✦ GENERAL ATOMICS

71-9277

June 7, 2001
SHP/NRC-3358-2

VIA EXPRESS DELIVERY
Mr. Timothy J. McGinty, Acting Chief
Licensing Section
Spent Fuel Project Office
Office of Nuclear Material Safety and Safeguards
U.S. Nuclear Regulatory Commission
Washington, DC 20555

Subject: Request for Renewal of Certificate of Compliance for the Model No. FSV-1 Unit 3
             Shipping Cask (CoC No. USA/9277/B( )F); **Submittal of Revised Report**

Reference: Asmussen, Keith E., Letter No. SHP/NRC-3358, dated April 11, 2001, Request for
             Renewal of Certificate of Compliance for the Model No. FSV-1 Unit 3 Shipping Cask
             (CoC No. USA/9277/B( )F)

Dear Mr. McGinty:

     General Atomics (GA) previously submitted a request for renewal of its Certificate of
Compliance for the Model No. FSV-1 Unit 3 Shipping Cask (CoC No. USA/9277/B( )F) (Ref.).
Subsequently, via a telephone conversation with a member of your staff, it was brought to our
attention that some references to regulations in the consolidated design report submitted in support
of our request, needed to have the citations updated. Accordingly, GA has revised six (6) pages,
to update the citations referencing applicable regulations, and reissued GADR 55, "Consolidated
Design Report for the Model FSV-1 Shipping Cask," Volumes I & 2. Revised copies of these two
volumes are enclosed. Also enclosed is a list of the pages that were revised and the corresponding
changes made.

     We appreciate your assistance, and apologize for the confusion and inconvenience. GA
trusts that you now have everything needed to complete the renewal of the Model No. FSV-1 Unit
3 Shipping Cask Certificate of Compliance.

     If I can be of further assistance, please do not hesitate to contact me at (858) 455-2823.

                              Very truly yours,

                              Keith E. Asmussen

                              Keith E. Asmussen, Ph.D., Director
                              Licensing, Safety and Nuclear Compliance

Enclosures: Volumes I and II of GADR 55 (GA Doc. No. 910013), Rev. C (1 copy)
              Change Notice for GA Doc. No. 910013 dated May 31, 2001 (1 copy)

NMSS public

Rec'd from NMSS
on 06/11/01

GENERAL ATOMICS

GA 974 (REV.10/99E) *Roll 2319*

APPROVAL LEVEL 5	☐ R&D ☐ DV&S	CHANGE CLASS	SYSTEM NO.	CAGE CODE	CM APPROVAL/DATE	**CHANGE NOTICE**
QA LEVEL IA	☒ DESIGN ☐ T&E	☐ I ☐ II ☐ III ☒ NA	2589	4V360	**2 RELEASED** ▷ MAY 3 1 2001	

ORIGINATOR *31 May 01* R. J. Meyer	CHECKER		LEAD ENGINEER *31 MAY 01* R. J. Meyer		CN NUMBER *007074*

REASON: ☒ LICENSING ☐ SUPPL REQ'ST ☐ TECHNICAL ☐ RECORD ☐ MFG REQ'S	PARTS DISPOSITION:	PAGE 1 OF 1
☐ CODES/STDS/REG	☒ USE ☐ STOCK	☐ RELEASE BY CN
☐ CUST REQST: CORRESPONDENCE NO:	☐ RETROFIT ☐ REWORK	☒ REL BY INCORP
☐ OTHER	☐ SCRAP ☐ NA	☐ DO NOT INCORP

APPROVALS

		REF CRs
BRANCH MGR L. Johnson *5/31/01 for L. Johnson*	QUALITY ASSURANCE R. Maxwell *31 May 01*	CR
DEPT MGR L. Johnson *for L. Johnson*	PROJECT MANAGER L. Johnson *for L. Johnson 5/31/01*	CR
INTERFACE ASSUR R.J. Meyer *31 MAY 01*	PROJECT ENGINEER R. J MEYER *31 MAY 01*	CR

CWBS/W.O. NO.	DOCUMENT NUMBER	CURRENT REV	DOCUMENT TITLE	NEXT REV
02589.912	910013	B	Consolidated Design Report for the Model FSV-1 Shipping Cask	C

1. Change Vol. 1 Page 7-6, para 7.2, 10CFR20.205 to 10CFR20.1906

2. Change Vol. 1 Page 7-17, para 7.4, 10CFR20.205 to 10CFR20.1906

3. Change Vol. 1 Page 7-20, para 7.5, 49CFR173.427 to 49CFR173.428

4. Change Vol. II Page 7-19, para 7.2.1, 10CFR20.205 to 10CFR20.1906

5. Change Vol. II Page 7-24, para 7.2.2, 10CFR20.205 to 10CFR20.1906

6. Change Vol. II Page 7-28, para 7.3 49CFR173.427 to 49CFR173.428

JUSTIFICATION: The changes were requested by the NRC in order to obtain a renewal of the FSV-1 cask CoC.

☐	☐					ITEM NO	PART NO.	DESCRIPTION	MATL/SPEC	SHT NO
ADD	REM	QTY REQ'D						PARTS LIST		

CONSOLIDATED DESIGN REPORT
FOR THE MODEL FSV-1 SHIPPING CASK

CONFIGURATIONS A, B, C, D

◆ GENERAL ATOMICS

CONSOLIDATED DESIGN REPORT
FOR THE MODEL FSV-1 SHIPPING CASK

CONFIGURATIONS A, B, C, D

GENERAL ATOMICS

ROLL 2243-NC

GA 1485 (REV. 9/89)

ISSUE/RELEASE SUMMARY

	APPVL LEVEL	DISC.	QA LEVEL	SAFETY	SYS	DOC. TYPE	PROJECT	DOCUMENT NO.	REV
☐ R&D ☐ DV&S ☒ DESIGN			I A	NA					
	5	NA	SEISMIC NA	ELECTRICAL NA	NA	LD	2589	910013	C

TITLE:

CONSOLIDATED DESIGN REPORT FOR THE MODEL FSV-1 SHIPPING CASK   VOLUMES I AND II

CM APPROVAL/ DATE	REV	PREPARED BY	APPROVAL(S)			REVISION DESCRIPTION/ W.O. NO.
			ENGINEERING	QA	PROJECT	
JAN 1 4 1992 3 RELEASED ▷	N/C	R.J. Meyer *R.J.Meyer 12-6-91*	N.D. Rickard *N.D. Rickard 1-9-92*	R.F. Maxwell *signature*	R.M. Grenier *signature 1-10-92*	2970620 Initial Release
2 RELEASED ▷ FEB 1 3 1995	A	J. Boshoven *signature 2/10/95*				(ROLL 2281) Released by CN 006423
2 RELEASED ▷ MAR 1 4 1996	B	R.J. Meyer *R.J.Meyer 3/13/96*				Released by CN 006486 (ROLL 2306)
2 RELEASED ▷ MAY 3 1 2001	C	R.J. Meyer *R.J.Meyer 3/1/01*				Released by CN 007074 (ROLL 2319)

CONTINUE ON GA FORM 1485–1

NEXT INDENTURED DOCUMENT(S)

NA

List of Effective Pages

Page Number	Page Count	Revision
i	1	C
ii to iii	2	C
iv	1	N/C
v	1	A
1-1	1	N/C
1-2	1	B
1-3 to 1-15	13	N/C
2-1 to 2-14	14	N/C
2-14a	1	N/C
2-15 to 2-38	24	N/C
2-39 to 2-40	2	B
2-41 to 2-74	34	N/C
2-74a to 2-74h	8	N/C
2-75 to 2-87	13	N/C
3-1 to 3-7	7	N/C
3-7a	1	N/C
3-8 to 3-45	38	N/C
4-1 to 4-7	7	N/C
5-1 to 5-9	9	N/C
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7-17	1	C
7-18 to 7-19	2	N/C
7-20	1	C
7-21 to 7-26	6	A
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9-1 to 9-2	2	N/C
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SECTION 1.0

GENERAL INFORMATION

1.0   GENERAL INFORMATION

1.1.   INTRODUCTION

Volume I of GADR 55 provides a description and an evaluation of Model FSV-1 in Configurations A, B, C and D and evaluations and diagrams that may apply to all configurations A, B, C, D, E, F and G.

Volume II of the GADR 55 provides a description and an evaluation of the Model FSV-1 packaging in Configurations E, F and G.

Table 1-1 provides a descriptive tabulation of the first four Model FSV-1 configurations. Figures 1-1 through 1-4 show the principal features of these Model FSV-1 configurations.

1.2.   PACKAGE DESCRIPTION

Model FSV-1 in Configurations A, B, C and D are designed and evaluated for the transport of large quantities of solid, nonfissile, irradiated and contaminated hardware. These packages have a loaded weight of approximately 46,000 pounds and the maximum weight of the cavity contents is 3,720 pounds. The length of the package is 208 inches with a maximum outside diameter of 31 inches for a length of 11.375 inches and the outside diameter of the remainder of the package is 28 inches. These configurations have been grouped together since the impact limiter is NOT used in any of these shipping configurations.

1.2.1.   Packaging

Model FSV-1 in Configurations A, B, C and D all use the same cask body as shown on National Lead Company drawings 70296F, Rev. 2; 70086F, Rev. 7; and General Atomics drawing 1501-003, Rev. C. Only Configuration A uses the inner container as shown on the above referenced drawings. Configuration B uses a

burial liner without supplemental shielding as shown on Fig. 1-2.
Configurations C and D have supplemental shielding in the form of a ring on
the cover plate for the burial liner. These configurations are described on
GA drawings GADR 55-2-10, issue D, and GADR 55-2-11, issue A and issue B. The
cover plate and supplemental shield ring for configuration C are made of
stainless steel as shown on GADR 55-2-11, issue B. In Configuration D these
components are made of carbon steel as shown on GADR 55-2-11 issue A.
Configurations C and D also have optional temporary shield tubes as shown on
drawing GADR 55-2-14, Issue N/C.

1.2.2 Operational Features

Model FSV-1 has a smooth external surface that simplifies decontamina-
tion. Lifting attachments on the package consist of sockets rather than
trunnions since sockets are less likely to be damaged in a manner that would
impair any safety function of the package. Tie-down of the package to the
rear support on the transport semitrailer is by means of four (4) socket head
cap screws which are installed into threaded inserts located in the base of
the cask body. This attachment arrangement prevents any damage that is likely
to impair any safety function of the package.

1.2.3 Contents of Packaging

The contents of Configurations A, B, C and D may consist of solid,
nonfissile, irradiated, and contaminated hardware. Examples of such reactor
hardware may include, but are not limited to, control rods, fuel channels,
poison curtains, shrouds, power range monitors, and miscellaneous structures.
Configurations C and D use optional temporary shield tubes inside the burial
liner for shipment of irradiated hardware requiring additional gamma
shielding.

TABLE 1-1
MODEL FSV-1 CASK CONFIGURATIONS A THROUGH D

Configuration	Reference Drawings	Authorized Contents	Allowable Weight-Cask Cavity	Allowable Weight-Contents	Remarks
A	70086F, Rev. 7[a] 1501-003, Rev. C[c] 70296F, Rev. 2[a]	Solid nonfissile irradiated and contaminated hardware	3720 lb	1800 lb	Impact limiter not used. Requires inner container. Loaded wt 46,025 lb
B	70086F, Rev. 7 1501-003, Rev. C[c] 70296F, Rev. 2	Solid nonfissile, irradiated and contaminated hardware	3720 lb	3080 lb	Impact limiter not used. Inner container not required; used with burial liner and coverplate. Loaded weight 46,025 lb.
C	70086F, Rev. 7 1501-003, Rev. C[c] 70296F, Rev. 2 GADR 55-2-10, Issue D GADR 55-2-11, Issue B (GADR-55-2-14, Issue N/C optional)	Solid nonfissile, irradiated, con-taminated hard-ware and optional temporary shield tube	3720 lb	2970 lb[b]	Impact limiter not used; inner con-tainer not required, used with burial liner. Supplemental shield ring and coverplate (both stainless steel) used for the burial liner. wt: 750 lb. Optional temporary shield tube may be used. Loaded weight 46,025 lb.
D	70086F, Rev. 7 1501-003, Rev. C[c] 70296F, Rev. 2 GADR 55-2-10, Issue D GADR 55-2-11, Issue A (GADR-55-2-14, Issue N/C optional)	Solid nonfissile, irradiated and contaminated hardware	3720 lb	2970 lb	Impact limiter not used; supplemental shield ring and coverplate (both carbon steel) used for burial liner. Optional shield tube may be used. wt: 750 lb Loaded weight 46,025 lb.

[a]National Lead Company drawing.

[b]Including optional shield tube.

[c]General Atomics drawing.

