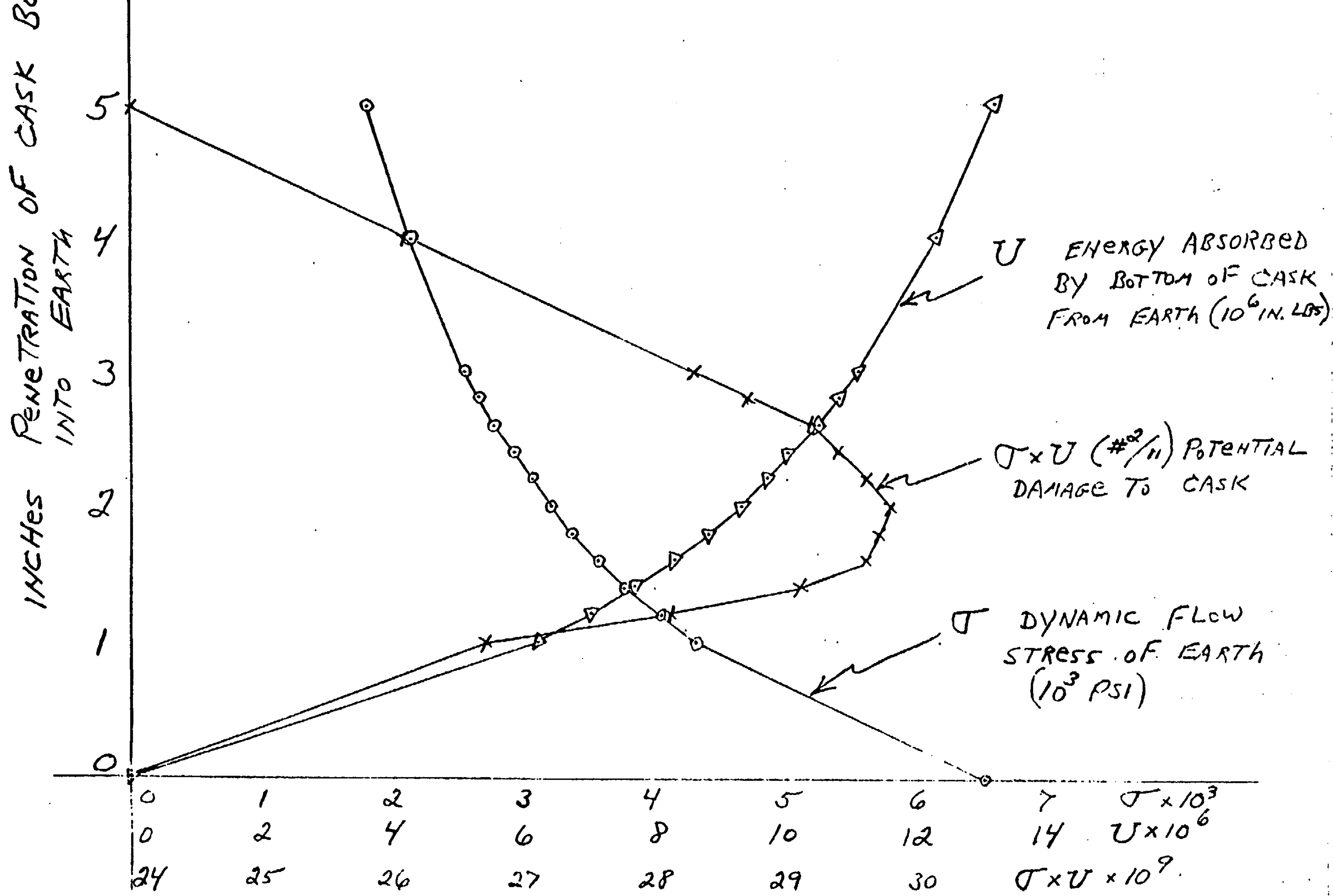
2.7.1.1.3 (continued)

The above data has been plotted in the Figure 6 curves. The dynamic flow stress of the earth (σ) is shown to be maximum when the bottom of the cask just "kisses" the earth (energy has been absorbed by pipe buffers), and no energy is available to damage the bottom of the cask. As the flow stress decreases the cask penetrates the earth to a greater depth and the cask bottom absorbs a large percentage of the available kinetic energy. More energy is absorbed by the cask bottom area as the earth dynamic flow stress is decreased, and at some point the earth pressure would be too low to damage the cask, however, the energy absorbed would be the greatest.

In order to evaluate the potential maximum damage to the cask, a factor of U was plotted and indicated a maximum peak at 2 inches penetration of the cask bottom into the earth. This point will be developed further to analyze for potential deformation or damage to the cask bottom.

FIG. 6

"COOKIE CUTTER" DATA



2.7.1.1.3 (continued)

The following data will be used in the analysis:

Dynamic flow stress of earth = 3200 psi.
Cask bottom penetrates earth = 2 inches
Available energy from earth on cask bottom = 9.3×10^6 in.-lbs.
Pipe buffers penetrate earth = 9 inches
Available energy from 30 foot drop = 25.2×10^6 in.-lbs.
Weight on bottom of cask = 15,400 pounds
Total weight of cask = 70,000 pounds
Area of pipe buffer ends = 552 sq. in.
Area of bottom of whole cask = 2002 sq. in.

<u>Penetration of pipe buffers into earth up to point that cask bottom touches earth - (7 inches)</u>.
Energy absorbed by pipe buffers = σ x area of pipe buffers x penetration.
$3{,}200 \times 552 \times 7 = \underline{12.4 \times 10^6 \text{ in.-lbs.}}$
Kinetic energy varies as square of velocity.

$$\frac{25.2 \times 10^6}{(25.2 - 12.4) \times 10^6} = \frac{44^2}{V_1^2}; \quad V_1^2 = 983$$

V_1 = 31 ft./sec. = velocity of cask after pipe buffers have penetrated earth 7 inches.

$$V_{av} = 44 - \left(\frac{44 - 31}{2}\right) = 37\text{-}1/2 \text{ ft./sec.}$$

$$S = V_{av}t; \quad t = \frac{7/12}{37.5} = 0.0155 \text{ sec.}$$

$$(44 - 31) = at; \quad a = \frac{13}{.0155} = 839 \text{ ft./sec.}^2$$

Assume peak deceleration = 2 x average

$$\text{Peak deceleration} = \frac{2 \times 839}{32.2} = \underline{52\ g}$$

2.7.1.1.3 (continued)

<u>Penetration of total cask bottom area an additional 2 inches.</u>

Energy absorbed = $(25.2 - 12.4)10^6$ = $12.8x10^6$ in.-lbs.

V_{max} = 31 ft./sec.

V_{av} = 15-1/2 ft./sec.

$S = V_{av}t; \quad t = \frac{2/12}{15.5} = 0.011$ sec.

$V_{max} = at; \quad a = \frac{31}{0.011} = 2818$ ft./sec.

Assume peak deceleration = 2 x average

Peak deceleration = $\frac{2 \times 2818}{32.2}$ = <u>175 g</u>

<u>Maximum loading on cask bottom (36 in. diameter area).</u>

$(\sigma \times \pi \times 18^2) - (15{,}400 \times 175) = 3.25x10^6$
$- 2.70x10^6$ = <u>$0.55x10^6$ lbs.</u>

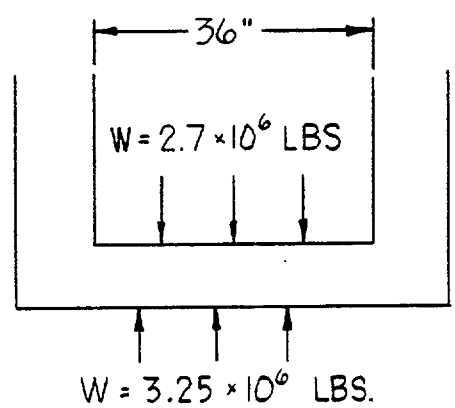
550,00 pounds net force pushing bottom into cask interior.
Net energy = F x penetration = $0.55x10^6$ x 2
= $1.1x10^6$ in.-lbs.

Referring to Figure 2 and Curve 2 of Appendix 2.10.2, which simulates the cask top and bottom outside plates (2-5/8 inch and 1-1/2 inch plate), assume the net energy pushing up on the bottom of the center area of the cask (1.1x106 in.-lbs.) by the earth is absorbed

2.7.1.1.3 (continued)

in deformation of these two plates. The bottom plate is pushed in by the earth, and this load is transferred through the lead and cask wall to the top cask outside plate. From Curve 2 this would give a total deformation of 0.8 inch in the middle section of each plate.

Evaluate the lead in the bottom end of the cask, which might deform instead of transferring the load to the upper end of the cask.



Maximum load = 3.25×10^6 pounds

$$\text{Stress on lead} = \frac{3.25 \times 10^6}{\pi \times 18^2} = 3195 \text{ psi}$$

This is less than the dynamic flow stress of lead (5,000 psi), and the lead would not significantly deform, except in a small local area.

It is concluded that the "cookie cutter" problem would not seriously damage the cask, and would only result in some minor deformation of the cask top and bottom plates ($\sim$0.8 inch).

2.7.1.2 <u>Side Drop 30 Feet</u>

2.7.1.2.1 <u>Drop NOT on Trunnion Area</u>

Calculate maximum loss of shielding from pages 58 - 62 of ORNL-NSIC-68:

$$\frac{WH}{t_s R L \sigma_s} = \left[F_1(\theta) \right] \left[\frac{R}{t_s}(\sigma_{pb}/\sigma_s) + 2\left(\frac{R}{L}\right)(t_e/t_s) \right] + F_2(\theta)$$

W = total weight (lbs.) = 70,000 lbs.

H = total drop height (in.) = 360 in.

t_s = outer shell thickness (in.) = 1 in.

R = outer radius (in.) = 25.25 in.

L = cask length (in.) = 134 in.

σ_s = dynamic flow stress in steel (psi) = 50,000 psi

σ_{pb} = dynamic flow stress in lead (psi) = 5,000 psi

t_e = thickness of steel end plate (in.) = 1.5 in.

$$\frac{WH}{t_s R L \sigma_s} = \frac{70,000 \times 360}{1 \times 25.25 \times 134 \times 50,000} = 0.149$$

$$\frac{R}{t_s}(\sigma_{pb}/\sigma_s) + 2\left(\frac{R}{L}\right)(t_e/t_s) = \frac{25.25}{1}\left(\frac{5,000}{50,000}\right) + 2\left(\frac{25.25}{134}\right)\left(\frac{1.5}{1}\right) = 3.09$$

From Figure 2.21 Nomograph in reference,

θ = 23 degrees

Maximum loss of shielding = d = $R(1 - \cos\theta)$ = $25.25(1 - \cos 23^{o})$ = 2 in.

As previously established, 3 inches of lead can be lost maximum.

M.S. = $\frac{3}{2} - 1$ = <u>0.5 M.S.</u>

Evaluation of Cask Cover Not Breaking Open

Energy to absorb on closure end of cask:

$1/2 \times 70{,}000 \times 30 \times 12 = 12.6 \times 10^6$ in.-lbs.

Assume steel on side of closure end is crushed down to a line 1-1/2 in. above the side of cask body. Further, assume 60% of steel volume crushed is absorbing energy at a rate of 50,000 in.-lbs. per cubic inch of steel (dynamic flow stress of steel); and the remaining 40% of steel volume is subject to bending and shear energy of steel which is assumed to absorb energy at the rate of 5000 in.-lbs. per cubic inch of steel.

Chord area $= \frac{r^2}{2} (\theta - \sin\theta)$.

	Thickness (in.)	Radius (in.)	Included Angle θ	Chord Area (sq.in.)	Vol. (cu.in.)	Remarks
1) 3-short pipe buffers	-	-	-	-	102	
2) pipe buffer end plate	1/2	29-3/4	46	37	19	
3) pipe buffer side plate	1/2	29-3/4	46	-	155	13 in. long
4) cask cover flange	1	29	43	29	29	
5) block under cover flange	1	-	-	5	5	2 = 1/2 blocks
6) two cask ring flanges	1/2x(2)	29	43	29	29	2 rings
7) load distributor blocks	2-1/2x4x7	-	-	-	70	1 block
TOTALS					409 cu. in.	

Total energy absorbed:

$409 \times 0.6 \times 50{,}000 = 12.3 \times 10^6$

$409 \times 0.4 \times 5{,}000 = \underline{0.8 \times 10^6}$

TOTAL $= 13.1 \times 10^6$ in.-lbs.

Available energy = 12.6×10^6 in.-lbs.

Distance crushed to side of cask = 5-1/2 in.

For calculating deceleration use distance of 5-1/2 in.

For 30 foot drop:

$V_{max} = 44$ ft./sec.

$V_{aw} = 22$ ft./sec.

$$S = V_{aw}t; \quad t = \frac{5.5/12}{22} = 0.021 \text{ sec.}$$

$$V_{max} = at; \quad a = \frac{44}{0.021} = 2095 \text{ ft./sec.}^2$$

Assume peak deceleration = 2 x average =

$$\frac{2 \times 2095}{32.2} = \underline{130\ g}$$

Cask closure weights approximately 5000 lbs.

Dynamic shear force of closure = 5000 x 130 = 650,000 pounds.

Shear resistance is primarily steel in closure step; this is 1/2 in. steel cylinder in shear plus closure lead in step in compression.

Shear of 1/2 of circular step steel in closure =
1/2 x 0.5 x   x 36 x 35,000 = 989,000 pounds.

Neglecting compression of lead, the shear resistance of the steel step cylinder in the closure is more than strong enough to resist the dynamic shear force of 650,000 pounds from the closure.

In addition, the added strength of the blocks under the cover flange (Item 3 in Figure 10) greatly increase the shear resistance of relative side movement of the cover relative to the cask opening. These blocks are used at the closure flange under each bolt location to provide a key block for locking the cover to the side of the cask.

With this design, it is conservative to conclude that no shear or bending stress would be imposed on the uncrushed closure bolts, and one to three closure bolts would be damaged on a side impact (either not on the trunnion area, or on the trunnion area), but that the remaining 9 to 11 bolts would not be damaged and would assure that the cask cover remain in place.

2.7.1.2.2 Drop onto Trunnion

See Figures 1, 2, 7, 8, and 9 for details of two end buffers and trunnion buffer; Figure 9 shows sketch of side drop onto trunnion.

Calculate energy absorption with metal deflection to line A-A shown on Figure 2.

Energy absorption at area "X" to line A-A

Buffer pipes will crush such that the center pipe deflection will be 5-7/8 in., and the two side pipes will deflect 5 inches.

From Curves 2 and 4 of Appendix 2.10.1 the energy absorbed by the buffer pipes would be:

center pipe = 5-7/8 in. deflection =

$1.65 \times 10^6 \times \frac{5.875}{4} = \underline{2.42 \times 10^6 \text{ in.-lbs.}}$

Extrapolating from 4 in. to 5-7/8 in. for energy absorption is conservative because load is rapidly rising at 4 in., as shown on Curve 1 of Appendix 2.10.1, which would provide a higher specific energy absorption.

two side pipes = 5 in. deflection = $1.9 \times 10^6 \times 2 = \underline{3.8 \times 10^6 \text{ in.-lbs.}}$

Total U to line A-A area X = $\underline{6.2 \times 10^6 \text{ in.-lbs.}}$

FIGURE WITHHELD UNDER 10 CFR 2.390

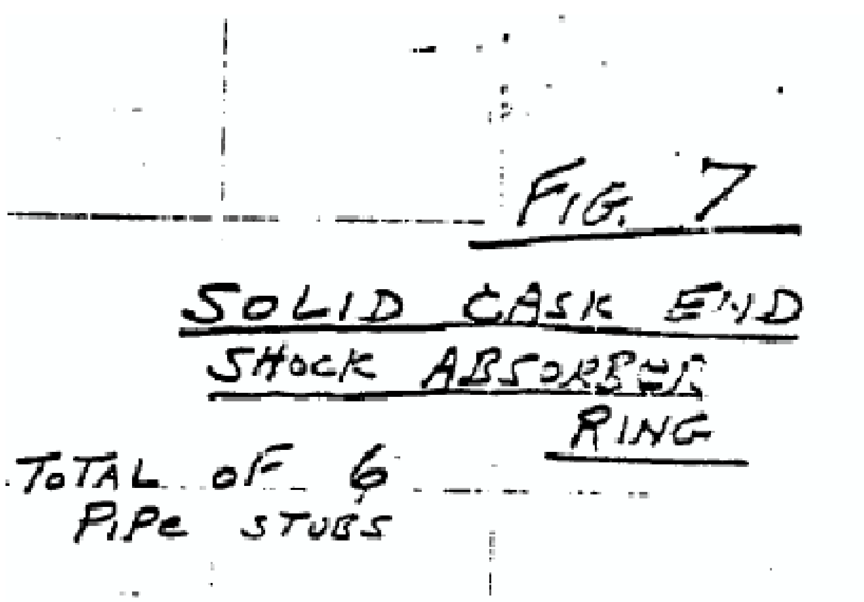


FIG. 7

FIGURE WITHHELD UNDER 10 CFR 2.390

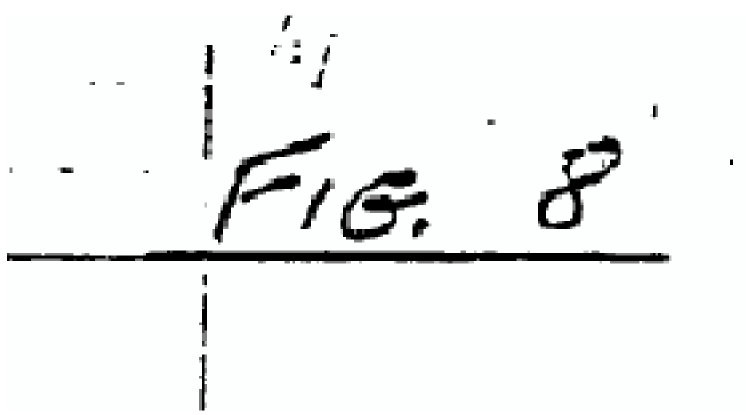


FIG. 8

FIGURE WITHHELD UNDER 10 CFR 2.390

FIG. 9

SCALE 1/4" = 1"

FIG. 10

FIGURE WITHHELD UNDER 10 CFR 2.390

2.7.1.2.2 (continued)

Energy Absorption at Area "Z" to line A-A

Buffer pipes will crush such that the center pipe deflection will be 2-1/8 in. and the two side pipes will be 1-1/4 in.