FIGURE 1-1

1-5

INNER CLOSURE

OUTER CLOSURE

CASK BODY

INNER CONTAINER

CENTER PLUG

PURGE CONNECTION COVER

REFERENCE DRAWINGS:

7008 6 F
1501-003 C
70296 F
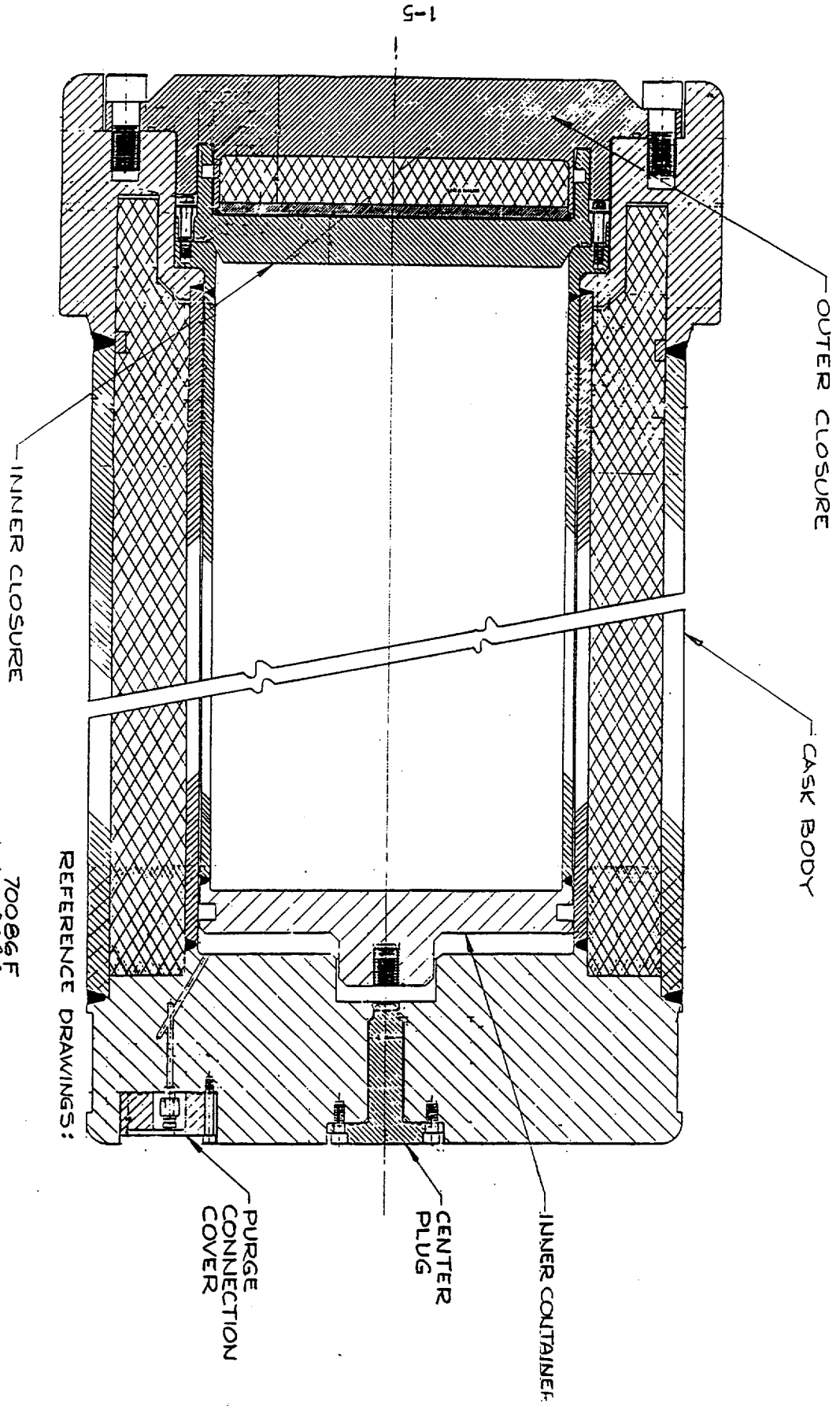
MODEL FSV-I CASK
CONFIGURATION A
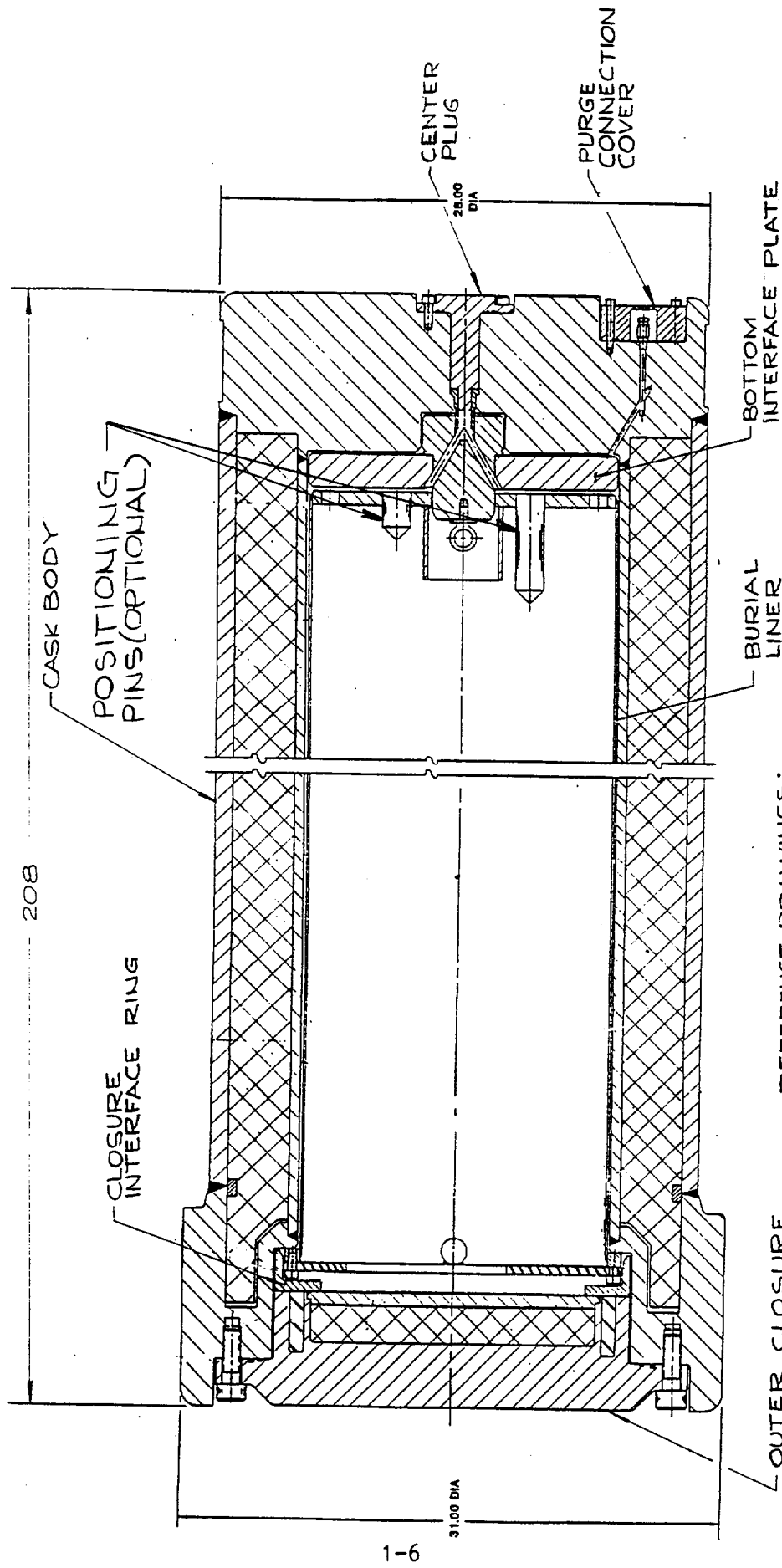
FSV-I CONFIGURATION **B**
FOR IRRADIATED HARDWARE
WITH BURIAL LINER

REFERENCE DRAWINGS:

70086F
1501-003C
70296F

FIGURE 1-2

GADR 55
Volume I

CENTER PLUG

PURGE CONNECTION COVER

BOTTOM INTERFACE PLATE

CASK BODY

POSITIONING PINS (OPTIONAL)

28.00 DIA

208

BURIAL LINER

CLOSURE INTERFACE RING

STAINLESS STEEL COVER AND SHIELD RING

OUTER CLOSURE

31.00 DIA
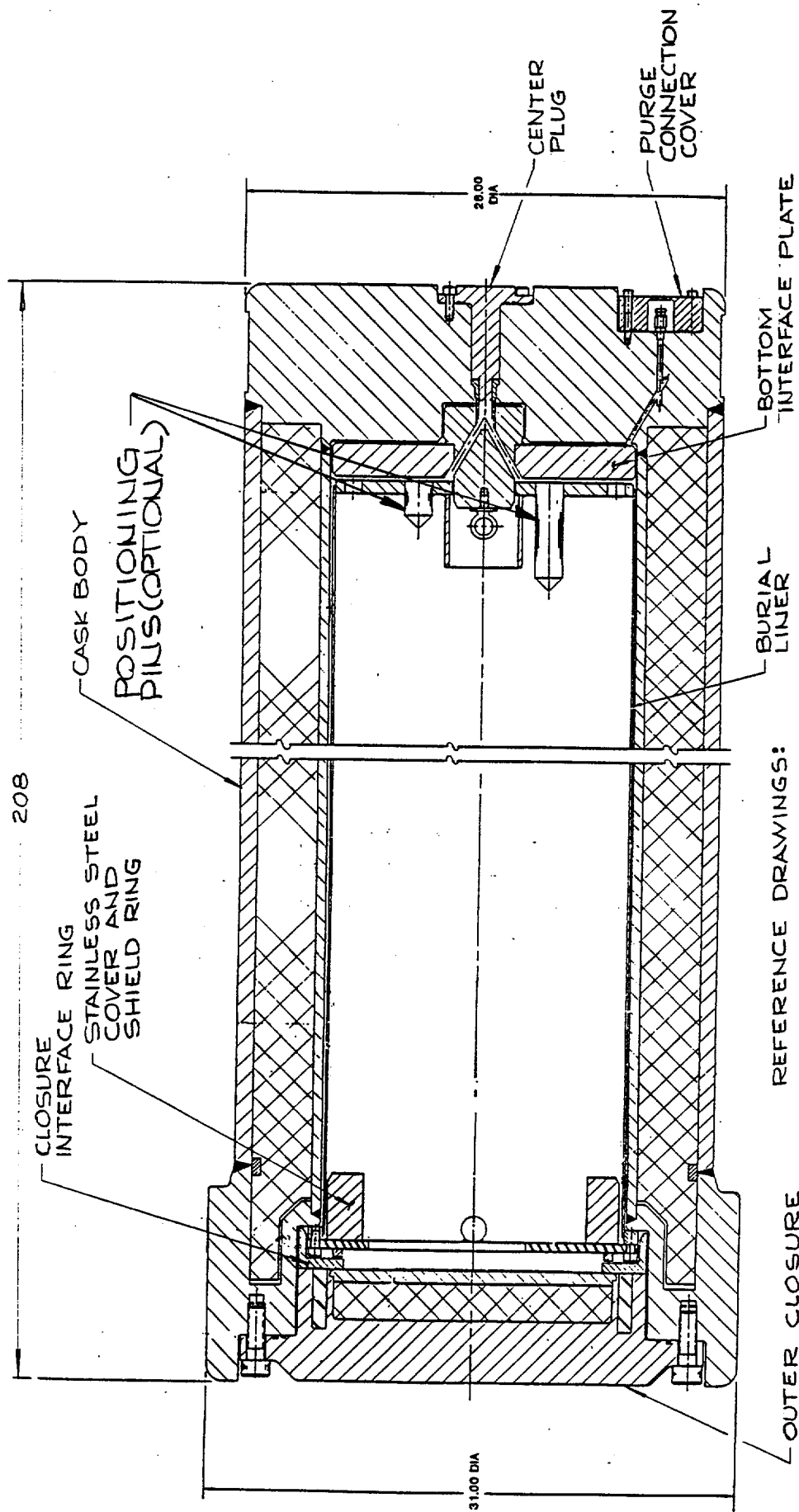
FSV-1 · CONFIGURATION C
FOR IRRADIATED HARDWARE
WITH SHIELDED BURIAL
LINER COVER

REFERENCE DRAWINGS:

70086F
1501-003C
70296F
GADR 55-2-10
GADR 55-2-11

FIGURE 1-3

BOTTOM PLUG

CENTER PLUG

PURGE CONNECTION COVER

28.00 DIA

BOTTOM INTERFACE PLATE

CASK BODY

POSITIONING PINS (OPTIONAL)

BURIAL LINER

208.0

CLOSURE INTERFACE RING

CARBON STEEL COVER AND SHIELD RING

OUTER CLOSURE

31.00 DIA

REFERENCE DRAWINGS:

70086F
1501-0030
70296F
GADR 55-2-10
GADR 55-2-11
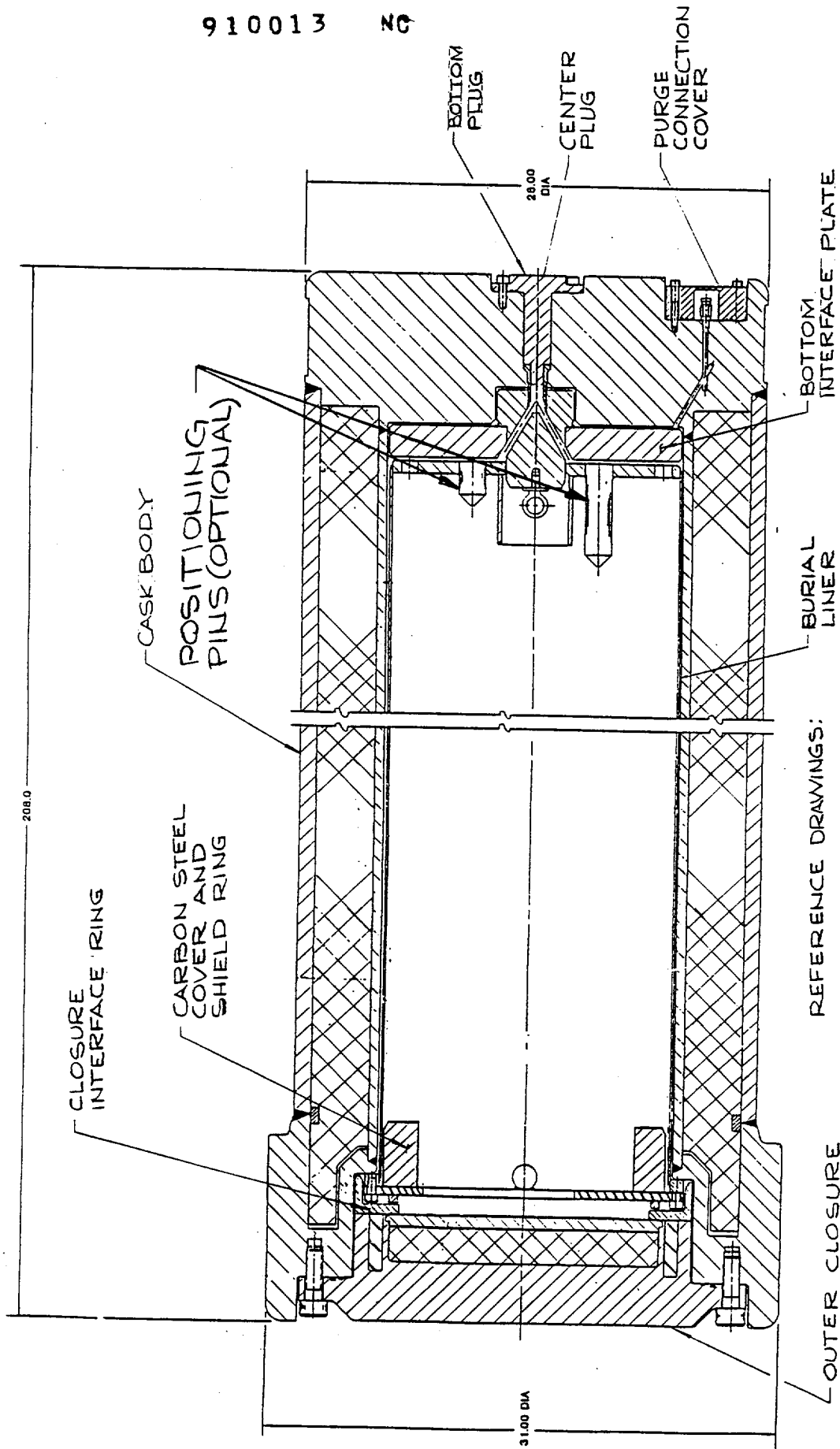
FSV-1   CONFIGURATION D
FOR IRRADIATED HARDWARE
WITH SHIELDED BURIAL LINER
COVER - CARBON STEEL

FIGURE 1-4

FIGURE WITHHELD UNDER 10 CFR 2.390

FIGURE WITHHELD UNDER 10 CFR 2.390

(Applicable   for   Configurations   A,B,C   and   D)

FIGURE WITHHELD UNDER 10 CFR 2.390

FIGURE WITHHELD UNDER 10 CFR 2.390

Configuration C
Supplemental Shield Ring and Cover Plate
(Stainless Steel)

GADR 55
Volume I

FIGURE WITHHELD UNDER 10 CFR 2.390

Configuration D
Supplemental Shield Ring and Cover Plate
(Carbon Steel)

GADR 55
Volume I

FIGURE WITHHELD UNDER 10 CFR 2.390

Configurations C and D
Shield Tube (Optional)

FIGURE WITHHELD UNDER 10 CFR 2.390

SECTION 2.0

STRUCTURAL EVALUATION

2.0 STRUCTURAL EVALUATION

2.1 STRUCTURAL DESIGN

2.1.1 Discussion

Principal structural components of Model FSV-1 in Configurations A, B, C, and D are the cask body with outer closure and either an inner container or an inner coverplate. Configurations A through D do not require use of the impact limiter.

2.1.2 Design Criteria

Model FSV-1 is designed to comply with the regulations in regard to the release of radioactive material and external radiation dose rate for the normal conditions of transport and the hypothetical accident conditions. Model FSV-1 was designed and evaluated in accordance with the applicable regulations that were in effect prior to April 1969.

2.2 WEIGHTS AND CENTERS OF GRAVITY

A summary of the weights of model FSV-1 Configurations A, B, C, and D is presented in Table 2-1. Model FSV-1 cask body Configurations A, B, C, D have been evaluated for the normal conditions of transport and the hypothetical accident conditions while containing a total weight of 3720 pounds. This total weight consists of 1920 pounds for the inner container and 1800 pounds for the radioactive contents.

2.2.1 Weight Calculations - Model FSV-1 Configuration A

TABLE 2-1

	Configuration A	Configuration B	Configuration C	Configuration D
Cask Body	42,305	42,305	42,305	42,305
Inner Container	1,920	not used	not used	not used
Burial Liner/ Spacer	not used	640	750	750
Contents - Allowable	1,800	3,080	2,970	2,970
Total	46,025	46,025	46,025	46,025

The center of gravity of Model FSV-1 Configurations A, B, C, and D is located 105.5 inches from the bottom of the package.

2.2.1.1   Cask Closure
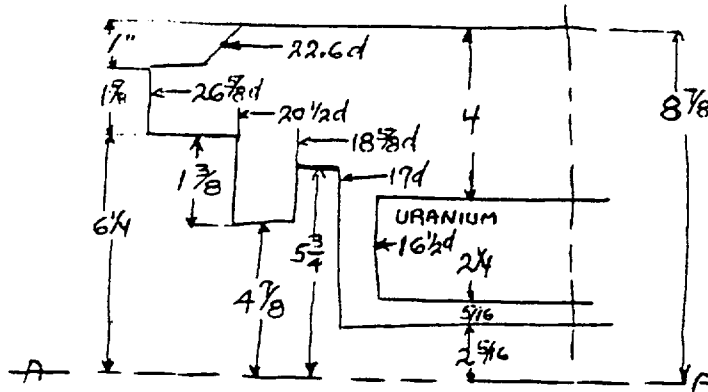
22.6 d $400 in^2 \times 1 = 400 in^3$ $x -8\frac{3}{8} = -3350$

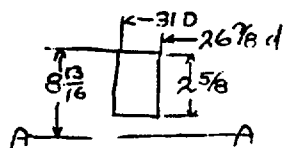26⅞d $560 \times 1\frac{5}{8} = 910$ $x -7\frac{1}{16} = -6430$

20½d $330 \times 1\frac{3}{8} = 455$ $x -5\frac{9}{16}$ -2530

17 d $227 \times 3\frac{7}{16} = 780$ $x -4\frac{1}{32} = -3140$

$+ 2545 in^3$ $-15,450 . in^4$

-18⅞d $-273 \times \frac{7}{8} = -239$ $x 5\frac{5}{16}$ $+1330$

-16½d $-214 \times 2\frac{1}{4} = -482$ $x 3\frac{3}{4} = +1809$

$\frac{- 721}{+1824 in^3}$ $\frac{+ 3139}{-12,311 in^4}$

$.29$ $.29$

(SS) $wt = 530 lbs$ $M = -3580 in.lbs.$

Uranium
16½d $+214 \times 2\frac{1}{4} = +482$ $x -3\frac{3}{4} = -1809$

$.683$ $.683$

$330 lbs$ $M = -1235 in lb$

TOTAL WT.
HEAD = 860 LBS. $M = -4815 in lb.$

2.2.1.2 Cask Body

$31d$ $+752 \text{in}^2$

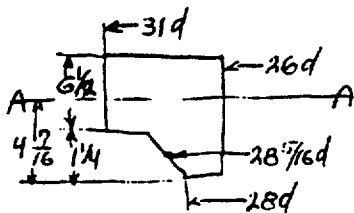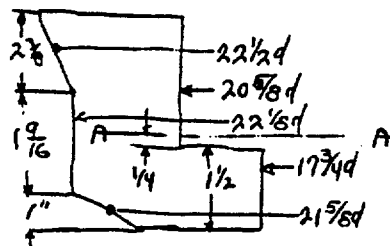$26\frac{7}{8}d$ $\dfrac{-568}{184} \times 2\frac{5}{8} = 483 \text{in}^3 \times -7\frac{1}{2}" = -3622$

$wt = \dfrac{.29}{140\,lbs}$ $M = \dfrac{.29}{-1050 \text{ in lb}}$

$31d$ $+752 \text{in}^2 \times 2\frac{7}{8} = 2160 \text{in}^3 \times -4\frac{3}{4}" = -10,250.$

$26d$ $+530 \times \frac{1}{8} = \dfrac{66}{+2226} \times -3\frac{1}{4} = \dfrac{-216.}{-10,466}$

$-20\frac{7}{8}d$ $-335 \times 3 = \dfrac{-1005}{+1221 \text{in}^3} \times -4\frac{11}{16} = \dfrac{+4850}{-5616.}$

$Wt = \dfrac{.29}{354.\,lbs}$ $M = \dfrac{.29}{-1628 \text{ in lbs}}$

$31d$ $752 \text{in}^2 \times 6.5 = 4880 \times -\frac{1}{16} = -305.$

$28\frac{7}{8}d$ $630 \times 1\frac{1}{4} = \dfrac{787}{+5667} \times +3\frac{13}{16} = \dfrac{+3000.}{+2695}$

$-26d$ $-530 \times 7\frac{3}{4} = \dfrac{-4100}{+1567 \text{in}^3} \times +\frac{9}{16} = \dfrac{-2300}{+395.}$

$Wt = \dfrac{.29}{455\,lbs}$ $M = \dfrac{.29}{+114 \text{ in lbs}}$

$22\frac{1}{2}d$ $397\text{in} \times 2\frac{3}{8} = 942 \text{in}^3 \times -2\frac{1}{4} = -2120$

$22\frac{1}{8}d$ $385 \times 1\frac{9}{16} = 601 \times -\frac{9}{32} = -169$

$21\frac{5}{8}d$ $367 \times 1 = \dfrac{367}{+1910} \times +1\frac{1}{4} = \dfrac{+458}{-1831}$

$-20\frac{5}{8}d$ $-335 \times 3\frac{7}{16} = -1154 \times -1\frac{15}{32} = +1670$

$-17\frac{3}{4}d$ $-247 \times 1\frac{1}{2} = \dfrac{-370}{\underline{-1524}} \times +1 = \dfrac{-370}{\underline{+1300}}$

$\dfrac{+386}{.29}$ $\dfrac{-531}{.29}$

$Wt = \dfrac{}{112\,lbs}$ $W = \dfrac{}{-154 \text{ in lbs}}$

2.2.1.2 Cask Body (Continued)
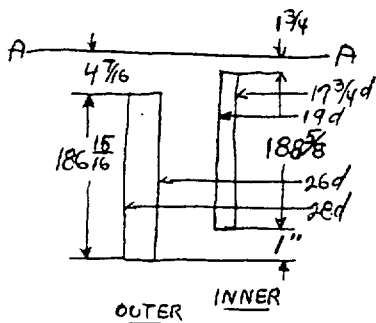
$$28d \quad 613.\text{in}^2$$
$$-26d \quad \frac{-530}{83\text{in}^2} \times 186\frac{15}{16} = 15,500\text{in}^3 \quad \times +97\frac{29}{32} = +1,511,000.$$

(outer) $wt \dfrac{.29}{4500\,lbs}$ $M = \dfrac{.29}{+439,000}$

$$19d \quad 283.5\text{in}^2$$
$$-17\frac{3}{4}d \quad \frac{-247}{36.5\text{in}^2} \times 188\frac{5}{8} = 6870\text{in}^3 \quad \times +9606 = +660,000$$

(inner) $wt = \dfrac{.29}{1990\,lbs}$ $M = \dfrac{.29}{191,200\,''lb}$
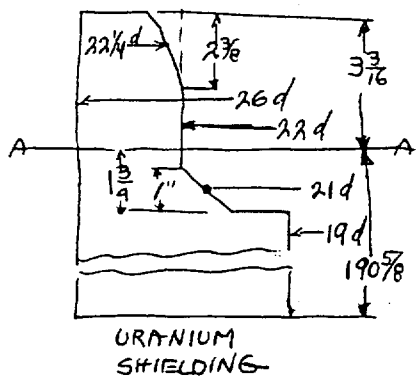
$$26d \quad 530\text{in}^2 \times 193\frac{13}{16} = 102,750\text{in}^3 \quad \times +93\frac{23}{32} = +9,635,000.$$

$$-19d \quad -283.5 \times 188\frac{3}{8} = -53,500 \quad \times +96\frac{7}{16} = -5,150,000$$
$$-21d \quad -346 \times 1 \quad = -346. \quad \times +1\frac{1}{4} = - \quad 432$$
$$-22d \quad -380 \times 1\frac{9}{16} = -594 \quad \times -\frac{1}{32} = + \quad 19$$
$$-22\frac{1}{4}d \quad -389 \times 2\frac{3}{8} = - \quad 925 \quad \times -2 = + \quad 1850$$

$$\frac{-55,365}{+47,385} \qquad \frac{-5,148,563}{+4,481,437}$$

URANIUM $wt \dfrac{.683}{32,400\,lbs}$ $M = +3,060,000^{''l}$
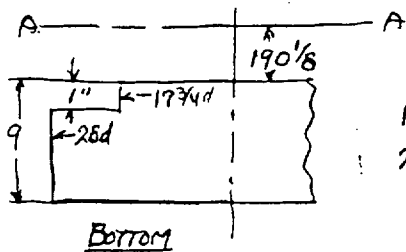
$$17\frac{3}{4}d \quad 247\text{in}^2 \times 1 = 247\text{in}^3 \quad \times +190\frac{5}{8} = +47,100$$
$$28d \quad 613 \times 8 = \frac{4904}{+5151} \quad \times +195\frac{1}{8} = \frac{+960,000}{+1,007,100.}$$

$$wt = \frac{.29}{1494\,lbs} \qquad M = \frac{.29}{292,000\,inlb}$$

OUTER   INNER

URANIUM SHIELDING

BOTTOM

2.2.1.3 Inner Container

HEAD
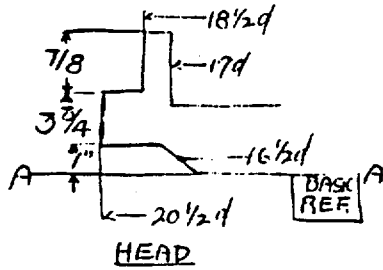
$18\frac{1}{2}d$ $268\,in^3 \times \frac{7}{8} = 234\,in^3 \times -5\frac{3}{16} = -1212$

$20\frac{1}{2}d$ $330 \times 3\frac{3}{4} = 1238 \times -2\frac{7}{8} = -3560$

$16\frac{1}{2}d$ $213 \times 1 = \underline{213} \times -\frac{1}{2} = \underline{-107}$

     $+1685\,in^3$ -4879

$-17d$ $-227 \times 3\frac{3}{8} = \underline{-766} \times -3\frac{15}{16} = \underline{+3010}$

     $+919$ -1869

     $.29$ $.29$

   $Wt = 266\,lbs$ $M = -542$
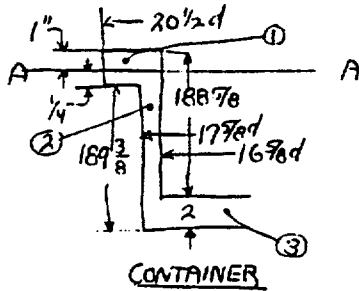
CONTAINER

$Wt_1 = \dfrac{330}{\underline{-217}}{113} \times 1\frac{1}{4} \times .29 = 41. \qquad \times -\frac{7}{8} = \qquad -16.$

$Wt_2 = \dfrac{244}{\underline{-217}}{27} \times 187\frac{3}{8} \times .29 = 1470. \qquad \times +93\frac{15}{16} = +138,000.$

$Wt_3 = 244 \times 2 \times .29 = \underline{142.} \qquad \times 185\frac{7}{8} = \underline{+26,800}$

     $Wt = 1653\,lbs \qquad M = +164,784\,in\#$

CONTENTS OF
CONTAINER

$300^\# \times 6 = \underline{1800\,lbs} \qquad\qquad \times 93\frac{27}{32} = M = +169,000\,in\#$

2.2.1.4 Weight Summation

		WT.	MOMENT
CASK	HEAD	860 LBS	-4815.
	BODY	140	-1050.
		354	-1628.
		455	+ 114.
		112	- 154.
		4500	+ 437,000.
		1990	+ 191,200
		32400	+ 3,060,000
		1494	+ 292,060
CASK -TOTAL EMPTY		42,305	+ 3,474,727
CONTAINER		266	- 542
		1653	+ 164,784
CONTENTS		1800	+ 169,000

. CASK -TOTAL -LOADED 46,024 lbs x [93·6] = + 4,307,969 in lbs,

2.2.2   Weight Calculation - Configuration B

2.2.2.1   Bottom Interface Plate

Plate: $(17.625^2 - 3.76^2) \, \pi/4 \times 1.94 \times 0.285 =$ 129

Pins: $[(3.75^2 \times \pi/4 \times 3.75) + (2.25^2 \times \pi/4 \times 2.25)] \, 0.285 =$ 14

<div align="right">Total   143 lb</div>

2.2.2.2   Burial Liner

Shell: $(17.5^2 - 17.26^2) \, \pi/4 \times 187.5 \times 0.285 =$ 350

Base: $[(17.26^2 - 3.91^2) \, \pi/4 \times 0.75 + (1.625^2 \times \pi/4$
        $\times 7.38)] \, 0.285 =$ 52

Flange: $(19.25^2 - 17.5^2) \, \pi/4 \times 0.75 \times 0.285 =$ 11

Cover: $[(19.25^2 \times \pi/4 \times 0.5) - (9.0 \times 2.0 \times 0.5) +$
        $(1.5^2 \times \pi/4 \times 15)] \, 0.285 =$ 46

<div align="right">Total   459 lb</div>

2.2.2.3   Closure Interface Ring

$[(19.5^2 - 15^2) \, \pi/4 \times 0.625 + (20.44^2 - 19.5^2)$
   $\times \pi/4 \times 2.38] \quad 0.285 =$ 42 lb

2.2.2.4   Total Allowable Content Weight - Configuration B

Model FSV-1 has been evaluated for a total weight of 46,024 lb of this, 3,719 lb were in the cavity of the cask body.

The bottom interface plate, the burial liner and the closure interface ring have a total weight of 644 lb, therefore, the allowable weight of the contents is:

$$3719 - 644 = 3075 \text{ lb (use 3080)}$$

2.2.3   Weight Calculation - Configurations C and D

2.2.3.1   Shield and Impact Ring

$$\left\{ \left[(17^2 - 13^2) \ \pi/4 \times 4 \right] + \left[(16^2 - 15^2) \ \pi/4 \times 0.5 \right] \right\} \ 0.285 = \quad 111 \text{ lb}$$

2.2.3.2   Optional Temporary Shield Tubes

$$\text{1-in. thick:} \quad \left\{ \left[(6^2 - 4^2) \ \frac{\pi}{4} \times 144 \right] + \left[(6^2) \ \frac{\pi}{4} \times 9 \right] \right\} 0.285 = 717 \text{ lb}$$

$$\text{2-in. thick:} \quad \left\{ \left[(6^2 - 2^2) \ \frac{\pi}{4} \times 144 \right] + \left[(6^2) \ \frac{\pi}{4} \times 9 \right] \right\} 0.285 = 1104 \text{ lb}$$

2.2.3.3   Total Allowable Weight - Configurations C and D

The 3,719 lb allowable for the cavity of the cask body is also applicable to the Configurations C and D. The weight of the shield ring and impact ring is added to the 644 lb from Section 2.2.2.4. Now the allowable weight of the contents is:

$$3719 - (644 + 111) = 2964 \text{ lb (use 2970)}$$

When an optional temporary shield tube is used, its weight will be considered part of the contents weight.

2.3 MECHANICAL PROPERTIES OF MATERIALS

2.3.1 Model FSV-1 Packaging (Configurations A, B, C, D)

2.3.1.1 "As-Cast" Unalloyed Depleted Uranium

Density

    18.9 grams/cc or 0.683 lb/in^3

Mechanical Properties

Ultimate Tensile Strength	60,000 to 100,000 psi
Yield Strength	25,000 to 45,000 psi
Elongation	8% to 15%
Modulus of Elasticity	Approximately 24 (10)^6 psi
Hardness	Rockwell B 65 to 90

Thermal Expansion   6.5 (10)$^{-6}$ in/in°F

2.3.1.2 Stainless Steel Pipe, Type 304 per ASTM Spec. A-351, Grade CF-8

Physical Properties

Density	0.287 lb/in.^{3}
Specific Gravity	7.94 grams/cc
Melting Range	2550° to 2650°F
Modulus of Elasticity	28 x 10^6 psi
Specific Heat (32° to 212°F)	0.12 Btu/lb/°F

Thermal Conductivity

At 200°F	9.4 Btu/hr/ft^2/°F/ft
At 1000°F	12.5 BTU/hr/ft^2/°F/ft

Mean Coefficient of Thermal Expansion

32° to 212°F	9.6 in./in./°F x 10^6
32° to 600°F	9.9 in./in./°F x 10^6
32° to 1000°F	10.2 in./in./°F x 10^6

Mechanical Properties (at 72°F)

Ultimate Tensile Strength	70,000 psi
Yield Strength	30,000 psi
Elongation	35%
Hardness	R_B 88

2.3.1.3 <u>Stainless Steel Forgings, Type 304 per ASTM Spec. A-182, Grade F-304</u>

Physical Properties

Same as 2.3.1.2

Mechanical Properties

Ultimate Tensile Strength	70,000 psi
Yield Strength	30,000 psi
Elongation	40%
Hardness	R_B 88

2.3.1.4 <u>Uranium Welds</u>. All uranium welding will be accomplished using single V-butt joints and inert direct current tungsten arc welding. The inert gas used for shielding and trailing shields shall be welding grade argon. The filler and base metals shall be depleted uranium.

2.3.1.5 <u>Stainless Steel Welds (Refs. 2-1 and 2-2)</u>. All stainless steel welds will be in accordance with the "Rules for Construciton of Nuclear Vessels" (Ref. 2-1). All of the welding procedures and welders will be qualified in accordance with "Welding Qualifications" (Ref. 2-2).

2.3.1.6 <u>Seals (Refs. 2-3, 2-4, and 2-19)</u>.

1. All metal O-rings shall be self-energized for use in bolted flange assemblies. These O-rings will be made of silver plated Inconel X tubing. Service temperature is $-320°$ to $+1300°F$. The following metal O-rings or equivalent have been selected for sealing Model FSV-1 (Configurations A through D).

    a) Cask Closure Seal - United Aircraft products, Inc. Cat. No. U-6420-22000-SEA; OD = 22.01; ID = 21.76; Tube Dia. = 0.125.

    b) Inner Container Closure Seal - United Aircraft Products, Inc. Cat. No. U-6420-17430-SEA; OD = 17.44; ID = 17.19; Tube Dia. = 0.125.

    c) Cask Center Plug Seal - United Aircraft Products Inc. Cat. No. U-6420-02813-SEA; OD = 2.81; ID = 2.56; Tube Dia. = 0.125.

    d) Cask Purge Connection Cover Seal - United Aircraft Products, Inc. Cat. No. U-6420-02630-SEA; OD = 2.62; ID = 2.38; Tube Dia. = 0.125.

All elastomer O-rings shall be molded per AMS Specification 3304 of silicone rubber or equivalent. Service temperature for this material is $-100°$

to +500°F. The material will resist temperatures up to 700°F for short
periods. The following silicone rubber O-rings or equivalent have been
selected for sealing Model FSV-1 (Configurations A through D).

  a) Cask Closure Seal - Parco No. PRP-568-392; OD = 23.375; ID - 23.00;
      Dia = 0.187.

  b) Inner Container Closure Seal - Parco No. PRP-568-386; OD = 17.375; ID
      = 17.00; Dia. = 0.187.

  c) Cask Center Plug Seal - Parco No. PRP-568-236; OD = 3.500; ID = 3.25;
      Dia. = 0.125.

  d) Cask Purge Connection Cover seal - Parco No. PRP-568-233; OD = 3.125;
      ID = 2.875; Dia. = 0.125.

2.3.1.7 Fasteners (Refs. 2-5, 2-6, and 2-7)

  a. The bolts used in the assembly of the inner container are alloy steel
      per AMS 6322 with cadmium plating per QQ-P-416, Type 11, Class 3.
      These bolts are heat treated for 180KSI min, ultimate tensile
      strength with a hardness of R_C40-44. The 1/2-inch size used in
      fastening the inner container closure to the container body has a
      minimum axial tensile strength of 26,700 lb. (Ref. 2-5).

  b. The bolts of the inner container closure are threaded into "screw-
      lock" inserts made of Type 18-8 stainless steel (per AMS-7245B) wire
      having an ultimate tensile strength of approximately 200,000 psi.
      These "screw-lock" inserts meet military specification for locking
      torque and vibration. The internal thread conforms to thread form
      standards issued by the Department of Commerce (Ref. 2-6).

c. The bolts used in the assembly of the cask closure to the shipping
cask body shall be cadmium plated per QQ-P-416 alloy steel per
FF-S-86C with the following physical properties for 1-1/4-7 UNC size
(Ref. 2-7):

Tensile Strength, min.	160,000 psi
Yield Strength, min.	130,000 psi
Heat Treatment	$R_C 36-43$

2.3.2 Configurations B, C and D

The following materials are used in addition to those used in the model FSV-1 Configuration A. Only Configuration D uses the stainless steel in the burial liner cover and the shield ring.