From Curves 2 and 4 of Appendix 2.10.1, the energy absorbed by the buffer pipes would be:
center pipe = 2-1/8 in. deflection = 1.07×10^6 in.-lbs.
two side pipes = 1-1/4 in. deflection = $0.265 \times 10^6 \times 2 = 0.53 \times 10^6$ in./lbs.

Total U to line A-A at area Z = $\underline{1.6 \times 10^6 \text{ in.-lbs.}}$

Energy Absorption at Area "Y" to line A-A

Four gussets on trunnion buffer (see Figure 8 and 9), will crush down 2 in. to the top of the trunnion; the absorbed energy of deflection will be:
$4 \times 4 \times 2 \times 1 \times 50{,}000 = \underline{1.6 \times 10^6 \text{ in.-lbs.}}$

Total absorbed energy to line A-A

Area X = 6.2×10^6
Area Y = 1.6×10^6
Area Z = $\underline{1.6 \times 10^6}$
$\underline{9.4 \times 10^6 \text{ in.-lbs. to line A-A}}$

Energy remaining to be absorbed =
$(25.2 - 9.4) \times 10^6 = 15.8 \times 10^6$ in.-lbs.

B

2.7.1.2.2 (continued)

Calculate energy absorption to line B-B shown on Figure 2.

<u>Energy absorbed at Area "X" to line B-B</u>

Deflection line is 2-1/2 in. lower than line A-A, and is 2-1/2 in. above outside surface of cask. Energy will be absorbed by crushing of steel closure flanges, gussets, and fittings in the localized impact area plus deformation of a small area of lead under the impact area. The total volume of steel displaced in the impact zone will be calculated; 60% of the B steel volume will be used to calculate energy absorbed by steel deformation at 50,000 psi dynamic flow stress, the other 40% of the steel volume will be applied to B lead deformation under the impact area at 5000 psi dynamic flow stress for lead. This appears to be a conservative assumption.

<u>Steel displaced (see Figures 1 and 2)</u>

Calculate the chord areas and multiply by thickness to obtain steel volume.

Chord area $= \frac{r^2}{2} (\theta - \sin\theta)$ B

	Thickness (in.)	Radius (in.)	Included Angle θ	Chord Area (sq.in.)	Vol. (cu.in.)
1) pipe buffer end plate	1/2	29-3/4	40	24	12
2) cover flange	1	29	36	18	18
3) cask flange	1/2	29	36	18	9
4) 2-1/2x4x7 load distributor block (1/3 block)	2-1/2	-	-	-	23
5) cask flange	1/2	29	36	18	9
6) 2 pipe buffers 2-1/2 in. long	-	-	-	-	68
TOTAL					139

B

Energy absorbed by steel deformation =
139 x 0.6 x 50,000 = $\underline{4.1\text{x}10^6 \text{ in.-lbs}}$.
Energy absorbed by lead displacement =
139 x 0.4 x 5,000 = $\underline{0.3\text{x}10^6 \text{ in.-lbs}}$.

At area "X" net increase in U from line A-A to B-B = $\underline{4.4\text{x}10^6 \text{ in.-lbs}}$.

Energy Absorption at Area "Z" to line B-B
Buffer pipes will crush such that the center pipe will deflect from the previous 2-1/8 in. to a depth of 4-5/8 in., and the two side pipes will deflect from 1-1/4 in. to 3-3/4 in., an additional 2-1/2 inches.

From Curves 2 and 4 of Appendix 2.10.1, the energy absorbed by this additional deflection will be:

center pipe = *4-5/8 in. = $1.91\text{x}10^6$ in.-lbs.
2-1/8 in. = $\underline{1.07\text{x}10^6}$ in.-lbs.
NET GAIN = $0.84\text{x}10^6$ in.-lbs.

* 4-5/8 in. is extrapolated conservatively from 4 in. = $1.65 \text{ x} \frac{4.625}{4}\text{x}10^6$ in.-lbs. = $1.91\text{x}10^6$ in.-lbs.

Net increase in energy absorbed =
$\underline{0.84\text{x}10^6 \text{ in.-lbs}}$.

side pipes = 3-3/4 in. = $1.25\text{x}10^6$ in.-lbs.
1-1/4 in. = $\underline{.265\text{x}10^6}$ in.-lbs.
NET GAIN/PIPE = $.985\text{x}10^6$ in.-lbs.

For 2 pipes = 2 x $0.985\text{x}10^6$ =
$2.8\text{x}10^6$ in.-lbs.

At area "Z" net increase in U from line A-A to B-B = $\underline{2.8\text{x}106 \text{ in.-lbs}}$.

2.7.1.2.2 (continued)

Energy Absorption at Area "Y" to line B-B (see Figure 2).

The side drop onto the trunnion from 30 feet is expected to initally crush the four gusset plates, and then pick up the main load on the 10 in. pipe and the trunnion. The steel trunnion buffer will transmit the energy into the main wall of the cask, the energy being absorbed mainly in deformation of lead, plus some in steel yielding. The 10 in. pipe in the buffer assembly will prevent the trunnion from penetrating the steel shell, the heavy 1 in. buffer base plate will move with the trunnion and prevent a shear movement of the trunnion relative to the cask shell.

The forces involved in this section are calculated as follows:

The volume of lead displaced in Zone A as shown on Figure 9, based on a 2-1/2 in. displacement is:

12 x 12 x 2-1/2 = 360 cu. in.

The volume of lead in Zone B is:

$(24^2 - 12^2) \times 1.875 =$ 810 cu. in.

The volume of lead in Zone C is:

$(36^2 - 24^2) \times 0.625 =$ 450 cu. in.

Total lead displacement = 1620 cu. in.

With a dynamic flow stress for lead of 5,000 psi, the energy absorbed by the lead deformation is:

$1620 \times 5000 = \underline{8.1\text{x}10^6 \text{ in.-lbs. lead deformation}}$.

For a movement of 2-1/2 in. to absorb $8.1\text{x}10^6$ in.-lbs. of energy, the average force in pounds applied to the trunnion area is:

$8.1\text{x}10^6/2.5 = \underline{3.24\text{x}10^6 \text{ lbs. force}}$

The steel surface of contact is the end of the trunnion and the end of the 10 in. pipe.

trunnion area = 50
pipe area = 34
84 sq. in.

The average pressure on this steel area is:

$3.24\text{x}10^6/84 = 38{,}570$ psi

Since this is below the dynamic flow stress of steel (50,000 psi), this pressure will not deform the trunnion or pipe steel, but will be transmitted into the steel support plate and cask shell where there will be local bending, and principally transmitted into lead deformation.

By dividing the area of lead under the 24 in. x 24 in. trunnion buffer support 1 in. plate by this pressure $\sim 3.24\text{x}10^6/24^2 =$ 5625 psi, which is slightly above the dynamic flow stress of lead, 5000 psi.

2.7.1.2.2 (continued)

This is estimated to be a conservative calculation of the energy absorption in the trunnion area. No allowance has been made for deformation of the cask shell and buffer support plate steel, which could account for 20% to 30% of the calculated lead energy absorption.

The reduction in lead shielding thickness is not a problem, since it has been shown that three inches of shielding could be removed without exceeding the accident condition limiting dose rate through the cask. Whereas the sketch in Figure 9 shows lead removed from the shielding thickness, in the actual case the reduction in shielding thickness would be less, since the inner wall of the cask would bend into the cask cavity rather than deforming all of the lead in a side-wise direction.

A check of the shear forces on the trunnion welds to the shell plates indicate the trunnion would not penetrate the cask shell by shear failure. The 10 in. pipe and buffer assembly is specifically designed to prevent any significant shear force on the trunnion welds, since the impact surfaces are such that the trunnion and the shell plates are displaced together.

2.7.1.2.2 (continued)

Assuming all of the force was imposed on the trunnion, and no load was picked up by the pipe, the maximum shear force on the trunnion welds would be:

Shear area = $\pi \times 8 \times 1.5 = 37.7$ sq. in.

Force is 3.24×10^6 lbs.

Shear stress = $3.24 \times 10^6/37.7$ = <u>86,000 psi</u>

This is over twice the ultimate shear strength of 35,000 psi, however, such a load could not be put on the trunnion with the trunnion buffer in place.

Summarizing the energy absorption for deformation between line A-A and line B-B on Figure 2:

Area "X"	= 4.4×10^6 in.-lbs.
Area "Z"	= 2.8×10^6 in.-lbs.
lead Area "Y"	= 8.1×10^6 in.-lbs.
steel Area "Y" (20% of lead)	= 1.6×10^6 in.-lbs.
	16.9×10^6 in.-lbs.

A summary of the energy balance for the 30 foot side drop on the trunnion is:

	cover end "X"	trunnion "Y"	solid end "Z"
Energy to line A-A ($\times10^6$ in.-lbs.)	6.2	1.6	1.6
Energy from line A-A to line B-B	4.4	9.7	2.8
TOTAL ENERGY (in.-lbs.)	10.6	11.3	4.4

NET TOTAL = 26.3×10^6 in.-lbs.

Available energy = 25.2×10^6 in.-lbs., which indicates a conservative energy balance and the energy has been accounted for.

2.7.1.2.2 (continued)

Evaluation of Cask Cover Not Breaking Open

In the previous evaluation of a 30 ft. side drop not on the trunnion area, it was shown that the cask cover closure mechanism would not fail with a 5-1/2 in. side crushing and 130 g. shock loading.

Since the crushing distance is greater (8-3/8 in.) for this evaluation of a 30 foot drop onto the trunnion area, the shock loading will be less and the closure mechanism will not fail.

2.7.1.3 <u>Corner Drop - 30 Feet</u>

2.7.1.3.1 <u>Corner Drop Onto Solid End of Cask</u>

To line 1-1 (Figure 3), which is minimum lead shielding requirement line.

<u>Pipe stubs crushed</u>

Pipes 5 through 8 and their counter parts, plus pipe 9 (total of 9 pipes) deflect fully; assume full length deflection of 6-5/8 in. and extrapolate up from 5 in. to 6-5/8 in. on Curve 4 of Appendix 2.10.1, total absorbed energy curve. From inspection of Figure 3, this is a very conservative assumption, since no credit will be taken for energy absorbed in buffer pipe support plates as they are deformed, and from Curve 3 of Appendix 2.10.1, the load in kips is rising faster above 4 in.

Pipe 4, and its counter part will deflect 2-1/2 in. axially to pipe, which is 3-1/2 in. deflection along axis of loading.

Pipe deflection 21-1/2^{o} loading:
9 pipes deflect 6-5/8 in.
2 pipes deflect 3-1/2 in.

From Curve 4 of Appendix 2.10.2:

$$9 \times \frac{6.625}{5} \times 1.9\text{x}10^{6} = 22.6\text{x}10^{6} \text{ in.-lbs.}$$

$$2 \times 1.3\text{x}10^{6} = \underline{2.6\text{x}10^{6} \text{ in.-lbs.}}$$

$$\text{PIPES TOTAL} = 25.2\text{x}10^{6} \text{ in.-lbs.}$$

The available energy is $25.2\text{x}10^{6}$ in.-lbs., which is equal to the absorbed energy of the pipe buffers.

2.7.1.3.1 (continued)

There would be additional energy absorbed in the deformation of the steel and lead, approximately equal to the pipe buffer absorbed energy; therefore, the real deflection line on the corner drop would be considerably less than the maximum allowable for displacement of lead, probably about to line 2-2 as shown on Figure B.

The corner drop onto the solid end would be acceptable.

2.7.1.3.2 <u>Corner Drop Onto Closure End of Cask</u>

The corner drop onto the closure end must be analyzed for hinged type rotation failure of the cover bolts and gross shearing of the cover.

The reduction of shielding is not a problem from inspection of the results above for the corner drop onto the solid end; especially so, since a lead ring is added to the inside of the cover to compensate for the lead slump from an end drop, and this would more than compensate for any loss of lead resulting from a corner drop.

<u>Hinged Type Rotational Failure of Cover Bolts</u>

Assume hinge line of rotation is through one closure bolt, which is conservative (see Figure 1).

2.7.1.3.2 (continued)

Weight of internals and cover tending to open cover:

Internals	9,434	
Cover	4,965	
Lead Ring	1,000	
TOTAL	15,399	- say 15,400 pounds

Total deflection in corner drop is 10 in., see Figure 1.

Deceleration in corner drop:

For 30 ft. drop:

V_{max} = 44 ft./sec.; V_{av} = 22 ft./sec.

$$S = V_{av}t; \quad t = \frac{10/12}{22} = 0.038 \text{ sec.}$$

$$V_{max} = at; \quad a = \frac{44}{0.038} = 1160 \text{ ft./sec.}^2$$

Assume peak deceleration = 2 x average

$$\text{Peak deceleration} = \frac{2 \times 1160}{32.2} = \underline{72\ g}$$

Load on closure bolts:

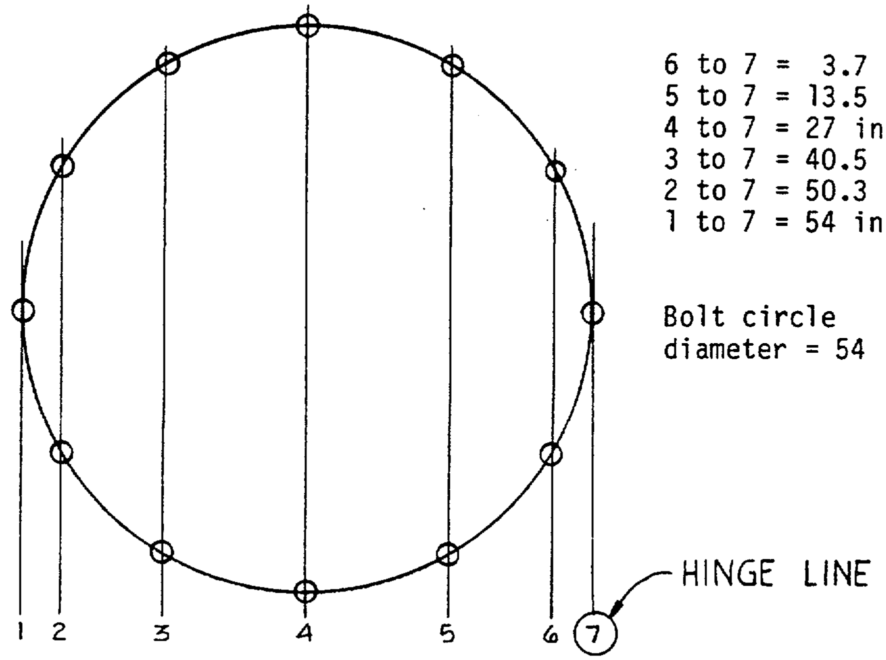
Bolts = (12) 1-1/2 in. diameter bolts

F_{Tu} = 160,000 psi

Total load on bolts = W cos 21-1/2^o x g

F = 15,400 x 0.93 x 72 = $\underline{1.03\text{x}10^6 \text{ lbs.}}$

2.7.1.3.2 (continued)



6 to 7 = 3.7 in.
5 to 7 = 13.5 in.
4 to 7 = 27 in.
3 to 7 = 40.5 in.
2 to 7 = 50.3 in.
1 to 7 = 54 in.