2.3.2.1 Carbon Steel (ASTM A36)

Physical Properties

Density	490 lb/ft^3
Modulus of Elasticity	30 x 10^6 psi
Specific Heat	0.11 Btu/lb°F

Thermal Conductivity

At 32°F	26.5 Btu/hr-ft°F
At 212°F	26.0 Btu/hr-ft°F
At 572°F	25.0 Btu/hr-ft°F

Coefficient of Thermal Expansion

70 - 250°F	6.77 x 10^{-6} in/in°F
70 - 550°F	7.34 x 10^{-6} in/in°F

Mechanical Properties

Ultimate Tensile Strength	58,000 - 80,000 psi
Yield Strength	36,000 psi
Elongation	23%

2.3.2.2 Stainless Steel Type 304 (ASTM A240)

Physical Properties:

Density	496 x lb/ft^3
Modulus of Elasticity	28 x 10^6 psi
Specific Heat	0.12 Btu/lb°F

Thermal Conductivity:

At 32°F	8.0 Btu/hr-ft°F
At 212°F	9.4 Btu/hr-ft°F
At 572°F	10.9 Btu/hr-ft°F

Coefficient of Thermal Expansion:

70 to 250°F	8.54 x 10^{-6} in/in°F
70 to 550°F	9.33 x 10^{-6} in/in°F

Mechanical Properties:

Ultimate Tensile Strength	75,000 psi
Yield Strength	30,000 psi
Elongation	40%

2.4 GENERAL STANDARDS FOR ALL PACKAGES

2.4.1 Chemical and Galvanic Reactions

Investigations have shown that uranium combines with stainless steel by solid state diffusion at temperatures above 1000°F. The iron-uranium eutectic melts at 1337°F so that if the two materials are in intimate contact at this temperature a molten alloy will be formed (Ref. 2-8).

Other investigations have shown that uranium in contact with stainless steel will penetrate the stainless steel by solid state diffusion in 24 hours at 1400°F. At 1355°F there was no attack on the stainless steel.

Recent tests of stainless steel - uranium - stainless steel assemblies wherein the surfaces of the stainless steel next to the uranium were spray coated with a 0.005-inch-thick coating of copper showed this coating to be an effective barrier to diffusion between the stainless steel and uranium at temperature up to 1750°F (Ref. 8). All surfaces of stainless steel in contact with the depleted uranium shielding will be coated with 0.005-inch-thick copper coating for the model FSV-1 packaging.

2.4.2 Positive Closure

Twenty four (24) high strength, socket head cap screws, torqued to 1000 foot-pounds are used to secure the outer closure to the cask body.

2.4.3 Lifting Devices

Model FSV-1 Configurations A through D are lifted by means of two sockets installed in machined recesses located in the enlarged diameter section of the

closure end of the cask body. A dedicated lifting device with a ball located on each arm is used to lift the package. The sockets are removable and can be replaced if damaged in any way.

2.4.4 Tiedown Devices

During transport the Model FSV-1 cask is attached to the rear support on the semitrailer by four (4) high strength socket head cap screws which are installed in threaded holes located in the base of the cask. The upper end of the package rests in a saddle and is restrained by a tie-down strap.

2.5 STANDARDS FOR TYPE B PACKAGING

2.5.1 Load Resistance
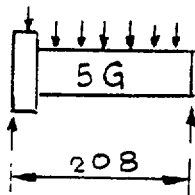
5g - Uniformly Distributed

WT = 46024 lb

W = 5(46024) = 230,120 lb

$$\text{Max at center} = \frac{W\ell}{8} = \frac{230120}{8} (208) = 6,000,000 \text{ in. lb}$$

Assume only outer shell is stressed

OD = 28" ID = 26"

$$I = \frac{\pi}{4}(R^4 - r^4) = \frac{\pi}{4}(14^4 - 13^4) = 7750 \text{ in.}^4$$

$$Z = \frac{7750}{14} = 554 \text{ in.}^3$$

$$S_b = \frac{M}{Z} = \frac{6,000,000}{554} = \underline{10,800 \text{ psi}} \text{ ok, less than } 30,000 \text{ psi}$$

2.5.2   External Pressure

25 psig Roark $\underline{XVI}$ Case 3c (Stability)

```
    ┌─────────────────────────┐
  ┌─┘                         │
  │        CONTAINER          │
  └─┐                         │
    └─────────────────────────┘
```

17-5/8 OD - 16-5/8 ID - 1/2 wall - 190-5/8 long

$$p^1 = \frac{1}{4} \frac{E}{1-\nu^2} \frac{t^3}{r^3} = \frac{1}{4} \frac{30(10^6)}{(1-.09)} \frac{(1/2)^3}{(8.31)^3} = \underline{1793} \text{ psi} \quad \text{o.k.}$$

2.6   NORMAL CONDITIONS OF TRANSPORT

2.6.1   Differential Thermal Expansions

2.6.1.1   Clearances. The cask and its container assemblies are stainless steel and uranium constructions. No lead is present and thus there are no problems associated with voids of this kind, which vary greatly in volume with changes in temperature and also shift in position within the cask. The uranium is monolithic and jacketed by the steel. The dimensional proportions of the cask shielding cylinder uranium require that there be minimal clearances for machining and assembly purposes. These clearances are a substantial part of the differential expansions which developed in several of the cases examined. The location of gaps and materials of construction are shown on Fig. 2-1.

2.6.1.2   Coefficients of Expansion. Coefficient of expansion used for depleted uranium is 6.5×10^{-6} in./in.°F. This value is obtained from records of the NL Company-Albany plant and refer specifically to as-cast 0.2% molybdenum uranium - unalloyed composition.
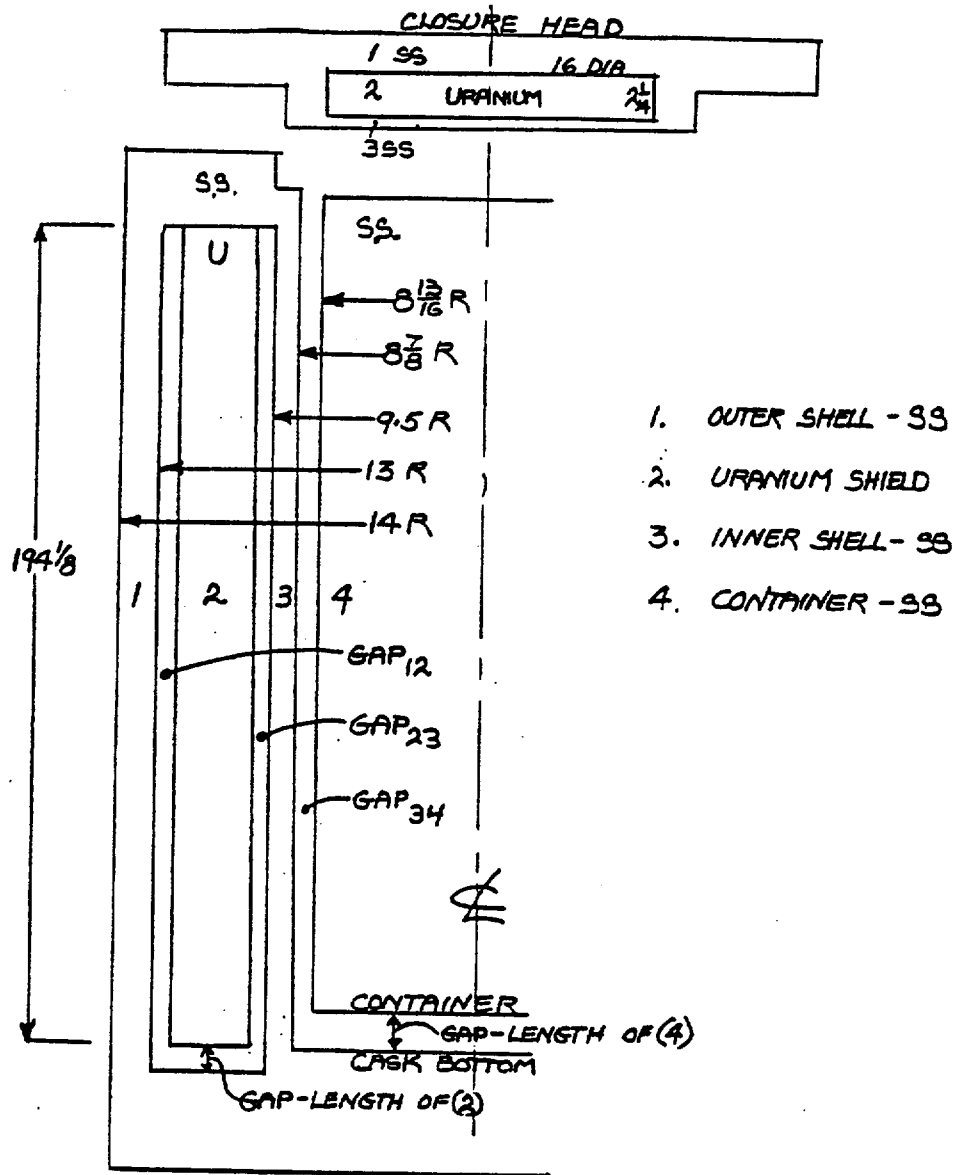
FIGURE 2-1   Location of Gaps and Materials for Model FSV-1
in Configurations A, B, C and D

Stainless Steel values are from Section III, Table N-426 of the Nuclear Code, as follows:

A = instantaneous values at given temperature

B = mean coefficient (from 70°F to indicated temperature)

A	B	Temp °F
9.11	9.11	70
9.73	9.47	300
10.43	9.82	600
10.90	10.05	800

2.6.1.3 Temperature Distribution. Temperature distribution through the cask under hypothetical accident conditions has been obtained from memorandum III, a part of the specification, dated 18 April 1968, and titled "Heat Transfer Calculations for PSC Fuel Shipping Cask". Heat generation rates were chosen in each case to give the maximum differential temperatures.

2.6.1.4 Cases Investigated. The following cases are investigated relative to axial and to radial differential expansions and contractions for the two uranium bodies contained within the stainless steel structure.

Case 1 - 30 minutes after start of fire (1101 Btu/hr fuel rate)

Case 2 - 10 hours after start of fire (2322 Btu/hr fuel rate)

Case 3 - start up - cask 70° - container and inner shell 240°

Case 4 - Immersion in water 70° - container and inner shell 240°

Case 5 - Low temp. - 40° whole cask - No container

Case 6 - Low temp. - 40° cask - Container and inner shell 240°

2.6.1.5 Analysis of Uranium Shielding in Cask Body

In the calculations + is a clearance or gap

$\qquad$ - is an interference (based on original 0 gap)

The various negative (-) dr or dl values thus indicate the minimum initial clearances at 70°F required to prevent interference and stressed conditions. These requirements are reflected in the drawings.

The maximum values required for such clearances for the cask itself are:

Gap 12 = -0.0322 from case 2

Gap 23 = -0.015 from case 2

Gap 34 = -.0367 from case 3

Length container = -0.309 from case 6

Interferences - to be prevented by suitable mfg. clearances.

Case 1

$T_1 = 1120°$ $T_3 = 430°$ $T\theta = 70°$ $T_4 = 370°$

<u>Gap 12</u> Assume $T_2 = T_3$ for max. diff.

$dr = (13")(T_1 - T\theta)(10.05)10^{-6} - (13")(T_2 - T_0)(6.5)10^{-6}$

$\quad = 0.137 - 0.0304 = + \underline{0.1066"}$ SS > U gap

<u>Gap 23</u> Assume $T_2 = T_3$ for min. clearance

$dr = (9.5")(T_2 - T_0)(6.5)10^{-6} - (9.5")(T_3 - T_0)(9.6)10^{-6}$

$\quad = 0.0222 - 0.328 = - \underline{0.0106}$ U < SS interference

Gap 34 Assume $T_4 = 370°$ from 2322 Btu/hr fuel rate

$dr = (8-7/8")(T_3 - T_0)(9.6)10^{-6} -(8-13/16")(T_4 -T_0)(_q .5)10^{-6}$

$= 0.0306 - 0.0251 = + 0.0055$ gap

Length of U. Assume $T_2 = T_3$ for max. differential

$dl = (194-1/8)(T_1 -T_0)(10.05)10^{-6} -(194 1/8)(T_3-T_0)(6.5)10^{-6}$

$= 2.05 - 0.455 = 1.595"$ expansion SS $>$ U gap

Container shows gap.