Bolt circle diameter = 54 in.

$1.03 \times 10^6 \times 27 = 2P(3.7^2 + 13.5^2 + 27^2 + 40.5^2 + 50.3^2) + P \times 54^2$

$2.78 \times 10^7 = 2P \times 5095 + 2916P$

$1.31 \times 10^4 \; P = 2.78 \times 10^7 \quad \underline{P = 2120}$

Maximum load is on bolt #1 = 2120 x 54 = <u>114,500 pounds</u>

Bolt ultimate strength = 1.404 x 160,000 = 225,000 pounds

Margin of Safety $= \frac{225{,}000}{114{,}500} - 1 = \underline{0.96 \text{ M.S.}}$

It is noted that the closure end pipe buffer assembly will be bolted onto the cask using the cover bolting flange, which will effectively add 6 additional 1 in. diameter bolts holding cover to cask.

2.7.1.3.2 (continued)

Shear Across Closure Cover

Shear force:

$F = W \times \sin 21\text{-}1/2^{o} \times g$

$= 70{,}000 \times 0.3665 \times 72 = \underline{1.85\times10^{6}\ \text{lbs}}.$

Shear resistance of pipe energy absorber skirt:

skirt diameter 59 in.

skirt thickness - 1/2 in. carbon steel (see Figure 1)

skirt metal in shear $\pi \times 59 \times 1/2 = 92.6$ sq. in.

F_{su} carbon steel = 35,000 psi

Shear strength of skirt = 92.6 x 35,000 = $\underline{3.24\times10^{6}\ \text{lbs}}.$

Margin of Safety = $\frac{3.24\times10^{6}}{1.85\times10^{6}} - 1 = \underline{0.75\ \text{M.S.}}$

Additional Shear Resisting Items

(6) 1 in. bolts on pipe skirt (Figure C)

3 guide pins in cover

cover support plates and bolting clips

lid cover step

(12) 1-1/2 in. cover bolts

2.7.1.4 Oblique Drops

From inspection of the drawings there is no need to analyze for oblique drops other than the corner drops evaluated above. The two ends of the cask are protected by pipe buffer assemblies; and drops of different orientations would not be as severe as the drops already analyzed.

2.7.1.5 Summary of Results

a) Drop 30 Feet Onto Solid End of Cask

Damage - (1) Bottom end plate of cask will yield and deflect until it just about touches ground, no tearing or rupture of shell plates, (2) lead slump 4.1 inches, compensated by new lead ring bolted to inside cover.

b) Drop 30 Feet Onto Closure End of Cask

Damage - (1) Bottom end plate of cask will yield and deflect until it touches ground, (2) some lead will be deformed in bottom cover, but less than the minimum allowable amount, (3) lead slumping in solid end will bend down from corner of cask and minor loss of shielding, (4) some local deformation of closure flange could result in minor leakage through O-rings.

c) "Cookie Cutter" 30 Foot Drop

Damage - Some yielding and deformation (0.8 in.) of the middle of both ends of the cask in an upward direction.

d) 30 Foot Side Drop NOT Onto Trunnion Area

Damage - (1) loss of 2 in. of shielding at side impact line (3 in. loss is allowable), (2) some local deformation of closure flange could result in minor leakage through O-rings.

e) 30 Foot Side Drop Onto Trunnion Area

Damage - (1) loss of 2-1/2 in. lead shielding around trunnion area (3 in. loss is allowable), (2) some potential leakage through O-rings due to local deformation of closure flange.

2.7.1.5 (continued)

f) <u>30 Foot Corner Drop Onto Solid End of Cask</u>
Damage - (1) some lead lost in corner but less than allowable amount, (2) steel deformation at corner area, but no tearing or rupture of outer cask shell.

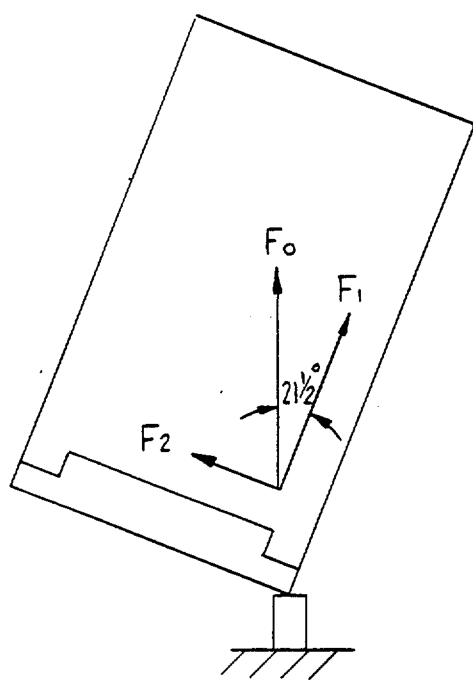
g) <u>30 Foot Corner Drop Onto Closure End of Cask</u>
Damage - (1) bolt loss of impact area, no loss of bolts due to hinge type rotation of cover, (2) some potential leakage through O-rings due to local deformation of closure flange, (3) minor lead loss in impact corner, but new lead ring added inside cover would more than offset this loss of lead.

2.7.2 Puncture - 40 Inch Drop

2.7.2.1 Corner Drop Onto Puncture Pin

Evaluation of additional damage (if any) resulting from a puncture drop in the same vicinity as the 30 foot corner drop. The worse case would be on the closure end and the possible reaction on the cover; the solid end has a 2-5/8 in. thick end plate, which would not be damaged to any significant extent by the 6 in. punch.

Assume maximum loading on the 6 in. punch, equivalent to 50,000 psi dynamic flow stress.



$F_o = 50,000 \times \pi 3^2 = 1.4 \times 10^6$ lbs.

$F_1 = 1.4 \times 10^6 \times \cos 21\text{-}1/2^o =$
1.3×10^6 lbs.

$F_2 = 1.4 \times 10^6 \times \sin 21\text{-}1/2^o =$
0.51×10^6 lbs.

Possible Shear Failure

The shear force across the cover is $F_2 = 0.51x10^6$ pounds.

By comparison to the shear force generated by the 30 foot corner drop, which was $1.85x10^6$ pounds, the shear resistance of the pipe buffer skirt is $\frac{3.24x10^6}{0.51x10^6}$ = 6.35 times strong enough to resist the available shear from this 40 inch punch drop.

Possible Cover Bolt Failure

The maximum loading on the furthest bolt from the impact corner would be:

Total g loading on cover (axially).

$$\frac{F_1}{W_{cask}} = \text{g loading}$$

$$\frac{1.3x10^6}{70,000} = 18.6 \text{ g}$$

Total load on cover = W_{cover} x g = 15,400 x 18.6 = 286,000 pounds.

By direct ratio with loading on the cover bolts from the 30 foot corner drop:

$$\frac{286,000}{1,030,000} = \frac{S}{114,500}; \quad S = 31,800 \text{ pounds}$$

Maximum loading on furthest bolt from the impact corner would be 31,800 pounds; the bolt ultimate strength is 225,000 pounds.

$$\text{Margin of safety} = \frac{225,000}{31,800} - 1 = \underline{6.1 \text{ M.S.}}$$

2.7.2.2 Puncture Drop Onto Top or Bottom Plate of Cask

Appendix 2.10.2 includes a description of a scale model test to determine the energy absorption capacity of the cask end plates when subjected to the punch test. Figure 1 of Appendix 2.10.2 shows the test arrangement, and Curves 1 and 2 plot the results.

Inspection of the problem shows that the punch will deform the bottom plate to a significant extent.

Assume initially that the total energy is absorbed by deflection of the top and bottom cask outside plates working together in accordance with Curve 2 of Appendix 2.10.2. Total available energy = $70{,}000 \times 40 = 2.8 \times 10^6$ in.-lbs.

With the cask dropping on the punch with the solid end down, the energy balance would be achieved by the 2-5/8 in. plate deflecting 1.9 inches and the 1-1/2 in. top plate deflecting 1.5 inches.

For the reverse end drop, the energy balance would be achieved by the 1-1/2 in. plate deflecting 2.2 inches and the 2-5/8 in. top plate deflecting 1.8 inches.

These deflections are all within the scale model test ranges and indicate there would be no rupture of the cask end plates.

2.7.2.2 (continued)

A check of the loading force on the 6 in. diameter puncture pin indicates the following:
Failure load (dynamic flow pressure for steel) = 50,000 psi.
Load for 6 in. diameter pin = $50,000 \times \pi \times 3^2$ = 1.4×10^6 lbs.

The deflections shown above for the cask end plates to satisfy the energy balance show a maximum loading force of 2.2×10^6 pounds which is in excess of the 1.4×10^6 pounds which the puncture pin can exert on the cask without failure of the pin. This indicates that there will be some failure of the puncture pin, plus deflection of the cask end plates, but to a lesser degree than first stated in the energy balance calculation.

A check on the strength of the closure cover bolts to react to the puncture test indicates the following:

Maximum loading is when cask is dropped onto its solid end and the closure top 1-1/2 in. cover is deflected 1.5 inches. This results from a loading of 437,000 pounds (see Table 3 of Appendix 2.10.2) on the cask cover.

There are (12) 1-1/2 in. closure bolts, each having a cross sectional area of 1.404 sq. in.

2.7.2.2 (continued)

The resulting tensile stress on each bolt would be = $\frac{437{,}000}{12 \times 1.404}$ = 26,000 psi, which is well below the tensile strength of the 160,000 psi closure bolts.

2.7.2.3 Puncture Drop Onto Cask Shell

From page 17 of ORNL-NSIC-68, the following formula is given to determine the minimum thickness required for the outside shell to withstand the 40 inch drop puncture test:

$$t = \left(\frac{W}{s}\right)^{0.71}$$

where: t = thickness of shell (in.)

W = weight of cask (lbs.)

s = ultimate tensile (psi)

$$t = \left(\frac{70{,}000}{71{,}200}\right)^{0.71} = 0.954 \text{ in.}$$

$$\text{M.S.} = \frac{1}{0.954} - 1 = \underline{0.05 \text{ M.S.}}$$

Check for excessive loss of lead under punch area. Assume total 40 in. drop energy is absorbed in lead deformation under punch area with a dynamic flow stress of lead = 5000 in.-lbs. cu. in.

$$\frac{70{,}000 \times 40}{5{,}000} = 560 \text{ cu. in. of lead deformation.}$$

Assume lead is deformed under a 12 in. radius area under the punch; the average depth of loss of lead under this area would be:

$$\frac{560}{\pi \times 12^2} = 1.24 \text{ in. of lead (3 in. loss permitted).}$$

This analysis is highly conservative, since no allowance was made for energy absorption by the steel shell and by deformation of the puncture pin.

$$\text{M.S.} = \frac{3}{1.24} - 1 = \underline{1.4 \text{ M.S.}}$$

2.7.3 Thermal (Accident Fire Conditions)

The thermal test follows the free drop and puncture tests and is reported in another section.

2.7.3.1 <u>Summary of Pressures and Temperatures</u>

Maximum Pressures (Accident Conditions):

Peak pressure inside cask = 16 psig.

Maximum Temperatures (Accident Conditions):

From inspection of the temperature curves for the cask under the fire accident conditions in Section 3.5, it shows the only steel plates reaching a temperature above the temperatures reached during the initial pouring of the cask shielding lead are the outer steel shell plates of the multiple steel plate outer cask shell.

The maximum temperature differential between the outer steel shell and the one under it is:

1425^{o} (outer shell) - 449^{o} (inner shell) =

$976^{o}F$.

The expansion of the lead shielding would not cause a problem. Axially the lead would lengthen, however, the outer shell would be at a higher temperature and eleongate more than the lead, resulting in an increase in clearance bwtween lead and steel. The inner steel shell would be cooler than the lead, and the outer steel shell would be hotter than the lead, therefore no radial interference problem would be encountered between the lead and the steel.

2.7.3.2 Differential Thermal Expansion (Accident Conditions)

α = Coefficient of linear thermal expansion 9.7×10^{-6} in./in./^{o}F

D = Outside diameter of cask shell = 50.5 in.

L = Length of cask shell = 133 in.

Radial thermal expansion of outer shell:

$R \times \alpha \times \Delta T$

$25.25 \times 9.7\times10^{-6} \times 976$ = <u>0.283 in. radial expansion</u>

Axial thermal expansion of outer shell:

$L \times \alpha \times \Delta T$

$133 \times 9.7\times10^{-6} \times 976$ = <u>1.25 in. axial expansion</u>

2.7.3.3 Stress Calculations (Accident Conditions)

The outer double steel shells of the cask are not tied to the inner cask shell except at the closure flange; therefore, the combined axial thermal expansion of the two outer shells will elongate at the solid end of the cask leaving a larger gap between the outer shell bottom and the lead.

Where the two outer cask shell plates are welded together at the top and bottom of the cask there will be differential thermal expansion in the axial direction. The outer shell will try to elongate 1.25 inches more than the inner shell. Since this will obviously create stresses beyond the yield stress limit in both shells, the shells will deform. The inner shell will yield in tension and the outer shell will yield and buckle under compression loading. This yielding will be orders of magnitude below the ultimate elongation of the material, and no rupture or tearing of the shell material will occur. Even if one of the two outer shells did rupture or tear, no unsafe condition would result. The lead and cask

2.7.3.3 (continued)

inner shell will never exceed the temperature conditions which were imposed on them during the lead pouring of the cask and its subsequent cooling cycle, therefore, no problems would be encountered for these inner sections of the cask.

2.7.3.4 Comparison with Allowable Stresses

Under the hypothetical accident conditions, the cask shell and lead shielding will be locally deformed and subjected to stresses beyond their ultimate stresses. However, they will not exceed their elongation properties and there will be no rupture, tearing, or breeching of the cask shell. The resulting accident conditions will fall within and satisfy the design criteria.

2.7.4 Water Immersion

Not applicable. The cask will not carry fissile material.

2.7.5 Summary of Damage (Accident Conditions)

The cask analyses show that the cask after the accident test sequence is damaged in localized areas, but fully complies with the requirements of 10 CFR 71.

The cask cover is shown to remain intact on the cask, however, the O-ring closure seal will probably leak gases due to localized deformation of the closure flange.

Localized impact areas of the outer shell of the cask together with the several impact limiters will be subjected to deformation and stresses beyond the yield strength of the material.

2.7.5 (continued)

There will be localized deformation of lead shielding resulting from the various drop sequences, however, there will be no loss of shielding in any area in excess of 3 inches, which is allowable under the 10 CFR 71 accident dose rate limits.

The fire test would not melt any shielding lead nor cause any unacceptable damage to the cask. The cask outer shell would buckle due to thermal expansion, but no breeching of the shell would result. The cask corners are additionally protected by the impact limiters, which would protect the corners from potential localized lead melting in the fire accident.

2.8 Special Form Material

Not applicable.

2.9 Fuel Rods

Not applicable.

2.10 Appendix

APPENDIX 2.10.1

TEST REPORT

SCALE MODEL TESTING OF SHORT PIPE SECTIONS FOR ENERGY ABSORPTION OF IMPACT LOADS

by

Raymond Eisenstadt
James Warden

August, 1975

SCALE MODEL TESTING OF SHORT PIPE SECTIONS FOR ENERGY ABSORPTION OF IMPACT LOADS

I. Background

The primary concern in the shipment of high level radioactive material is safety. To assure a high degree of safety, the Federal Government in the 10-CFR-Part 71 regulations has outlined a group of hypothetical accident conditions which a fuel shipping container must survive while maintaining adequate shielding, containment, and subcritical conditions. The regulations require such a shipping container withstand a hypothetical accident in which the container free falls 30 feet onto a hard, unyielding surface. To preclude excessive deformation or unacceptable high impact forces, the container is often equipped with a cushioning material or shock absorber buffers system. Although many types of impact limiters have been used in the past, many of them have serious design limitations.

II. Introduction

Tri Nuclear Corporation was engaged by Atcor, Inc. to assist them in the preparation of a Safety Analysis Report for one of their heavy lead shielded casks, which was to be used to transport irradiated nuclear fuel. It became apparent that to meet the design accident requirements for this service, the cask would require certain modifications to reduce the impact and shock loading resulting from the hypothetical accident drop conditions.

Due to certain design limitations of the existing cask, its mounting structure on the truck trailer, and its total weight (approaching the upper limit for over-the-road transport), it was necessary to carefully select a shock absorber system to afix to the cask during transport. Such a system should be light weight and require the least amount of modifications and retro-fitting of the existing equipment.

A feasible system was devised using a concept of short pipe sections welded onto a support plate, which would crush on impact with a predictable absorption of kinetic energy.

Theoretical calculational methods available for predicting the performance of such a system were not considered adequate, and it was decided to conduct a series of tests using scale models to obtain the necessary design and performance data.

III. Test Requirements

An experimental program was devised to determine the effects of pipe wall thickness, angle of impact to axis of pipe, and pipe end conditions (open or closed with a welded plate). Appropriate fixtures were made to hold the pipe test specimens under the compression test machine, which was a 60 Kip Baldwin Machine located at Union College, Schenectady, N.Y.

Each pipe speciment was loaded to fall failure, and load and deflection measurements were recorded and photographs taken at the completion of each test.

IV. Tests Conducted

The following tests were made and recorded data ploted as shown on the attached curves:

[1]Test No.	Pipe Wall	Crush Angle	End of Pipe
1	Thick	0	Open
2	Thick	0	Closed
4	Thick	45°	Open
5	Thick	45°	Closed
6	Thin	45°	Closed
7	Thin	45°	Open
8	Thin	$22\frac{1}{2}^\circ$	Closed
10	Thick	$22\frac{1}{2}^\circ$	Open
11	Thin	$22\frac{1}{2}^\circ$	Open
12	Thick	$22\frac{1}{2}^\circ$	Closed
13	Thin	0°	Open
14	Thin	0°	Closed
15	Thin	$22\frac{1}{2}^\circ$	Closed
17	Thick	45°	Open
18	Thin	$22\frac{1}{2}^\circ$	Open
19	Thin	$22\frac{1}{2}^\circ$	Closed

Notes:

1. Missing numbers were invalid tests.
2. Fig. A shows the test fixtures to mount the test speciments at $22\frac{1}{2}^\circ$ or 45° in the testing machine.
3. Fig. B is a sketch of the test specimens.
4. Fig. C shows a sketch of the testing machine.
5. Fig. D through G are pictures of the test specimens.

V. Test Results

1. Summary of Data Showing Effects of Variables

Pipe Wall	Crush Angle	End of Pipe	Energy Absorbed per cu.in.(in.lbs.)
Thin	0	Open	33,000
Thin	0	Closed	33,000
Thin	22½	Open	18,100
Thin	22½	Closed	19,600
Thin	45	Open	12,100
Thin	45	Closed	20,200
Thick	0	Open	37,000
Thick	0	Closed	37,000
Thick	22½	Open	23,300
Thick	22½	Closed	30,500
Thick	45	Open	10,700
Thick	45	Closed	19,100

Notes:

1. For 0° crush angles, the open and closed end pipes gave the same energy absorption, which was expected; however, the thick wall pipes gave a higher energy absorption per cu. in. of material than the thin wall pipes.
2. For the $22\frac{1}{2}^{\circ}$ crush angles, the closed end pipes are stronger than the open end ones, and the heavy wall pipes are stronger per cu. in. of material than the thin pipes.
3. For the 45° crush angles, the effect of the open and closed ends of the pipes was more pronounced, although the thin wall pipes showed a slightly higher specific strength per cu. in. than the thick wall pipes. This latter observation may be within the normal variations of the tests, but may indicate a slightly higher specific strength of the thin wall pipes for the sever angle of 45°.
4. Fig. H. through M. plot the scale model testdata for the various tests, plotting load in kips-vs-deflection in inches.
5. It is concluded that the best selection of pipe configuration is one with a closed end, heavy wall, and having a length about equal to its diameter.

V. Test Results - continued

2. Scale-up of Data

	Pipe Tested	Scale-up Pipe	Ratio
O.D. (in.)	2.375	6.625	2.8
wall (in.)	0.154	0.432	2.8
metal area (sq.in.)	1.08	8.41	7.8
pipe metal vol. (cu.in.)	2.56	55.7	21.8
pipe length (in.)	2.375	6.625	2.8

Notes:

1. Dimensions will scale-up directly by a factor of 2.8.
2. Loading will scale-up by a square factor (7.8)
3. Energy absorption will scale-up by a cube factor (21.8).
4. See Tables 1 through 4 for scale-up data.

3. Data Curves

Curves 1 through 4 plot the load in kips-vs-deflection in inches and the total absorbed energy in inch-pounds-vs-total deflection in inches for the full scale 6" sch. 80 pipes for 0 angle of impact and $22\frac{1}{2}^{\circ}$ angle of impact.

FIGURE WITHHELD UNDER 10 CFR 2.390

CARBON STEEL TEST SUPPORTS

MAKE ONE OF EACH TYPE

FIG. A

CARBON STEEL

TEST PIECES

FIGURE WITHHELD UNDER 10 CFR 2.390

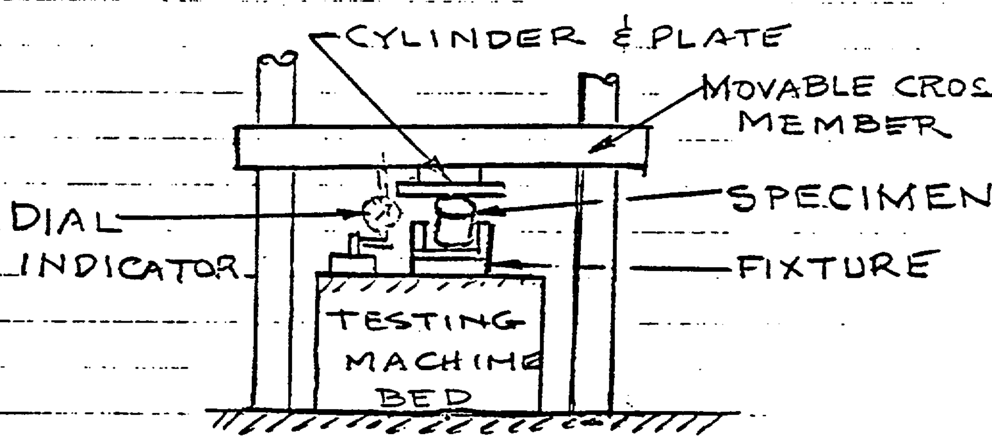
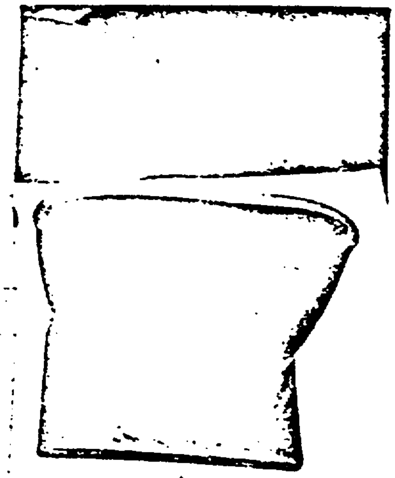

CYLINDER & PLATE
MOVABLE CROSS MEMBER
SPECIMEN
DIAL INDICATOR
FIXTURE
TESTING MACHINE BED

FIG. C

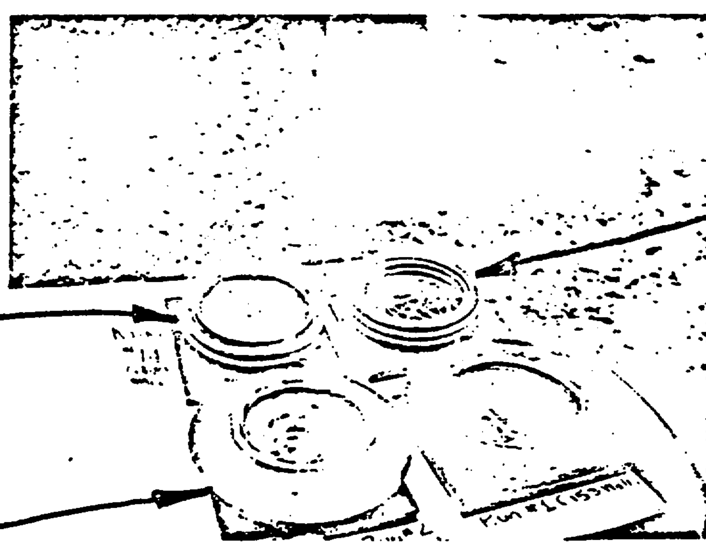
UNTESTED THIN WALL- CLOSED SPECIMEN L=D=2 3/16" THIN WALL

45° FIXTURE WITH RUN #17- OPEN SPECIMEN THICK WALL



Run #4

RUN #4 - 45° THICK WALL OPEN-SPECIMEN L=D=2 3/8" THICK WALL



RUN #14 THIN(.068") CLOSED END

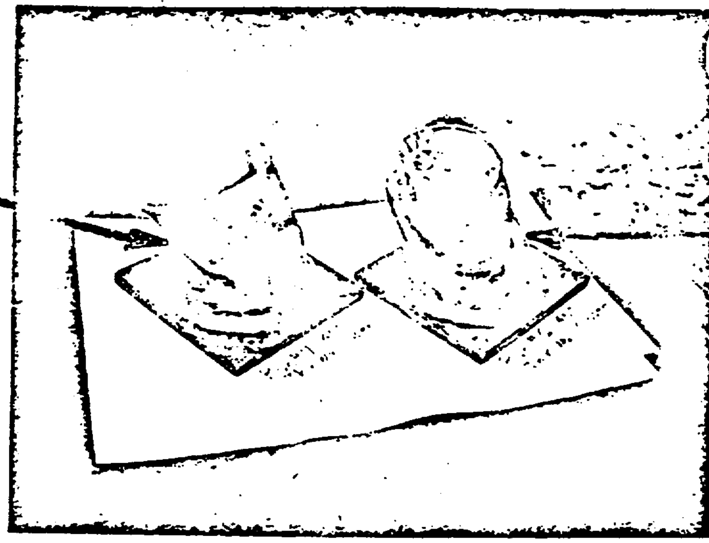
RUN #13 THIN(.068") OPEN END

RUN #1 THICK(.153) OPEN END

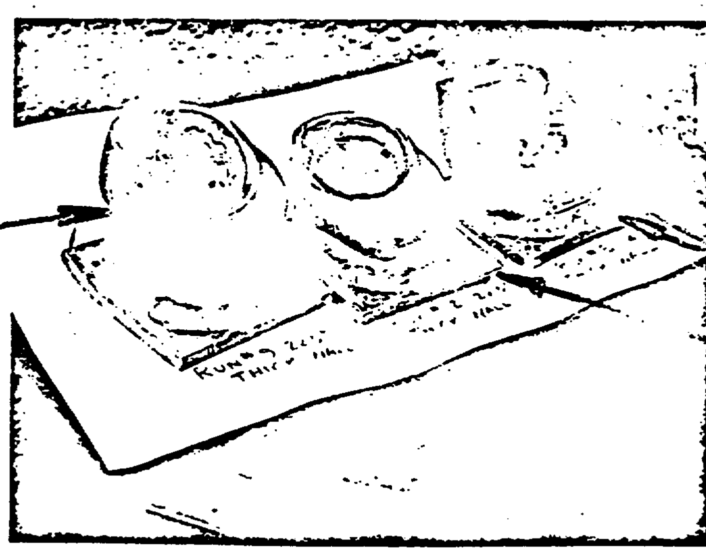
RUN #2 THICK(.153) CLOSED END

FLAT (0°) SPECIMENS

FIG. D



THIN WALL (.068") SPECIMENS
AT 45°



THICK WALL (.153) SPECIMENS
CLOSED ENDS

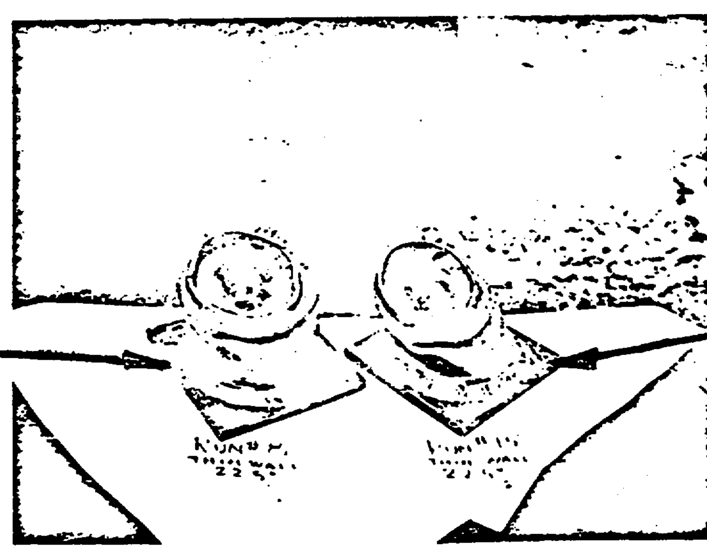
FIG. E

RUN #10
OPEN
END

RUN #19
CLOSED
END
(THIN WALL)