Case 2

$T_1 = 220°$ $T_3 = 340°$ $T_0 = 70°$ $T_4 = 370°$

GAP 12 Assume $T_2 = T_3$ for min. clearance

$dr = (13")(T_1 - T_0)(9.4)10^{-6} -(13)(T_2 - T_0)(6.5)10^{-6}$

$= 0.0183 - 0.0228 = -0.0045$ U $>$ SS interference

Gap 23 Assume $T_2 = T_1$ for min. clearance

$dr = (9.5")(T_2 - T_0)(6.55)10^{-6} -(9.5)(T_3 - T_0)(9.47)10^{-6}$

$= 0.00925 - 0.0243 - 0.015$ U $<$ SS interference

Gap 34 Assume $T_4 = 370°$

$dr = (8-7/8")(T_3 - T_0)(8.47)10^{-6} -(8-13/16)(T_4 -T_0)(9.47)10^{-6}$

$= 0.0227 - 0.025 - 0.0023$ interference

Case 3   Cask at original dimen. $T_4 = 241°$ $T_0 = 70°$ $T_3 = 70°$

Gap 34 $dr = 0 - (8-13/16)(T_4 - T_0)(9.7)10^{-6}$

$= - 0.0367"$ container increase - interference

Length Container

$$dl=(187-5/8)(T_4-T_0)\ (9.7)10^{-6}$$

$=-0.309$ container increase interference

Case 4

$T_1 =70^{0}$ T_2 assumed $= T_3 = 450^{\circ}$ $T_0 = 70^{\circ}$ $T_4 = 240^{\circ}$

Gap 12 $= 0\ -(13)(T_2-T_0)\ (6.5)10^{-6}$

$= 0 - 0.0322 = -.0322$ SS $<$ U interference

Gap 23 $= 0$

Gap 34 $=$ negligible

Length Container – in time same as case 3

Case 5

$T_1=T_2=T_3 = 40^{\circ}$ $T_0 = +70$

Gap 12 $dr = (13)(T_1 - T_0)(9.11)10^{-6}\ -(13)(T_1-T_0)(6.5)10^{-6}$

$= -0.013 + 0.0093 = 0.0037$ SS $<$ U interference

Gap 23 $dr = (9.5)(T_2 - T_0)(6.5)10^{-6}\ -(9.5)(T_3-T_0)(9.11)10^{-6}$

$= -0.0068 = 0.0095 = + 0.0027$ gap

Case 6

$T_1=T_2 = -40^{\circ}$ $T_3=T_4 = 240^{\circ}$ $T_0 = 70^{\circ}$

Gap 23 $dr = (9.5)(T_2 - T_0)(6.5)10^{-6}\ -(9.5)(T_3-T_0)(9.11)10^{-6}$

$= -0.0068 - 0.0147 = -0.0218$ interference

2.6.1.6 <u>Analysis of Uranium Shielding in Cask Closure</u>

<u>Case 1</u>

Assume same gradients as given for cask.

$T_1 = 1120°$ $T_2 = 340°$ $T_0 = 70°$

$\Delta t_s = 1050°$ for SS space expansion

$\Delta t_u = 430-70 = 360°$ for U expansion
      from original machined dimension

<u>Length</u> dl $= (16") [\Delta t_s 10.05 - \Delta t_u 6.5]10^{-6}$

$= 16 [1050(10.05) - 360 (6.5)]10^{-6}$

$= 16 (11000 - 2340)10^{-6} = 0.1384$ gap

Thickness – also negligible

$dt = \dfrac{2-1/4}{16} (0.1384) = 0.0195$ gap

<u>Case 2</u>

$T_1 = 220°$ $T_3 = 340°$ $T_0 = 70°$

$\Delta t_s = 150°$

$\Delta t_u = 340 - 70 = 270°$

<u>Length</u> dl $= 16 [\Delta t_s (9.47) - \Delta t_s (6.5)]10^{-6}$

$= 16 [150 (9.47) - 270 (6.5)] 10^{-6}$

$= 16 (1420 \text{ - } 1750)10^{-6}$

$$= 5280 \ (10^{-6}) = - \underline{0.00528"}$$

$$\underline{\text{Thickness}} \ dw - 0.00528 \times \frac{2\text{-}1/4"}{16} = - \underline{0.00075 \text{ interference}}$$

Case 3

No diff. exp.

Case 4

No diff exp.

Case 5

Temp. drop = -40 - (+70) = -110°

$\underline{\text{Length}}$ dl = 16 (-110) (9.11 - 6.5)10^{-6}

$$= -4600 \ (10^{-6}) = 0.0046 \text{ interference}$$

Case 6

No diff exp.

$\underline{\text{Summary all cases}}$. For disc. max. interference is only 0.00528 in. assuming metal to metal fit at 70°.

Manufacturing clearances would be greater than this for the uranium disc in the closure head.

2.6.2 $\underline{\text{Vibration}}$

This cask is designed for transport by trailer only. Therefore, the only concern is that the fundamental frequency of vibration for the cask as a simply supported beam, loaded by its own weight, be appreciably higher than the repeatable impulse frequencies for the trailer itself.

The cask is considered to have a total moment of inertia (I) equal to the sum of the individual I values of the two shells and the uranium cylinder.

$$I_{outer\ shell} = \pi/4\ (14^4 - 13^4) \quad = 7{,}750 \text{ in.}^4$$

$$I_{inner\ shell} = \pi/4\ (9.5^4 - 8.875^4) = 1{,}530$$

$$I_{uranium} = \pi/4\ (13^4 - 9.5^4) \quad = \underline{16{,}041}$$

$$I_{total} = 25{,}321 \text{ in.}^4$$

Total weight of cask and contents = $\underline{46{,}500 \text{ lb}}$
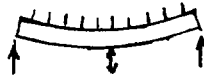
$\ell = 208$ in.

$$\text{Frequency} = \frac{3.55}{\sqrt{\dfrac{5}{384}\dfrac{W\ell^3}{EI}}} = \frac{3.55}{\sqrt{\dfrac{5}{384}\dfrac{(46{,}500)(208)^3}{29 \times 10^6\ (25{,}321)}}}$$

$$= \underline{41.2} \quad \text{cycles per second (cps)}$$

Fundamental frequencies developed in trailers are generally in the range of 4 to 16 cps (see "Shock and Vibration Handbook, Vol. 3, Sect. 45).

2.6.3 <u>Internal Pressure</u>   50 psig Roark <u>XIII</u> - Case 1

See section 2.5.2 for inner container description

Hoop Stress

$$S_2 = \frac{PR}{t} = \frac{50(8.62)}{1/2} = 872 \text{ psi} \quad \text{ok, less than 30,000 psi}$$

Meridional Stress

$$S_1 = \frac{PR}{2t} = \frac{862}{2} = 481 \text{ psi}$$

Radial Displacement

$$= \frac{R}{E}(S_2 - \nu S_1)$$

$$= \frac{8.62}{30(10^6)}[862 - 0.3(431)z] = \frac{6300}{30(10^6)} = 0.00021 \text{ inches} \quad \text{ok}$$

2.7 HYPOTHETICAL ACCIDENT CONDITIONS

2.7.1 Free Drop

The dynamics of the 30-foot free fall requires that a value be found for the maximum stress at impact in order to use structural analysis methods based on statics.

Literature on the subject of dynamic stresses is largely theoretical and seldom of engineering application value. The complexity can be reduced by limiting the problem to (1) compressive stresses on flat impact (2) at 44 ft/sec and (3) to steel, aluminum and uranium materials.

The most promising engineering formula is that for dynamic compression of rigid-plastic cylinders and is the one used herein. It is well authenticated by:

(1) Goldsmith: Impact - eq. 5.97 page 191

(2) Cristensen: Dynamic plasticity - eq 7.8 page 55

$$\rho v_1{}^2 = \epsilon_1 \ (P_1 - P_y)$$

$$\rho = \frac{lb/in^3}{386} \qquad v_1{}^2 = [44 \ ft/sec \times 12]^2 = 278,784 \ (in/sec)^2$$

where P_1 = initial, and maximum stress corresponding to moment of impact,

ϵ_1 = max. strain corresponding to P_1, and

P_y = static compressive yield point (log 0.002 in/in strain method)

Values of ρ and ρv_1^2

Aluminum	ρ = 0.097/386 = 0.000251	ρv_1^2 = 70
Steel (incl SS)	ρ = 0.290/386 = 0.0007512	= 209
Copper	ρ = 0.332/386 = 0.00086	= 240
Lead	ρ = 0.41/306 = 0.001062	= 296
Uranium	ρ = 0.683/386 = 0.001769	= 493

Typical Dynamic Properties

Material	Static Curve	P_y 0.002 offset	P_1	ϵ_1	P_1-P_y	$\epsilon_1 (R-P_y)$	$\dfrac{P_1-P_y}{P_y}$
Uranium Cast	NLC (Computed)	73,000	90,000	0.029	17,000	493.	0.233
302 SS	Goldsmith Fig. 250	41,350 F_{cy} = 35,000 (304SS)	53,800	0.01675	12,450	208	0.301

$$P_1 = 40,000 + \frac{825,000}{\varepsilon}$$

$$\varepsilon_1 = \frac{P_1 - 40,000}{825,000}$$

| 0.24 Carbon | Goldsmith Fig. 372 | 35,000 | 45,000 | 0.020 | 10,000 | 200 | |
| Ni-Cr Stl | Goldsmith Fig. 118 | 220,000 | 228.000 | 0.024 | 8,000 | 192 | 0.036 |

2.7.1.1 <u>End Drop</u>. The dynamic analysis required for the cask and its parts, in the flat drop attitude, can be based on the concept of simple cylinders which behave as solid rods impacting squarely upon a rigid surface at the velocity of the specified free fall distance of 30 feet. This velocity is 44 ft/sec.

Values for stress, strain, K-E and G loadings can be derived for the individual masses and for the cask as a whole.

MODEL FOR ANALYSIS

S_1 = Outer S.S. shell

S_2 = Inner S.S. shell

U_5 = Uranium cylinder

These are considered separate cylinders

S_1 and S_2 weights include parts of
the head weight

S_3 = SS plate - acting as striking
plate under S_1, U_5 and S_2

S_4 = part of S.S. plate stressed by
impact from container, etc
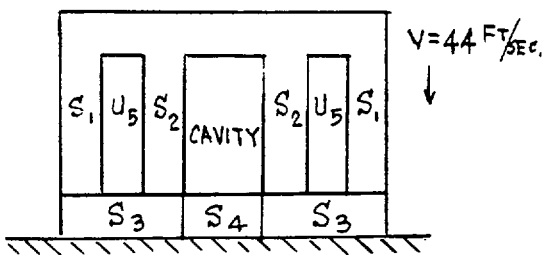
Area		Weight
S_1	86.3 in²	5,125 lb (incl. 635 lb head)
S_2	36.8 in²	2,260 lb (incl. 270 lb head)
U_5	250.9 in.²	32,400 lb
	374 in.²	39,785 lb
S_3	378.5 in.²	990 lb
S_4	248.5 in.²	5,249 lb (1,919 container, 1,800 contents)
	Total	46,024 lb

Initial and maximum stresses developed by striking a rigid mass are calculated for uranium and steel, derived from

$$\rho V_1{}^2 = \epsilon(P_1 - P_y) \text{ as previously shown}$$

$$P_1 \text{ uranium} = 90,000 \text{ psi}$$

$$P_1 \text{ stainless steel} = 53,800 \text{ psi}$$

For cylinders S_1, U_5 and S_2, the assumption of striking a rigid mass is conservative. Actually, they strike against an intermediate mass S_3, with some consequential reduction in stress.

S_3, on its under-surface, does strike a rigid mass, and counts S_1 U_5 and S_2 only as added load, considered as an equivalent weight of steel, added to the height of S_3 as a cylinder.

Stress differentials at the interfaces with S_3 are considered localized and quickly find equilibrium, producing a local increase in stress intensity in S_3 to match a reduced stress in U_5.

Total force developed at impacting face F = P_1 (area)

For S_1 F_1 = 86.3 in² x 53,800 psi = 4,640,000 lb

S_2 F_2 = 36.8 in² x 53,800 psi = 1,975,000 lb

U_5 F_5 = 250.9 in² x 90,000 psi = 22,581,000 lb

G's Developed   G = F/wt

S_1 4,640,000/5125 =   905 G

S_2 1,975,000/2260 =   875 G

U_5 22,581,000/32,400 =   697 G

S_3 29,196,000/39,785 =   735 G for bottom of cask and closure head.

Strength of Weld - Closure Head Flange

Weight   Uranium block 330 lb

SS enclosure   30

360 lb
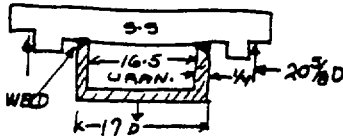
Area of weld = area of shell

= π/4 (17² - 16.5²) = 12.5 in.²

Force in weld = (360 lb) x (735 G) = 264,000 lb

Stress in weld = 264,000/12.5 = 21,100 psi

The full penetration weld has substantially the strength of the parent metal. Assume 80% of 30,000 psi yield strength for stainless steel = 24,000 psi.

$$= \frac{24,000}{21,100} - 1 = 13.7\%$$

Strength of Plate in Bending and Shear
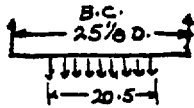
Weight = 850 lb - SS plate and uranium

Load = (860) x (735 G) = 632,000 lb

r_o = 10.25"

a = 12.56"

Roark table X

Stress at center of head, Case 2: t = 4 in. max

$$\text{Max } S_r = S_t = \frac{-3W}{2\pi\,mt^2}\left[m + (m+1)\,\log_e a/r_o - (m-1)\,\frac{r_o^2}{4a^2}\right]$$

$$= \frac{-3(632,000)}{628(10/3)4^2}\left\{10/3 + 13/3\,\log_e \frac{12.56}{10.25} - 7/3\,\frac{10.25^2}{4 \times 12.56^2}\right\}$$

$$= 5650\,[10/3 + 13/3\,(0.230) - 3.9] = 5650\,(0.31)$$

$$= \underline{1750\text{ psi}} \quad \text{ok} \quad 30,000 \text{ psi yield strength}$$

Stress at edge, Case 7: t = 2-5/8 in.

$$\text{Max } S_r = \frac{3W}{2\pi t^2}\left[1 - \frac{r_o^2}{2a^2}\right] = \frac{3(632,000)}{628(2.625)^2}\left[1 - \frac{10.25^2}{2(12.56)^2}\right]$$

$$= 43,700\,(1 - 0.334) = \underline{29,200\text{ psi}} \quad \text{ok } 30,000 \text{ psi yield strength}$$

Shear at edge (20.625 diam) t = 2-5/8 in.