THICK WALL (.153) SPECIMENS
22.5°

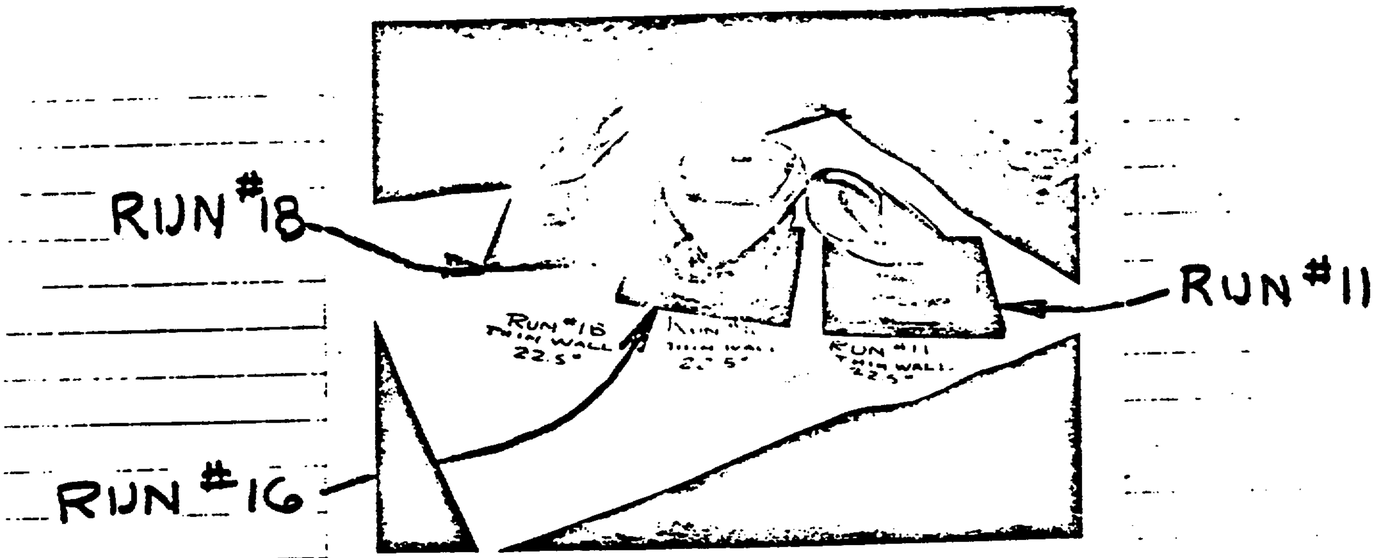
RUN #8



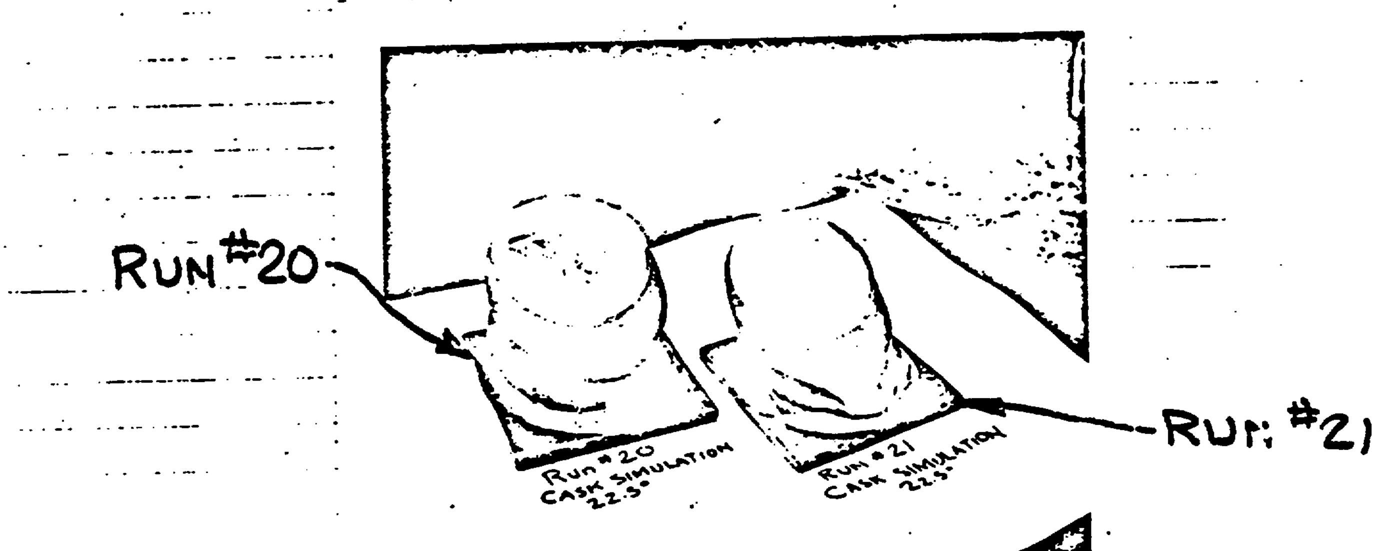
RUN #15

THIN WALL (.068") SPECIMENS
CLOSED ENDS - 22.5°

FIG. F

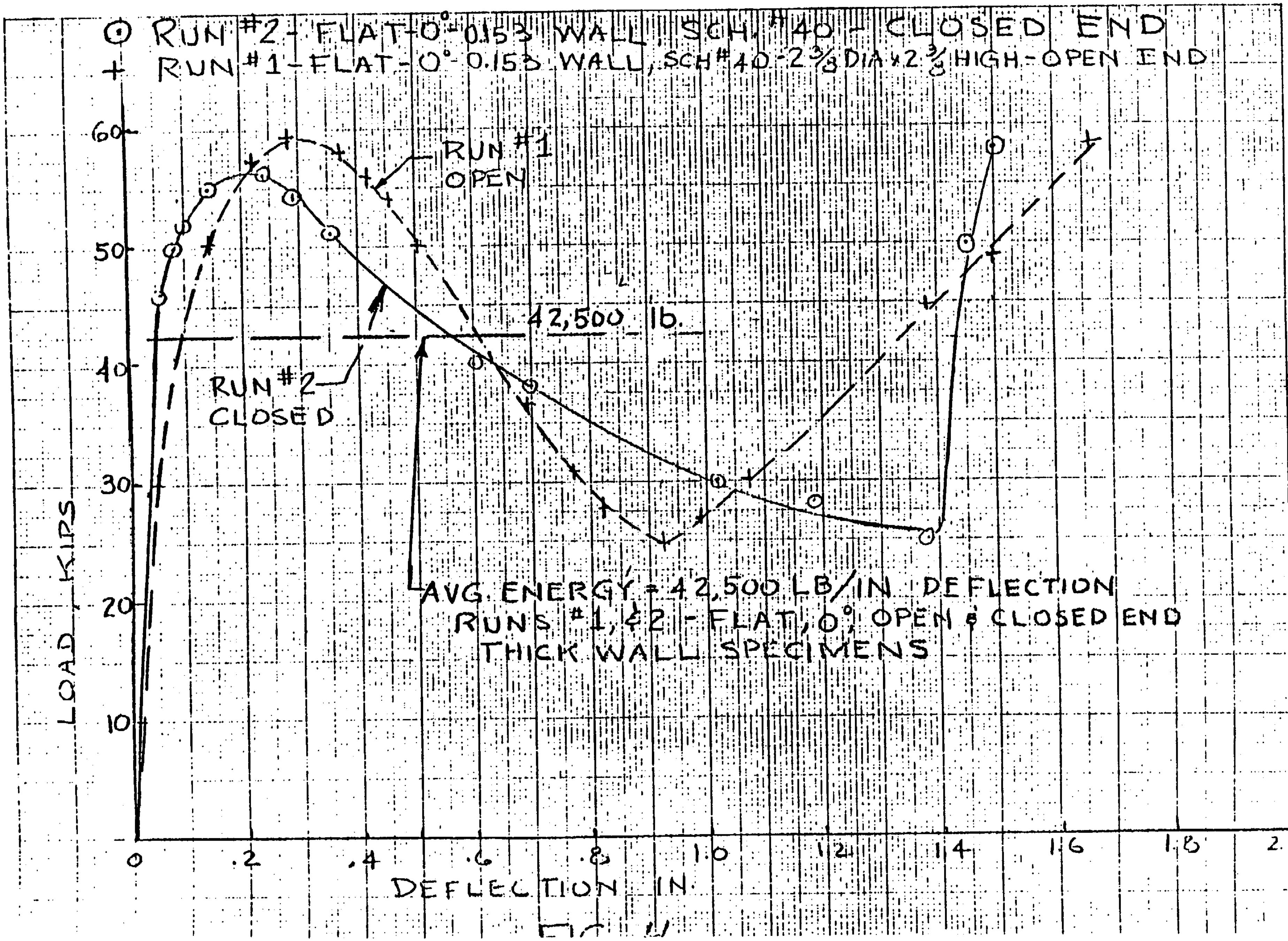


THIN WALL (.068") - OPEN END
SPECIMENS AT 22.5°



CASK SIMULATION SPECIMENS
CLOSED ENDS - 22.5°

FIG. G


⊙ RUN #2 - FLAT - 0° - 0.153 WALL, SCH. #40 - CLOSED END
+ RUN #1 - FLAT - 0° - 0.153 WALL, SCH #40 - 2 3/8 DIA × 2 3/8 HIGH - OPEN END
RUN #1
OPEN
42,500 lb.
RUN #2
CLOSED
AVG. ENERGY = 42,500 LB/IN DEFLECTION
RUNS #1, & 2 - FLAT, 0°, OPEN & CLOSED END
THICK WALL SPECIMENS
LOAD KIPS
10
20
30
40
50
60
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
1.8
2.0
DEFLECTION IN

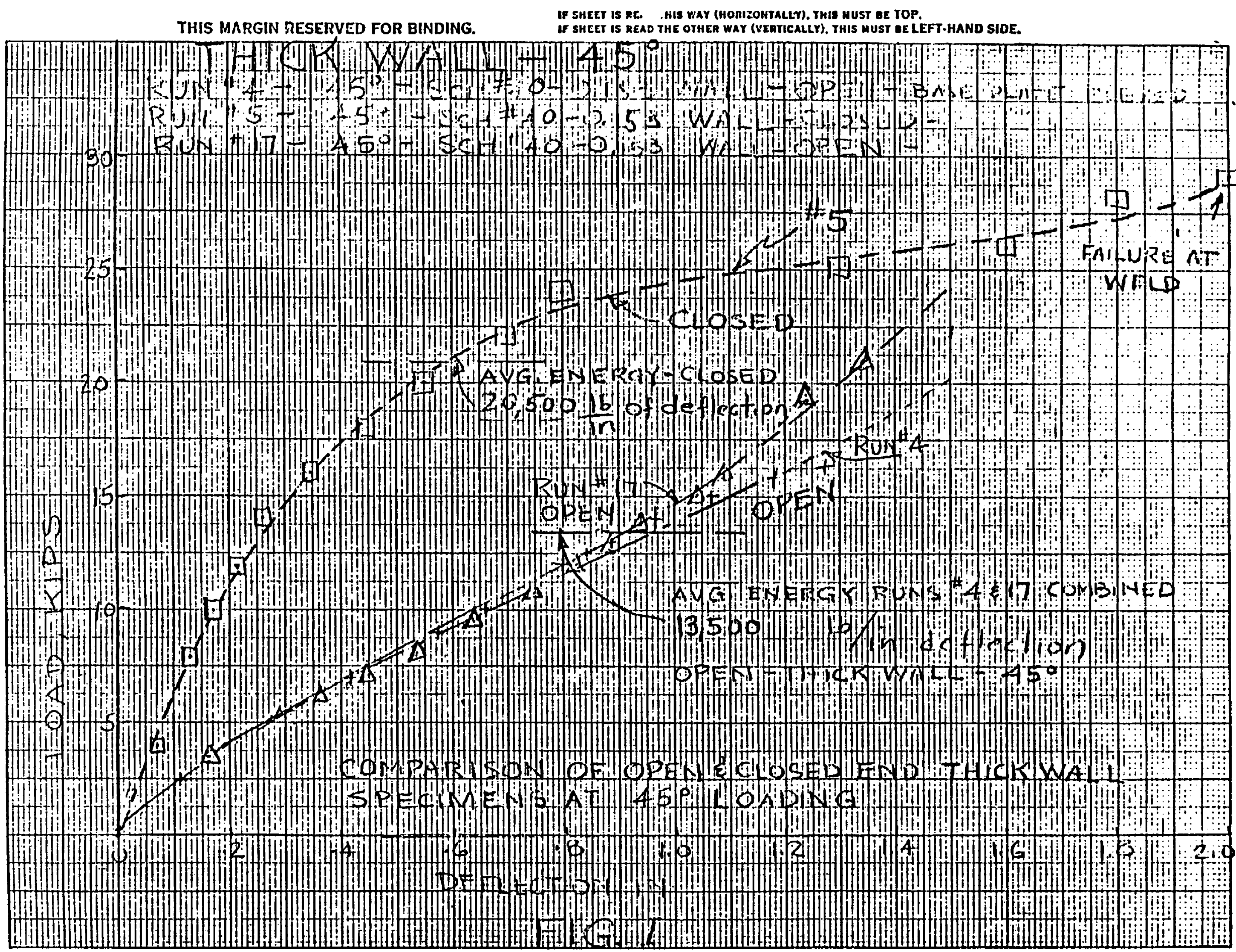
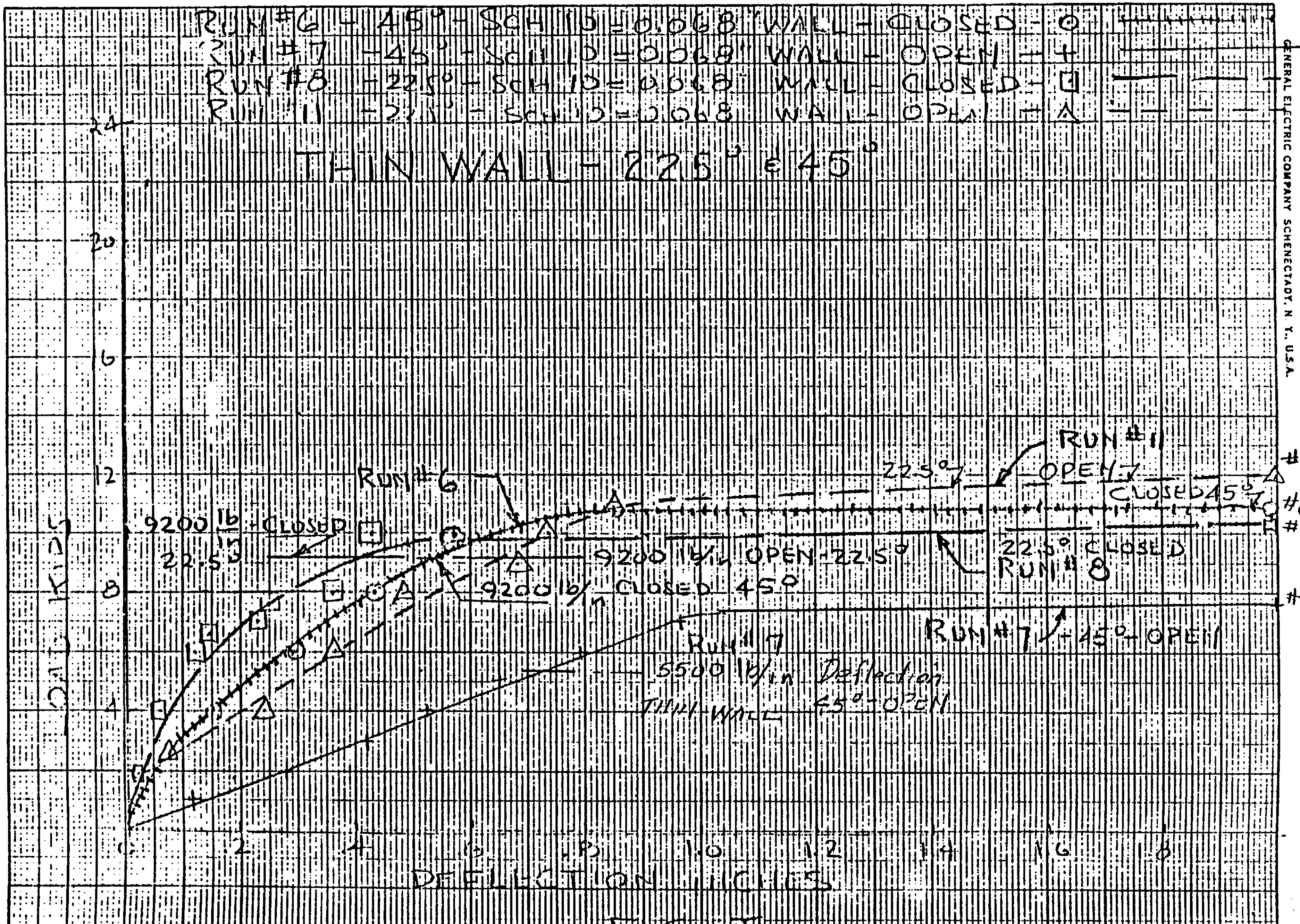

THIS MARGIN RESERVED FOR BINDING.
IF SHEET IS RE. HIS WAY (HORIZONTALLY), THIS MUST BE TOP.
IF SHEET IS READ THE OTHER WAY (VERTICALLY), THIS MUST BE LEFT-HAND SIDE.
THICK WALL - 45°
RUN #4 - 45°
RUN #5 - 45° - SCH #40 - 0.53 WALL
RUN #17 - 45° - SCH 40 - 0.53 WALL - OPEN
LOAD, KIPS
DEFLECTION, IN
#5
CLOSED
FAILURE AT WELD
AVG. ENERGY - CLOSED
26,500 lb/in of deflection
RUN #4
RUN #17 OPEN
OPEN
AVG ENERGY RUNS #4 & 17 COMBINED
13,500 lb/in deflection
OPEN - THICK WALL - 45°
COMPARISON OF OPEN & CLOSED END THICK WALL
SPECIMENS AT 45° LOADING
GENERAL ELECTRIC COMPANY SCHENECTADY, N. Y., U.S.A.
29/64 Inch Divisions

FIG. 1

ST-82

THIS MARGIN RESERVED FOR BINDING.

IF SHEET IS READ THIS WAY (HORIZONTALLY), THIS MUST BE TOP.
IF SHEET IS READ THE OTHER WAY (VERTICALLY), THIS MUST BE LEFT-HAND SIDE.



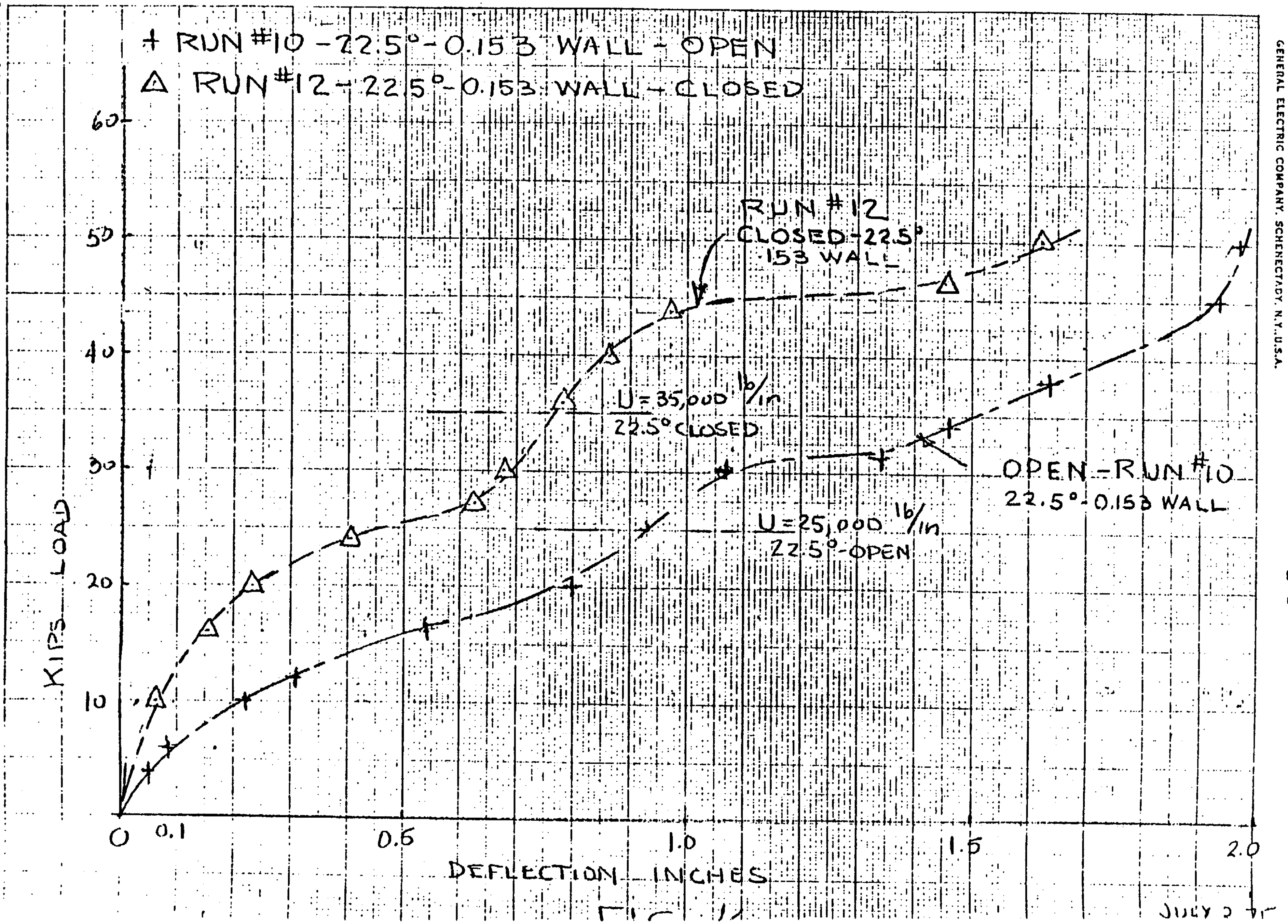
GENERAL ELECTRIC COMPANY SCHENECTADY, N. Y. U.S.A.

20/64 inch Divisions

2-96

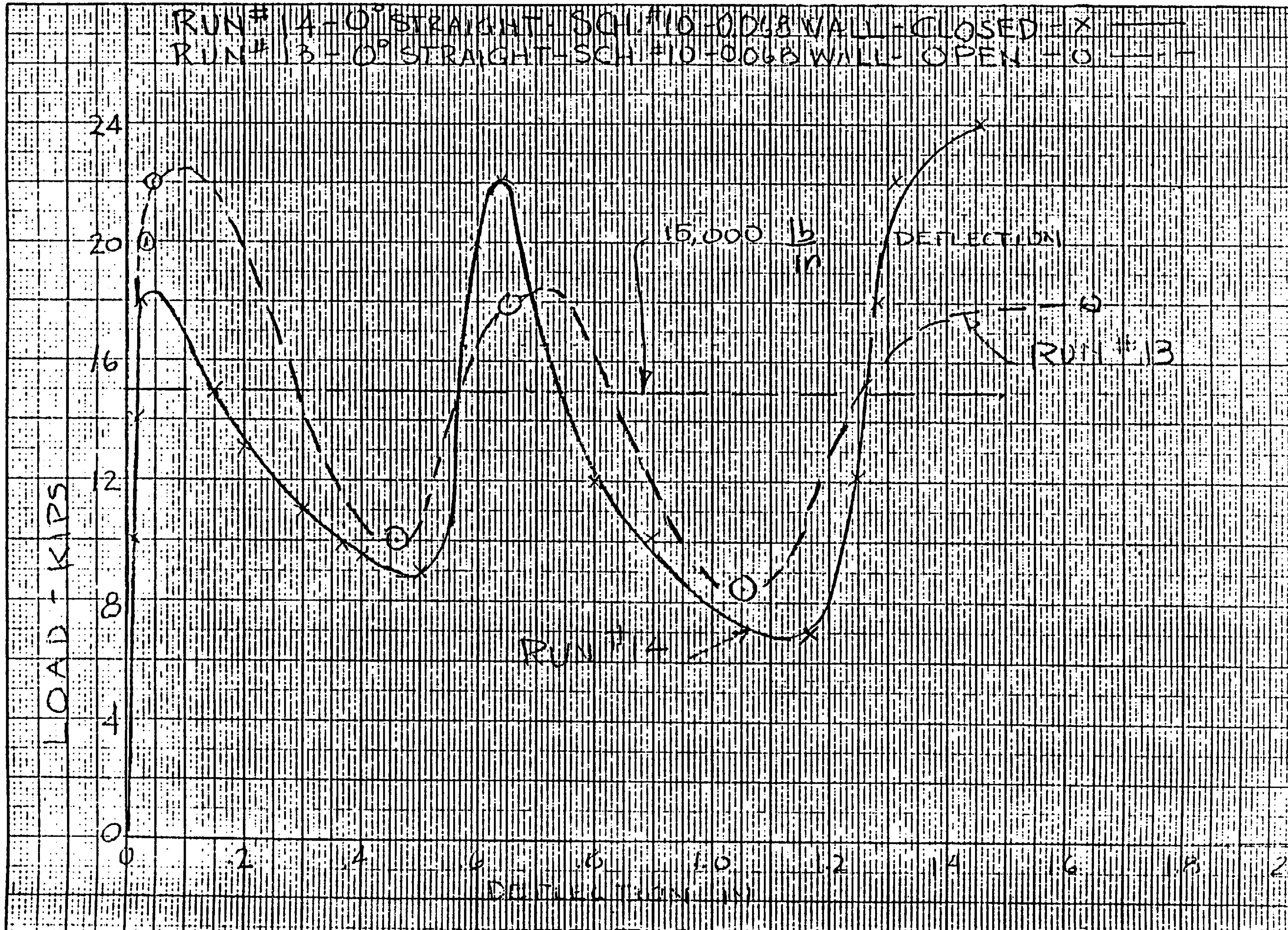
THIS MARGIN RESERVED FOR BINDING.

IF SHEET READ THIS WAY (HORIZONTALLY) THIS MUST BE TOP
IF SHEET READ THE OTHER WAY (VERTICALLY) THIS MUST BE LEFT-HAND SIDE.



THIS MARGIN RESERVED FOR BINDING.

IF SHEET IS REA.. .HIS WAY (HORIZONTALLY), THIS MUST BE TOP.
IF SHEET IS READ THE OTHER WAY (VERTICALLY), THIS MUST BE LEFT-HAND SIDE.



GENERAL ELECTRIC COMPANY SCHENECTADY, N. Y., U.S.A.

28/64 Inch Divisions

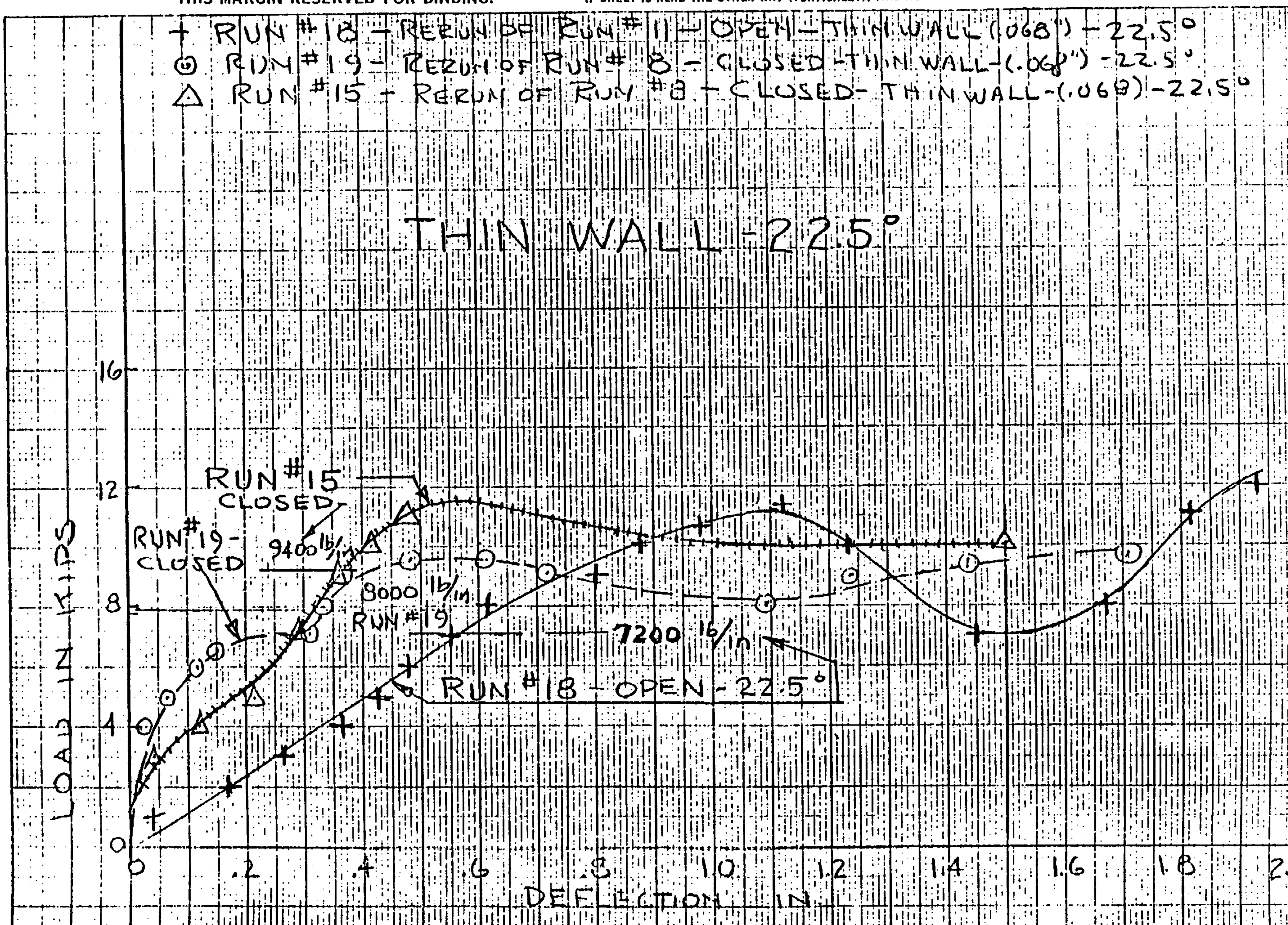

+ RUN #18 - RERUN OF RUN #11 - OPEN - THIN WALL (.068") - 22.5°
⊙ RUN #19 - RERUN OF RUN #8 - CLOSED - THIN WALL - (.068") - 22.5°
△ RUN #15 - RERUN OF RUN #8 - CLOSED - THIN WALL - (.068) - 22.5°
THIN WALL - 22.5°
RUN #15 CLOSED
RUN #19 - CLOSED
9400 lb/in
8000 lb/in
RUN #19
7200 lb/in
RUN #18 - OPEN - 22.5°
LOAD IN KIPS
0
4
8
12
16
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
1.8
2.0
DEFLECTION IN.

FIG. M

TABLE 1

Force - vs - Deflection for 6" Sch. 80 Carbon Steel Pipe
Angle of Load to Axis of Pipe = 0°

From Scale Model Test		* Scale-Up to 6" Pipe	
Load(kips)	Deflection(in.)	Load(kips)	Deflection(in)
46	0.05	469	0.14
54	.15	551	.42
56	.25	571	.70
52	.35	530	.98
47	.45	479	1.26
43	.55	439	1.54
40	.65	408	1.82
37	.75	377	2.10
33	.85	337	2.38
31	.95	316	2.66
29	1.05	296	2.94
28	1.15	286	3.22
27	1.25	275	3.50
26	1.35	265	3.78
42	1.45	428	4.06

* 1. Scale up factor for deflection is 2.8.
2. Scale up factor for load is 2.8^2 times a dynamic stress factor for steel of $1.3 = 2.8^2 \times 1.3 = 10.2$.

TABLE 2

Total Absorbed Energy (in.lbs.) - vs - Total Deflection (in.)
6" Sch. 80 Carbon Steel Pipe - Angle of Load to Axis of Pipe = 0°

Total Absorbed Energy (in.lbs.) x 10^6	Total Deflection (in.)
0.13	0.14
.28	.42
.45	.70
.59	.98
.73	1.26
.85	1.54
.96	1.82
1.07	2.10
1.16	2.38
1.25	2.66
1.33	2.94
1.41	3.22
1.49	3.50
1.57	3.78
1.69	4.06

TABLE 3

Force vs. Deflection for 6-inch, Schedule 80 Carbon Steel Pipe
Angle of Load to Axis of Pipe = 22-1/2°

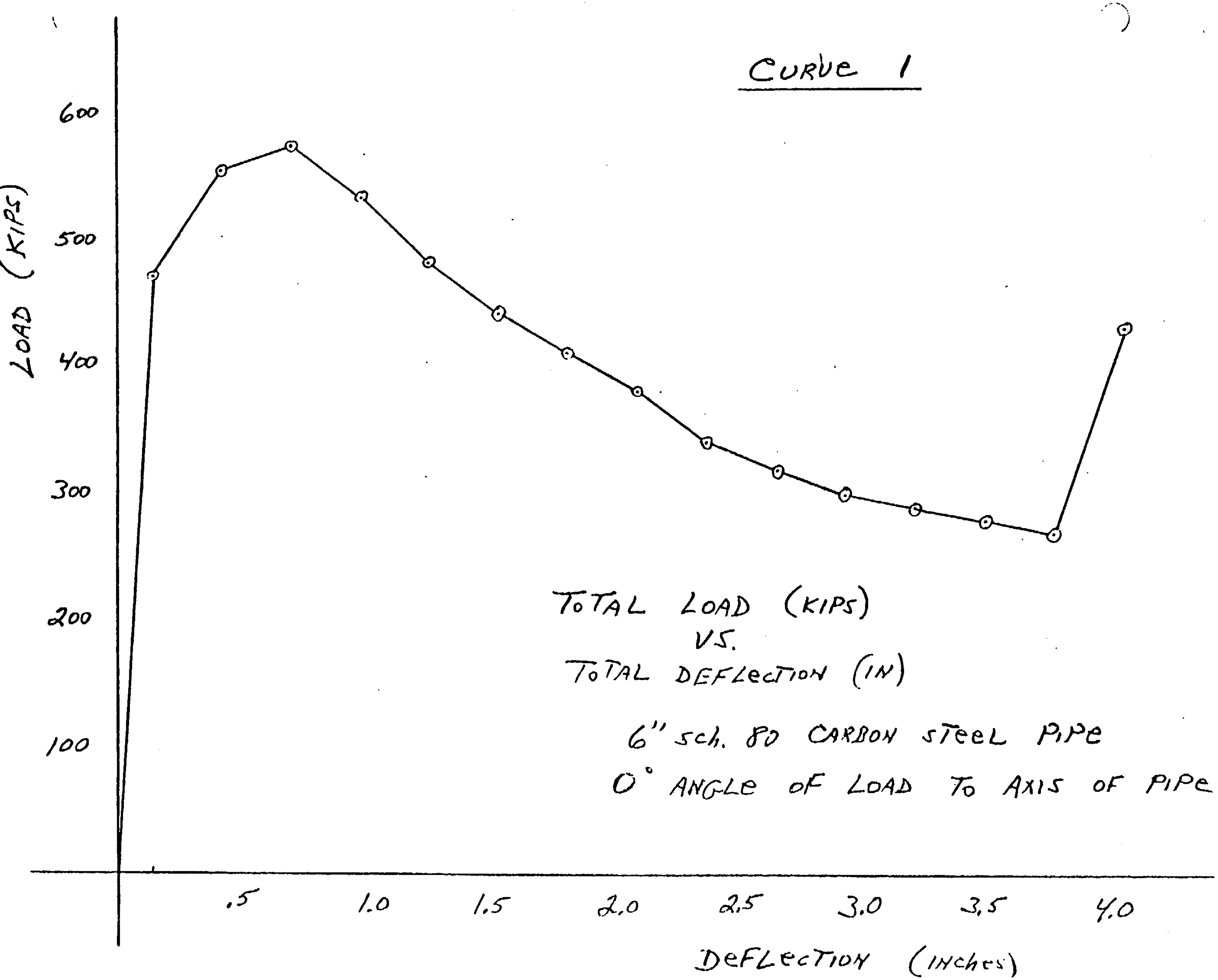
From Scale Model Tests		* Scale-Up to 6-inch Pipe	
Load (kips)	Deflection (inches)	Load (kips)	Deflection (inches)
7	0.05	71	0.14
17	.15	173	.42
21	.25	214	.70
23	.35	235	.98
25	.45	255	1.26
26	.55	265	1.54
28	.65	286	1.82
34	.75	347	2.10
40	.85	408	2.38
43	.95	439	2.66
44	1.05	449	2.94
45	1.15	459	3.22
46	1.25	469	3.50
46	1.35	469	3.78
47	1.45	479	4.06
48	1.55	490	4.34
50	1.65	510	4.62
53	1.75	541	4.90