Area in shear $= 2\pi rt = \pi (20.625) 2.625$

$= 170.5$ in.^{2}

$$S_s = \frac{632,000}{170.5} = \underline{3700}$$

$F_{SU} = 0.60 (70,000 \text{ psi})$

$= 42,000$

$$M.S = \frac{42,000}{37,000} - 1 = 10.3$$

2.7.1.2 Side Drop
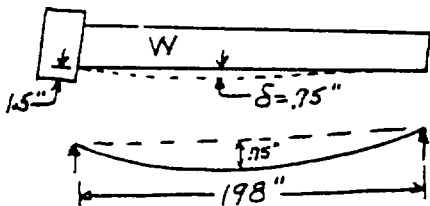
Deflection at mid-length is limited to 0.75 in. by contact with base

Find $(W \cdot g)$ to give $\delta = 0.75$ in.

$$\delta = \frac{5}{384} \frac{W\ell^3}{EI} = 0.75 \text{ in.}$$

$$\therefore (W \cdot g) = \frac{0.75 (384)}{5} \frac{EI}{\ell^3} = \frac{57.6 (29) 10^6 I}{(198)^3}$$

$(W \cdot g) = 215 I$

Let $W = 46,024$ lb = total weight of loaded cask.

Let $I = \Sigma I$ for outer shell, inner shell, and 1-1/2 in. of welded uranium. The rest of the 3-1/2 in. total thickness of uranium contributes mass and not stiffness. This is very conservative, since the middle cylinder spans the region of max stress and deflection. All three cylinders have the same deflections.

I_1 outer shell $= \dfrac{\pi}{4} (14^4 - 13^4) = 7750$ in.^{4} $\Sigma I/r = \dfrac{17,980}{14} = 1284$

I_2 inner shell $= \dfrac{\pi}{4} (9.5^4 - 8.875^4) = 1530$ in.^{4} $= \dfrac{17,980}{9.5} = 1895$

I_3 uranium $= \dfrac{\pi}{4} (13^4 - 11.5^4) = 8700$ in.$^4 = \dfrac{17,980}{13} = 1385$

$\Sigma I = 17,980$ in.^{4}

Now $(W \cdot g) = 215 \, (\Sigma I)$

$= 215 \, (17,980) = 3,880,000$ lb

$g = \dfrac{3,880,000}{46,024} = \underline{83.8 \, g}$

$M = \dfrac{(W \cdot g)\ell}{8} = \dfrac{3,880,000}{8} \, (198) = 95,900,000$ in. lb

and

S_1 outer shell $= \dfrac{95,900,000}{1284} = \underline{74,700 \text{ psi}}$

S_2 inner shell $= \dfrac{95,900,000}{1895} = \underline{50,600 \text{ psi}}$

S_3 uranium $= \dfrac{95,900,000}{1385} = \underline{69,200 \text{ psi}}$

All these stresses are momentary peaks and local S_3 for uranium is less than that sustained in the test.

The side drop test is considered satisfactory, with no permanent beam deformations - only local crushing deformations.

2.7.1.3 Corner Drop

Bottom Corner

In corner drop calculations the assumption is made that the ungula developed at the impact position represents the total compressive effects at a local point, rather than the actual nonlinear distribution of strain throughout the whole impacting mass, both elastic and plastic. It allows calculation of maximum stress values, etc., without regard for actual distribution of stress and strain at noncritical values.

$\alpha = \arctan \frac{12}{104} = .1152$

$= 6.57°$
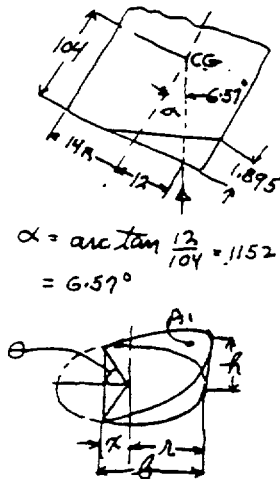
The calculated ungula is a greater volume than will actually result from impact, for this reason.

$$V_{(ungula)} = \frac{h}{b} r^3 \left[\sin \theta - \frac{\sin^3 \theta}{3} - \theta \cos\theta \right]$$

$\frac{h}{b} = 0.1152$ $r^3 = 14^3 = 2744$

$\therefore = (0.1152)2744[\ ] = 317[\ ]$

Assume center of impact 2 in. from edge, at R = 12 in.
Total weight impacting on SS bottom plate = 46024 lb
KE = (46024)(360 in.) = 16,600,000 in. lb
Mean flow stress for SS. (Approximately)

P = 53,800 psi

$$\frac{KE}{P} = \frac{16,600,000}{53,800} = 308 \text{ in.}^3$$

This is the calculated ungula volume

$$\therefore \text{Vol} = 317 \, [\;] = 308 \qquad [\;] = \frac{308}{317} = 0.972$$

Let $\theta = 100°$, then $[0.9848^3 - \dfrac{0.9848^3}{3} - (1.745)(-0.1736)] = 0.970$

and $\theta = 1.745$ radians

$$x = r\cos\theta = 14(0.1736) = \underline{2.43 \text{ in.}}$$

$$b = r + x = 14 + 2.43 = \underline{16.43 \text{ in.}}$$

$$h = b\,(0.1153) = 1.895 \text{ in.}$$

Contact area $A_1 = \text{approx. } \dfrac{\pi r^2}{2} + 2rx = \dfrac{615.75}{2} + 28\,(2.43)$

$$A_1 = 307.9 + 68 = \underline{375.9 \text{ in.}^2}$$

$$F_{max} = A_1 P_1 = (375.9)(53{,}800) = \underline{20{,}200{,}000 \text{ lb}}$$

$$F/wt = \frac{20{,}200{,}000}{46{,}024} = \underline{438 \text{ G}} \text{ vertically,}$$

<u>Component along axis</u>

$$(438 \text{ G})(\cos 6.57°) = 438\,(0.99343) = \underline{435 \text{ G}}$$

Top Corner   (Method similar to bottom impact analysis)

$$\alpha = \arctan \frac{13.4}{104} = 7.35°$$
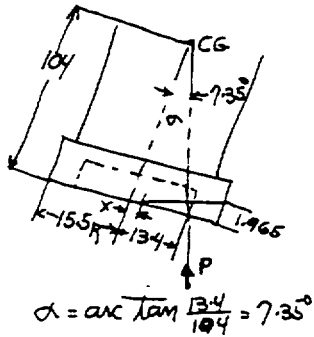
Assume center of impact at edge of closure, R = 13.4 in.

$$\frac{KE}{P_1} = 308 \text{ in.}^3 = \text{Volume of ungula}$$

$$V = 308 \text{ in.}^3 = \frac{h}{b} r^3 \left[\sin\theta - \frac{\sin^3\theta}{3} - \theta \cos\theta\right]$$

$$\frac{h}{b} = \frac{13.4}{104} = 0.129 \qquad r^3 = (15.5.)^3 = 3724$$

$$[\ ] = \frac{308}{0.129(3724)} = 0.641$$

Let θ = 89°, then [ ] = 0.99985 − 0.333 − (1.5533 × 0.0174) = 0.639

$$x = r \cos\theta = 15.5 \times 0.0174 = \underline{0.27 \text{ in.}}$$

$$b = 15.5 - 0.27 = \underline{15.23}$$

$$h = 0.129 \ b = \underline{1.965}$$

Contact area = approximately $\frac{\pi r^2}{2} - 2rx = \frac{754.8}{2} - 31(0.27)$

2-38

A_1 = 377.4 - 8.4 = 369 in.^{2}

F_{max} = $A_1 P_1$ = (369) 53,800 = 19,850,000 lb

= F/wT = 19,850,000/46,024 = 430 G vertically

Component along axis

(430G)(cos 7.35°) = 430 (0.99178) = 426.5G

BOLT LOADS

line of impact at P above
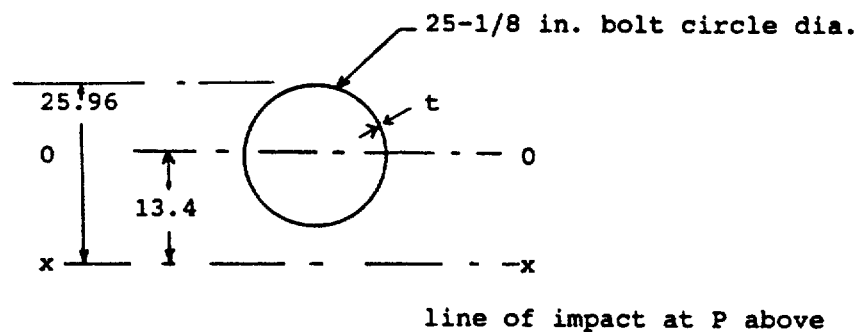
Let A = Tensile area of 24 bolts (1-1/4 - 7) = 24 x 0.9684 in.^{2}

= 23.24 in.^{2} assume this area as a thin ring with a bolt circle diameter

= 25-1/8 in.

Then $t = \dfrac{A}{2\pi r} = \dfrac{23.24}{\pi \times 25.125} = 0.294$

$I_{00} = \dfrac{\pi}{4}(R^4 - r^4) = \dfrac{\pi}{4}(12.71^4 - 12.42^4) = 1808$ in.^{4}

$I_{xx} = I_{00} + A(13.4)^2 = 1808 + 4173 = 5980$ in.^{4}

Weights 860 closure Moment about axis xx

1919 container $M = (4579 \text{ lb})(426.5 \text{ G}) \times 13.4 \text{ in.} =$

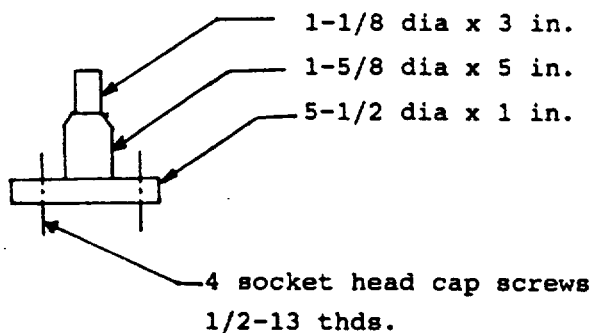<u>1800 contents</u> <u>26,169,443</u> in. lb

4579 lb Total

$$S_t = \frac{My}{I_{xx}} = \frac{26,169,433 \times 25.96}{5980} = \underline{113,605 \text{ psi}}$$

Bolts are heat-treated to 130,000 psi yield strength

$$\text{M.S.} = \frac{130,000}{113,605} - 1 = 14.4\%$$

Note that the actual <u>dynamic</u> strength is somewhat greater, therefore, the bolts will not yield under dynamic loading and the cask design provides the required integrity.

<u>Bottom Plug</u>

1-1/8 dia x 3 in. $\pi/4(1.125)^2 \times 3 = 3.0$ in.^{3}
1-5/8 dia x 5 in. $\pi/4(1.625)^2 \times 5 = 10.4$ in.^{3}
5-1/2 dia x 1 in. $\pi/4(5.5)^2 \times 1 = \underline{23.8}$ in.^{3}

37.2 in.^{3} x .284 =
<u>10.57 lbs</u>

4 socket head cap screws
1/2-13 thds.

Component of acceleration along axis of cask in corner drop is 435 G.

Impact force on drain equation is

$F = (W \cdot g) = (10.57) \, 435 = \underline{4600 \text{ lb}}$

Each socket head cap screw has a tensile yield strength of 2700 lb.

$$M.S. = \frac{4 \times 27000}{4600} - 1 = 23.5$$

2.7.2 Puncture

This cask has uranium as shielding between the inner and outer shells. Therefore, piercing of the outer shell with potential loss of shielding in the fire case cannot occur.

Instead, a shallow indent appears on the stainless steel outer shell where the edge of the 6-in. diameter pin partly cuts into it. This is clearly shown in report KY-546 by Clifford (Ref. 2.9). Actually, the pin is more damaged than the cask.

Energy is, however, absorbed in the three concentric shells of the cask in bending, with the possible formation of a plastic hinge.

The approximate distribution of moments among the shells can be made, assuming elastic deflections and an extreme fiber stress of 30,000 psi for both uranium and stainless steel inner and outer shells, with the same deflections, of course. Relative values follow;

34.2%	25,300 in. lb	1 in. SS outer shell
59.%	43,500 in. lb	3-1/2 in. uranium
6.8%	5,030 in. lb	5/8-in. inner shell
100%	73,830 in. lb	

Reference 2-10, pages 1302 thru 1307, gives the plastic region diagram in a beam when the midsection is just completely plastic.

For the uranium cylinder
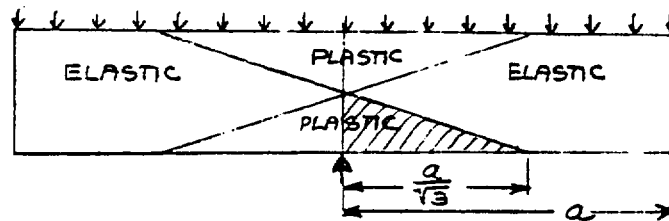
The plastic volume = 4 x volume of shaded ungula

$$= 4 \quad \left(\frac{2}{3}\right)(r_o^2 - r_i^2) \; H = 4\left(\frac{2}{3}\right)(13^2 - 9.5^2) \; 55$$

$$= 11,520 \text{ in.}^3$$

Each plastic cubic inch has been elongated 0.2% to reach a terminal stress of 30,000 psi (yield point for uranium).