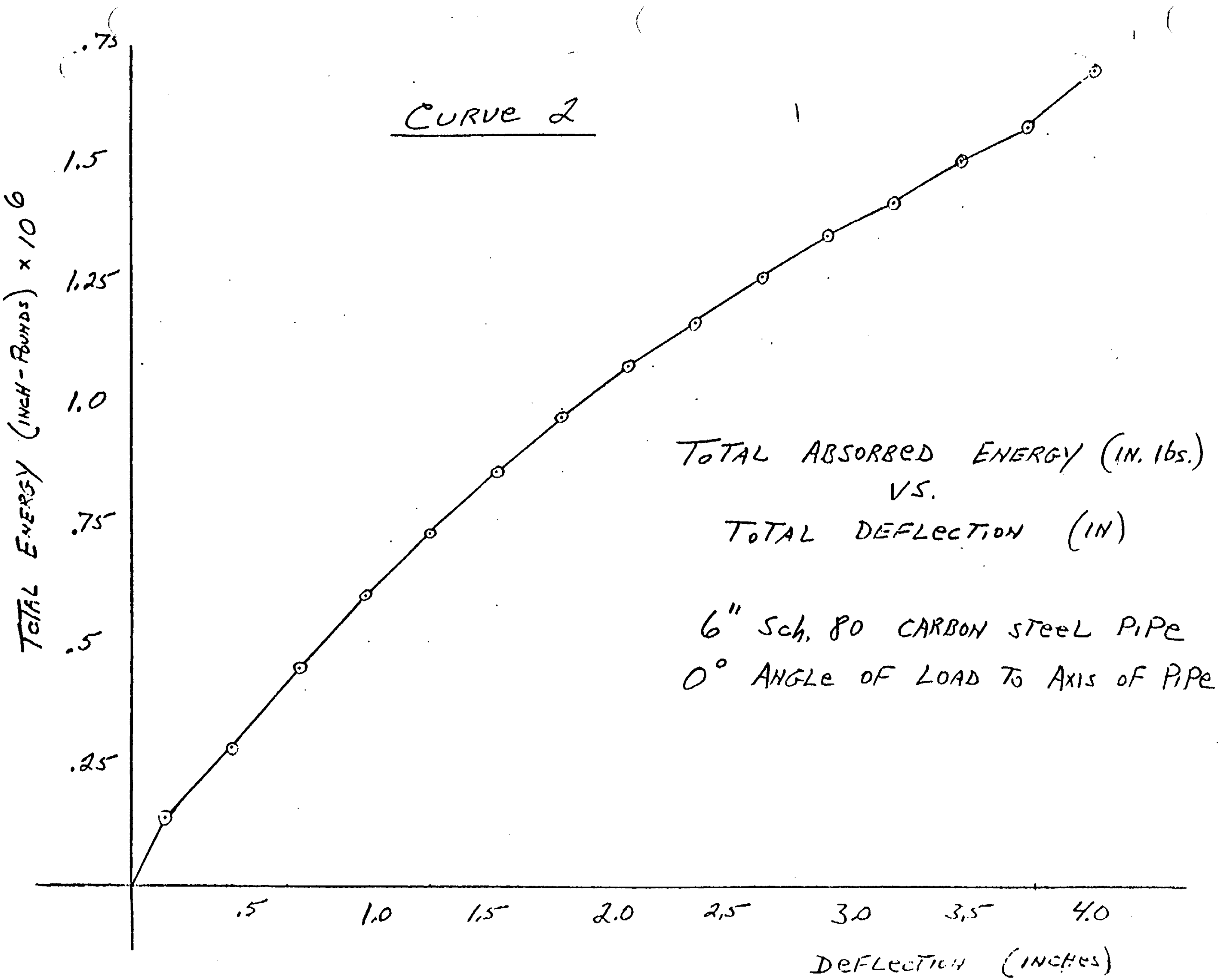
*Scale-up factor for deflection is 2.8.
Scale-up factor for load is 2.8^2 times a dynamic stress factor for steel of $1.3 = 2.8^2 \times 1.3 = 10.2$.

<u>TABLE 4</u>

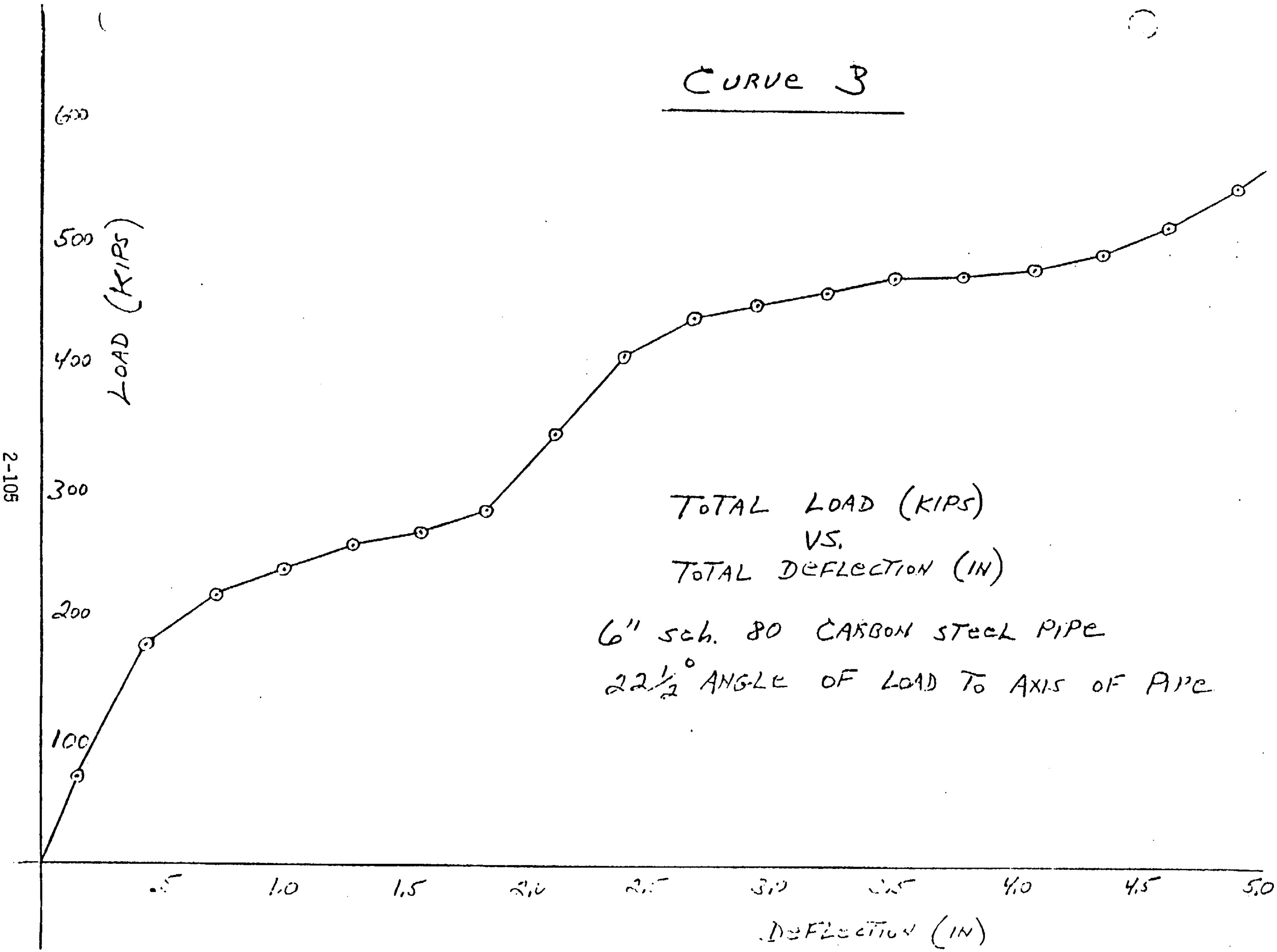
Total Absorbed Energy (in./lbs.) vs. Total Deflection (in.)
6-inch Schedule 80 Carbon Steel 1/2 Pipe
Angle of Load to Axis of Pipe = 22-1/2°

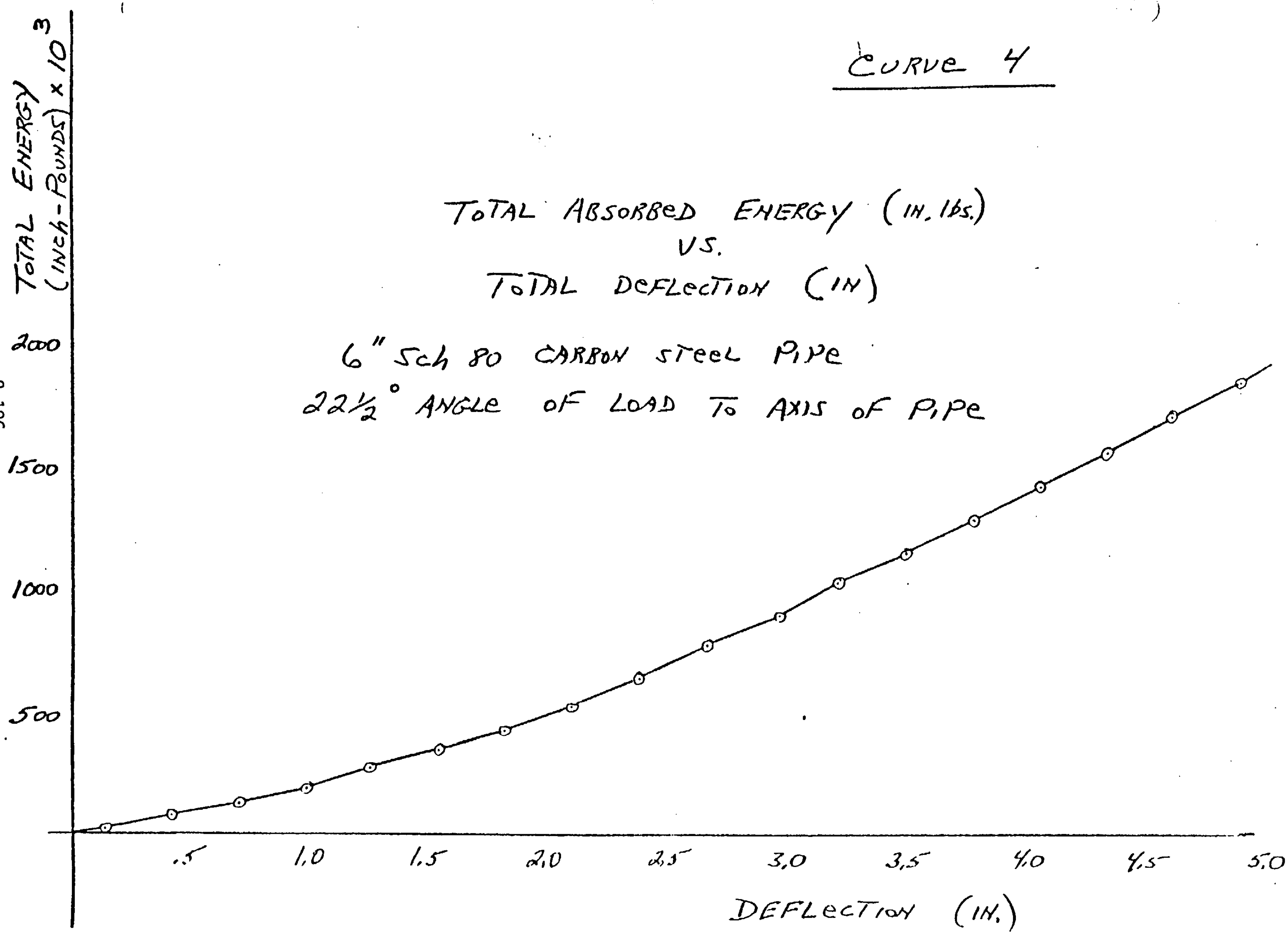
Total Absorbed Energy (in./lbs.) x 10^6	Total Deflection (in.)
0.02	0.14
.07	.42
.13	.70
.19	.98
.27	1.26
.34	1.54
.42	1.82
.52	2.10
.63	2.38
.75	2.66
.88	2.94
1.01	3.22
1.14	3.50
1.27	3.78
1.40	4.06
1.54	4.34
1.68	4.62
1.84	4.90


CURVE 1
TOTAL LOAD (KIPS)
VS.
TOTAL DEFLECTION (IN)
6" sch. 80 CARBON STEEL PIPE
0° ANGLE OF LOAD TO AXIS OF PIPE
LOAD (KIPS)
600
500
400
300
200
100
.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
DEFLECTION (INCHES)


CURVE 2
TOTAL ABSORBED ENERGY (IN. lbs.)
VS.
TOTAL DEFLECTION (IN)
6" Sch. 80 CARBON STEEL PIPE
0° ANGLE OF LOAD TO AXIS OF PIPE
TOTAL ENERGY (INCH-POUNDS) x 10^6
.25
.5
.75
1.0
1.25
1.5
1.75
.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
DEFLECTION (INCHES)

2-104


CURVE 3
TOTAL LOAD (KIPS)
VS.
TOTAL DEFLECTION (IN)
6" sch. 80 CARBON STEEL PIPE
22½° ANGLE OF LOAD TO AXIS OF PIPE
LOAD (KIPS)
100
200
300
400
500
600
DEFLECTION (IN)
1.0
1.5
4.0
4.5
5.0


CURVE 4
TOTAL ABSORBED ENERGY (IN. lbs.)
VS.
TOTAL DEFLECTION (IN)
6" Sch 80 CARBON STEEL PIPE
22½° ANGLE OF LOAD TO AXIS OF PIPE
TOTAL ENERGY (INCH-POUNDS) x 10³
2000
1500
1000
500
.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
4.5
5.0
DEFLECTION (IN.)

APPENDIX 2.10.2

TEST DATA

SCALE MODEL TESTING OF CASK END PLATES FOR DEFLECTION AND ENERGY ABSORPTION

by

James Warden

August, 1975

FIGURE WITHHELD UNDER 10 CFR 2.390

FIG. 1

Fig. 2

FIGURE WITHHELD UNDER 10 CFR 2.390

TABLE 1

Force - vs - Deflection for Bottom Plate of Solid End
2.625" Plate With 37" Span

From Scale Model Tests		* Scale-Up To Cask	
Load(kips)	Deflection(in.)	Load(kips)	Deflection(in.)
16	0.05	752	0.3
22	.10	1030	.6
25.5	.15	1200	.9
28.5	.20	1340	1.2
32	.25	1500	1.5
35	.30	1650	1.8
39	.35	1830	2.1
43	.40	2020	2.4
47	.45	2210	2.7
51	.50	2400	3.0

* 1. Scale up factor for deflection is 6.
2. Scale up factor for load is 6^2 times a dynamic stress factor for steel of $1.3 = 6^2 \times 1.3 = 47$.

TABLE 2

Total Absorbed Energy (in.lbs.) - vs - Total Deflection (in.)
2.625" Plate with 37" Span

Total Absorbed Energy (in.lbs.) x 10^6	Total Deflection (in.)
0.23	0.3
.54	.6
.90	.9
1.30	1.2
1.75	1.5
2.25	1.8
2.80	2.1
3.40	2.4
4.06	2.7
4.78	3.0

TABLE 3

Force - vs - Deflection for Bottom Plate of Closure End
1.5" Plate with 37" Span

From Scale Model Test		* Scale-Up to Cask	
Load(kips)	Deflection(in.)	Load(kips)	Deflection(in.)
3.2	0.05	150	0.3
6.0	.10	282	.6
7.1	.15	334	.9
8.1	.20	381	1.2
9.3	.25	437	1.5
10.4	.30	489	1.8
11.8	.35	555	2.1
13.3	.40	625	2.4
14.9	.45	700	2.7
16.5	.50	776	3.0

* 1. Scale-up factor for deflection is 6.
2. Scale-up factor for load is 6^2 times a dynamic stress factor for steel of 1.3 = 6^2 x 1.3 = 47.

TABLE 4

Total Absorbed Energy (in.lbs.) - vs - Total Deflection (in.)
1.5" Plate with 37" Span

Total Absorbed Energy (in.lbs.) x 10^6	Total Deflection (in.)
0.05	0.3
.13	.6
.23	.9
.35	1.2
.48	1.5
.63	1.8
.80	2.1
.99	2.4
1.20	2.7
1.43	3.0

TABLE 5

Force - vs - Deflection for Puncture Test Solid End
2.625 Plate with $50\frac{1}{2}$" Span

From Scale Model Test		* Scale-Up To Cask	
Load(kips)	Deflection(in.)	Load(kips)	Deflection(in.)
8.5	0.05	400	0.3
12.5	.10	588	.6
14.5	.15	682	.9
15.5	.20	729	1.2
16.8	.25	790	1.5
17.8	.30	837	1.8
18.9	.35	888	2.1
20.0	.40	940	2.4
21.2	.45	996	2.7
22.4	.50	1050	3.0

* 1. Scale-up factor for deflection is 6.
2. Scale-up factor for load is 6^2 times a dynamic stress factor for steel of 1.3 = $6^2 \times 1.3 = 47$.

TABLE 6

Total Absorbed Energy (in.lbs.) - vs - Total Deflection (in.)
2.625 Plate with $50\frac{1}{2}$" Span

Total Absorbed Energy (in.lbs.) x 10^6	Total Deflection (in.)
0.12	0.3
.30	.6
.50	.9
.72	1.2
.96	1.5
1.21	1.8
1.48	2.1
1.76	2.4
2.06	2.7
2.38	3.0

TABLE 7

Force - vs - Deflection for Puncture Test Closure End
1.5" Plate with 50½" Span

From Scale Model Test		* Scale-Up To Cask	
Load(kips)	Deflection(in.)	Load(kips)	Deflection(in)
1.4	0.05	66	0.3
2.4	.10	113	.6
3.2	.15	150	.9
3.7	.20	174	1.2
4.4	.25	207	1.5
5.2	.30	244	1.8
6.3	.35	296	2.1
7.4	.40	348	2.4
8.3	.45	390	2.7
9.4	.50	442	3.0

* 1. Scale-up factor for deflection is 6.
2. Scale-up factor for load is 6^2 times a dynamic stress factor for steel of $1.3 = 6^2 \times 1.3 = 47$.

TABLE 8

Total Absorbed Energy (in. lbs.) - vs - Total Deflection (in)
1.5" Plate with 50½" Span

Total Absorbed Energy (in.lbs.) x 10^6	Total Deflection (in.)
0.02	0.3
.05	.6
.10	.9
.15	1.2
.21	1.5
.28	1.8
.37	2.1
.47	2.4
.59	2.7
.72	3.0

3.0 THERMAL EVALUATION

This section examines the thermal engineering design features of the CNS-3-55 Cask that are important to safety and compliance with the performance requirements of 10 CFR 71.

3.1 DISCUSSION

The CNS-3-55 Cask package thermal system consists of three basic areas which provide thermal protection. these are the cylindrical shell (sides), the closed end of the cask, and the closure lid of the cask.

The cylindrical shell has four basic layers. The outer layer consists of 0.50-inch stainless steel; this encloses a 0.50-inch inner cylindrical shell of carbon steel. Following this, a 6-inch thickness of lead provides shielding; finally, the inner cavity liner is formed by 0.25-inch carbon steel with stainless steel cladding on the cavity side. A 1/32-inch air gap is assumed to exist between the outer stainless steel layer and the inner carbon steel layer.

The exterior surface may be optionally painted with a nuclear grade epoxy coating. The thickness of this coating (about 15 mils minimum) and the large thermal conductivity (620 $BTU/hr/ft^2/°F$) serve to limit the affect of the coating on cask temperatures. No difference in predicted cask temperatures occurs for a painted cask surface.

The closed end of the cask consists of an outer shell of 1.50-inch stainless steel plate, covering an inner circular carbon steel plate 0.62-inch thick; this, in turn, covers a 0.50 inch thickness of lead; finally, the inner cavity liner is formed by a 0.50-inch carbon steel plate with stainless steel cladding. A 1/32-inch air gap exists between the 0.62-inch and 0.50-inch carbon steel plates; and also between the 0.62-inch carbon steel plate and the 1.50 inch thick stainless steel plate.

The closure lid of the cask is made up of an outer shell plate of 0.38-inch stainless steel plate, covering a 1.25-inch steel plate. A 6.0-inch thickness of lead is contained between this 1.25-inch plate and the inner cavity liner, which consists of 0.62-inch thick carbon steel, with stainless steel cladding. A 1/32-inch air gap exists between the outer 0.38-inch stainless steel plate, and the 1.25-inch carbon steel plate.

(continued on page 3-2)

A representation of cask layered construction may be found, as Figures 3.1.1, 3.1.2, and 3.1.3.

Thermal transfer mechanisms for the CNS-3-55 Cask are as follows:

3.1.1 <u>Normal Transport Conditions</u>

A. Decay heat of the payload is transferred to the inner liner surface by radiant heat transfer.

B. This heat is then transferred by thermal conductance through the lead shield.

C. From the lead shield, the heat is transferred by conductance to the outer shell steel plate. When an air gap is encountered, heat transfer takes place across the gap by radiation, and also by conduction through air.

D. The outer shell plates then radiate the heat to the surroundings.

FIG. 3.1.1: REGION 1 - CY[illegible] [illegible]

FIGURE WITHHELD UNDER 10 CFR 2.390

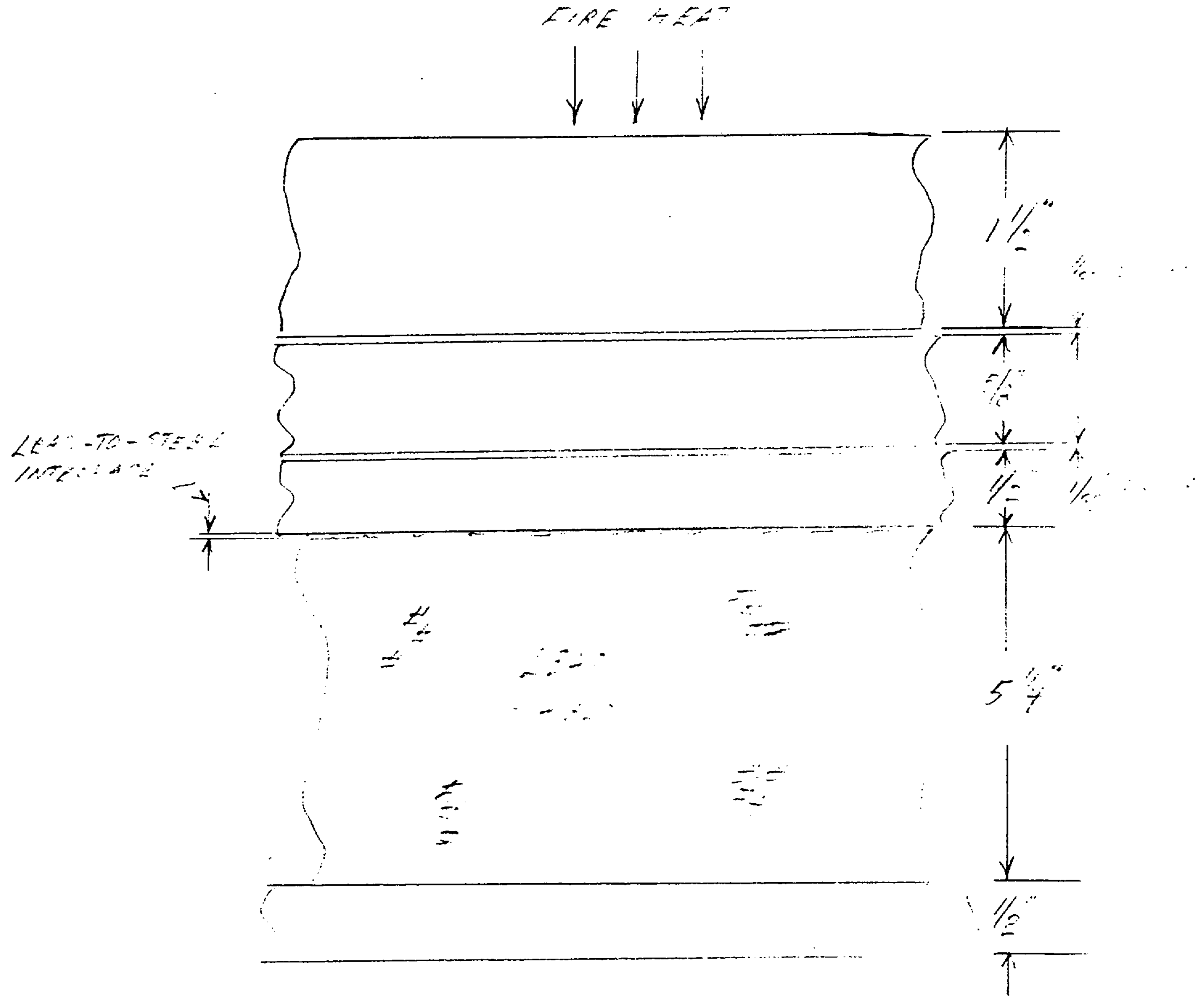


FIG. 3.1.5:

3.1.2 Hypothetical Thermal Accident Conditions

A. Heat from the hypothetical fire is transferred to the outer steel shell plates by radiation.

B. The outer steel shell plates transfer the heat from the fire to the inner steel shell plates by conductance. When an air gap is encountered, heat transfer is by radiation, and conduction.

C. The inner steel shell plates conduct heat into the lead shield.

D. The lead shield conducts the heat to the inner steel liner shell.

E. The steel shell plates and the lead shielding will store heat during the period of the fire, and subsequent to the fire, will dissipate this heat by heat transfer mechanisms as detailed under 3.1.1.

F. During the fire, the impact limiter skirt plate covering the closure end of the cask also provides additional protection for the closure O-ring seals, due to its shielding effect, and the long conduction path through air. (See Figure 10, Section 2, page 2-56.)

Important maximum temperatures of the CNS-3-55 cask package for both Normal Transport Conditions, and Hypothetical Thermal Accident Conditions are summarized below.

o Normal Transport Conditions

Temperatures, ^{o}F *

Ambient Outside Air	Inner Cavity	Lead Shield	First O-ring	Second O-ring	Outside Surface
$100^{o}F$	130.2	130.1	129.3	129.2	129.4
$130^{o}F$	161.2	161.0	161.1	160.0	160.3

o Hypothetical Thermal Accident Conditions

Temperatures, ^{o}F, Maximum *

Inner Cavity	Lead Shield	First O-ring	Second O-ring
449	492	461	474

* Includes heating due to solar insolation.

The important conclusions derived from the above results include:

1. The lead shield does not melt under hypothetical Thermal Accident Conditions (maximum lead temperature = 492^{o}F, at outside of lead shield (hottest); $T_{lead\ melt} \doteq$ 621^{o}F).

2. The components of the closure system (O-ring seals) are exposed to temperatures not in excess of 474^{o}F. Silicone seals retain excellent sealing properties to 500^{o}F. The maximum predicted temperature of the pressurized containment cavity is 449^{o}F. This temperature is used to derive maximum containment chamber pressures. A discussion of silicone seal performance as related to cask thermal performance may be found in Section 3.5.6.1 below.

The package has been evaluated* for a payload decay heat of 250 watts per hour. Steady state, and transient thermal analyses have been performed for this decay heat value, for ambient air temperatures of 100^{o}F, and 130^{o}F; and also for the conditions specified by the hypothetical thermal accident.

3.2 Summary of Thermal Properties of Materials

Thermal material properties used for analysis were taken from several sources. These include:

1. Handbook of Heat Transfer, Rohsenow and Hartnett, McGraw-Hill, 1973.
2. ASME Boiler and Pressure Vessel Code, Section III, Appendices, 1980.
3. Cask Designer's Guide, Nuclear Safety Information Center, Oak Ridge National Laboratory.
4. Mark's Standard Handbook for Mechanical Engineers, McGraw-Hill, 1980.

For each material which represents a thermal regime within the cask (stainless steel, carbon steel, lead, air) relevent material properties were selected. For metallic properties, density was assumed constant; density of air was not needed for modelling. However, thermal properties such as specific heat and conductivity, (which can vary significantly with temperature) were represented by tables which showed how the particular property varied over the range of temperatures encountered. This helped in producing a more accurate modelling of the cask thermal system.

Surface emissivities and absorptivities, used in radiant heat transfer applications, were taken from a variety of the above sources, and are as follows:

	EMISSIVITES	ABSORBTIVITES
Stainless Steel	.15 (Ref. 4)	.40 (Ref. 1)
Carbon Steel	.85 (Ref. 3)	.60 (Ref. 3)

3.3 Technical Specifications of Components

3.3.1 The drain plug relief valve is factory set at 25 psi $\pm$ 5%. Temperature range is -100^{0}F to $+450^{0}$F. (See Appendix 3.6.3 for specifications and details).

3.4 Thermal Evaluation for Normal Conditions of Transport

3.4.1 Thermal Model

A steady state thermal analysis was performed, in order to evaluate the CNS-3-55 Cask system performance during normal conditions of transport. Specific conditions and assumptions were as follows:

1. 250 watts/hour internal decay heat.
2. 300 Btu/hr.-ft.^{2} solar load. (Reference 3, page 130.) It was the severest load, on the longest day of the year. This was used in conjunction with a stainless steel absorptivity of 0.40, to give a net solar load of 120 Btu/hr.-ft.^{2}. This is comparable to the maximum insolation data given in Table 1, Regulatory Guide 7.8:

   $$\frac{1475 \text{ Btu}}{\text{ft.}^2} / 12 \text{ hours} = 122.9 \text{ Btu/hr.-ft.}^2$$

   The figure used in this analysis is conservative, in that it adopts an absorption coefficient, and is not figured as a sine function (which would give a lesser net Rms value to the amount of insolation).

3. Effects of the stainless steel cladding on the inner jacket were ignored.

4. Effects of the side impact limiters, which protrude out radially, at the closure end of the cask, were ignored in this analysis. This is conservative in the case of the fire, inasmuch as their conduction path is long, and they would shield a portion of the outer shell from the thermal radiation of the fire, and also from the solar loading.

5. The thermal volume and mass of the top and bottom of the cask was increased to account for the extra mass and surface area of the end impact limiters.

6. All radiation shape factors (F) have been assumed to be equal to 1.0, as virtually all heat emitted from one radiating surface is received by the next surface.

7. Standard English units were used throughout the analysis. Units used:
   Length: Feet
   Heat: Btu
   Time: Hour

   Steady state analyses were performed for both 100°F, and 130°F ambient air temperatures. Engineering judgement was used to assess when the system had come up to a "steady-state" condition.

8. Thermal effects of the payload were ignored.

3.4.2 Maximum Temperatures

Maximum temperatures of the CNS-3-55 cask package for Normal Transport Conditions are listed below.

o Normal Transport Conditions

Temperatures, °F *

Ambient Outside Air	Inner Cavity	Lead Shield	First O-ring	Second O-ring	Outside Surface
100°F	130.2	130.1	129.3	129.2	129.4
130°F	161.2	161.0	161.1	160.0	160.3

3.4.3 Minimum Temperatures

The minimum possible temperatures of the package could approach -40°F. The planned payloads may have very low decay heats. The steel and lead materials of construction of the cask are not significantly affected by an ambient temperature of -40°F. (See Section 2.6.2 for discussion.)

3.4.4 Maximum Internal Pressures

Maximum internal pressure for normal conditions of transport is less than 3 psig at an internal temperature of $160^{o}F$. Referencing Section 3.5.4, Internal Pressure calculations, air expansion within the cask will be as follows (for 60% of volume occupied by payload):

Mass of air taken into cask upon loading - 2.04 lbm air

T - $160^{o}F = 620^{o}R$

R - universal gas constant = $53.3 \frac{ft.-lbf}{lbm\ ^{o}R}$, for air

$V = 27.3\ ft.^3$

$$P = \frac{MRT}{V} = \frac{(2.04)(53.3)(160 + 460)}{27.3}$$

$$= 2469\ lbf/ft.^2 = 17.14\ psia$$

Total mass of components within the cask, at 60% payload volume:

water - .16 lbm

air - 2.04 lbm

2.20 lbm total

Partial pressures within the cask will be proportional to the appropriate component masses:

For air: $17.14 \times \frac{2.04}{2.20} = 15.9$ psia

For water, it is assumed that since the $160^{o}F$ temperature is below $212^{o}F$, and the pressures are relatively low, that all water will be present in liquid form. Under these conditions, the water will contribute a proportionate amount of the vapor pressure at $160^{o}F$:

$P_{water\ vapor}$ at $160^{o}F = 4.717$ psia

Pressure due to water = $\frac{.16}{2.2} \times 4.717 = .34$ psia

Total pressure in cask, under normal transport conditions:

$.34 + 15.9 = 16.24$ psia = 1.54 psig