The energy absorbed is

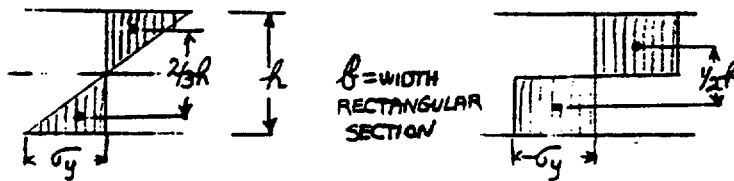$$U = \frac{30,000}{2} \; (11,520) \; (0.002) = \underline{345,600 \text{ in. lb}}$$

The total energy of the 40-in. fall is

$$U_{tot} = (46,500 \text{ lb}) \; 40 \text{ in.} = 1,860,000 \text{ in. lb}$$

The difference must be absorbed in bending of the "plastic hinge."

$$U_B = 1,860,000 - 345,600 = \underline{1,514,400 \text{ in. lb}}$$

Determination of plastic section modulus

Elastic

$$M_E = \frac{\sigma_y}{2} \times \frac{h}{2} \left(\frac{2}{3} h\right) b$$

$$= \sigma_y \frac{bh^2}{6}$$

Plastic

$$M_p = \sigma_y \times \frac{h}{2} \left(\frac{h}{2} b\right) \qquad \therefore M_p = \frac{3}{2} M_E$$

$$= \sigma_y \frac{bh^2}{4}$$

For the <u>circular</u> section of the uranium cylinder

$$I = \frac{\pi}{4} (r_o^4 - r_i^4) \qquad Z_E = \frac{\pi}{4} \frac{(r_o^4 - r_i^4)}{r_o}$$

and

$$Z_p = \frac{3}{2} \frac{\pi}{4} \frac{(r_o^4 - r_i^4)}{r_o}$$

$$= \frac{3\pi}{8} \frac{(13^4 - 9.5^4)}{13} = 1860$$

The plastic moment at the midplane is

$$M_p = \sigma_p \, Z_p$$

$$= (30{,}000)(1860) = 55{,}800{,}000 \text{ in. lb}$$

But

$$U_B = M\theta$$

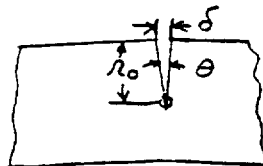
or

$$1{,}574{,}400 = 55{,}800{,}000 \, (\theta)$$

$$\theta = \frac{1{,}514{,}400}{55{,}800{,}000} = 0.0271 \text{ radians} = \underline{1.56°}$$

Estimated Actual Angle of Bend and Elongation

We are now justified in assuming that the actual conditions, showing loadings on all three shells, would reduce their common angle of bend to

$$(0.59 \, (1.56°) = 0.925° \, \theta$$

For the outer fibers of the outer shell this gives

$$r_o = 14 \quad \delta = r_o \theta = (14)(0.925)(0.01745) = 0.226 \text{ in.}$$

Consider this to be the measured elongation over a 2-in. gage length tension test piece.

$$\frac{0.226}{2} = 11.3\% \text{ elongation}$$

Stainless steel has <u>40%</u> elongation   <u>o.k.   No Rupture</u>

Uranium elongation 13/14 in. (11.3) = 10.5% elong. at R = 13 in.

10.5/14 in. (11.3) = 8.5% elong. at R = 10.5 in.

9.5/14 in. (11.3) = 7.7% elong. at R = 9.5

2.7.3   <u>Evaluation of FSV-1 Configuration C</u>

2.7.3.1   <u>Discussion</u>. The following sections provide the structural analyses used to verify that the burial liner cover with the 2 in. x 4 in. shield ring will remain in place during the hypothetical accident conditions. For these analyses the burial liner is part of configuration C and the unit is subjected to a 30-ft drop onto an essentially unyielding surface.

2.7.3.2   <u>Evaluation Criteria</u>. The allowable stress criteria set forth in NRC Regulatory Guide 7.6, "Design Criteria for Structural Analysis of Shipping Cask Containment Vessels" (Ref. 2-11) are specified for linear analysis only. Nonlinear analysis is acceptable and the acceptance criteria will be reviewed on a case-by-case basis by NRC staff.

Due to the anticipated nonlinear response of the structure during the hypothetical drop from 30 ft, a nonlinear, large deformation, large strain, finite element computer program was used to perform the analysis. The drop condition was evaluated by the criteria set forth in Appendix F, Section III of the ASME Boiler and Pressure Vessel Code for Level D Service. These limits have been established for extremely unlikely loadings for which the public

safety is to be assured; and are comparable to the hypothetical accident condition for cask design. The loading is assumed to be applied once and therefore, no consideration is required for fatigue.

When using inelastic component analysis, section F-1324.6 specifies a limit on the primary membrane stress (P_m) as defined in Subsection NB 3221 using a value for stress intensity of (S_m) equal to the greater of 0.7 S_u or S_y + 1/3 (S_u - S_y). S is the ultimate stress and S_y is the yield stress for the material. No correction factor α is allowed for evaluation of primary bending stresses. Therefore, the limit on primary local membrane plus primary bending: (P_l + P_b) is the same as for primary membrane (P_m).

A further degree of conservatism was added to the criteria by using the lower limit of S_y + 1/3 (S_u - S_y) to check the primary membrane stresses and the larger 0.7 S_u limit when checking primary membrane plus primary bending stresses.

The above criteria are conservative for dynamic impact since they are based on load controlled events for which margin is evaluated against a parameter such as internal pressure or applied load. The cask drop condition is an energy controlled event where the structure must absorb the kinetic energy of impact. Limits on stress restrict the allowable strain energy absorbed by the cask material to a small fraction of the energy which it is capable of absorbing.

2.7.3.3 Computer Program. The response of the burial liner cover and shield ring will be significantly nonlinear. This nonlinear response results from both material nonlinearity and nonlinearity caused by significant geometrical distortion from the initial configuration. Suitable programs for this type of analysis must incorporate these nonlinearities and be effectively coded if reasonable solutions are to be achieved.

HONDOII (Ref. 2-12) is the computer program used for the structural analysis which meets the above requirements. HONDOII has a wide selection of nonlinear material subroutines formulated to accommodate both large displacements and finite strains. It also has slide line capabilities which allow rigorous treatment of contact, sliding and release conditions between two or more bodies. Spatially, the program employs a four-node isoparametric quadralateral element. In time, the program performs the integration using a central difference time integrator. Since the time integration scheme is only conditionally stable with respect to time step size, the program continuously monitors the step size and adjusts it to keep the calculation stable. Initial conditions on velocity are allowed, and can correctly simulate the hypothetical drop condition.

The HONDOII Type 6 material model was used in the analysis. This model is formulated for finite strain elastic-plastic behavior with strain hardening. The strain hardening behavior may be isotropic, kinematic, or a combination of the two. Isotropic hardening was used since significant stress reversal is not anticipated and large strain formulations using kinematic hardening have recently been shown to give erroneous results under certain stress conditions (Ref. 2-13). The material model uses Cauchy or true stress and Signorine strain measure for finite deformations. Therefore, the program requires the input of the true stress vs. true strain. All comparisons of the computed stresses to the allowable stresses was done using true stresses to be mathematically and physically consistent.

2.7.3.4 Material Model Properties.

Type 304 Stainless Steel

The true stress vs. true strain curve used in the analysis was constructed from data taken from the ASME Code (Ref. 2-1) and the Aerospace Structural Metals Handbook (Ref. 2-14). The tabulated minimum yield strength

and ultimate strength at a representative temperature of 150°F were taken from the ASME Code Tables I.2.2 and I.3.2, respectively. The Aerospace Structural Metals Handbook was used to interpolate an engineering strain value at the maximum load in order to determine the true stress and true strain at the ultimate stress.

The following equations from EPRI report NP-1921 (Ref. 2-16) were used to convert engineering stress and strain values to true stress and true strain.

$$\sigma_{true} = \sigma_{engineering} \, (1 + \epsilon_{engineering})$$

$$\epsilon_{true} = \ln \, (1 + \epsilon_{engineering})$$

These relations are valid up to the maximum load in the uniaxial tension test.

The constructed stress vs. strain curve for stainless steel is shown in Fig. 2-2. Also shown on the figure is a typical true stress vs. true strain curve for type 304 stainless steel at room temperature taken from Ref. 2-15. The hardening slope of the two curves is essentially the same, however, the ASME code value for the ultimate stress and strain are significantly below typical values. This implies that for a given applied load the strains from a calculation using minimum properties will be larger than if nominal values had been used and a larger percentage of the allowable stress will be calculated. Thus, conservatism has been introduced by using the code values. The hardening modules (E_h) for the material is developed form the ASME code values.

Carbon Steel ASTM A36

The true stress vs. true strain curve for carbon steel (ASTM A36) was constructed in much the same way as those for stainless steel. The strain at
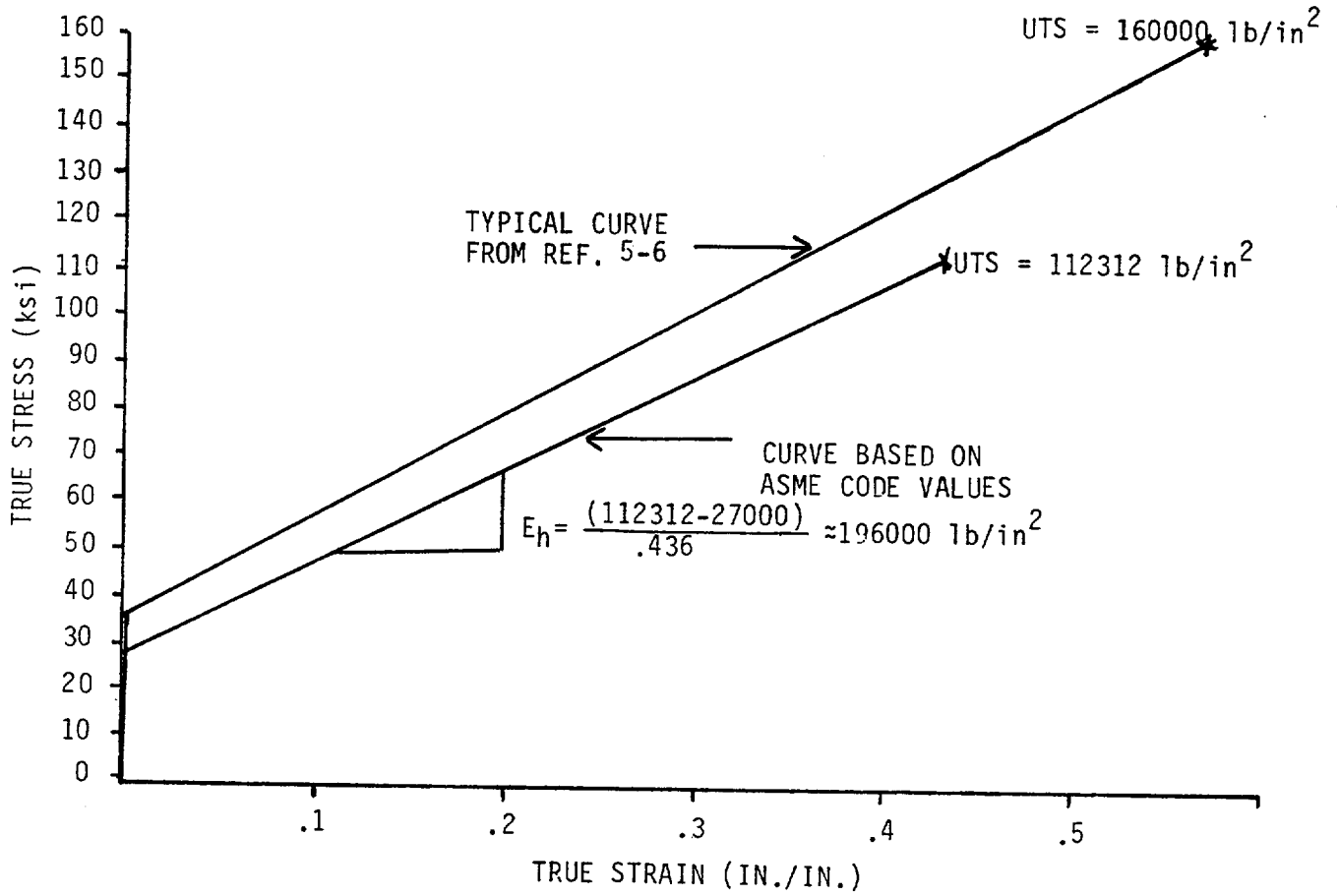
FIGURE 2-2   TRUE STRESS VS. TRUE STRAIN
FOR 304 STAINLESS STEEL

NC

maximum load was estimated to be 17.5% based on representative data for carbon steels. The true stress vs. true strain curve (Fig. 2-3) was constructed for the carbon steel flange and burial liner.

The burial liner does not have to survive the drop condition. It is included in the analysis only to apply an appropriate boundary condition to the burial liner cover plate during the 30-ft drop event.

The following table summarizes the material data used in the analysis and the reference source for each property.

Property	Value at 150°F	Reference
Carbon Steel ASTM A36		
Young's Modulus	27.8×10^2 lb/in.^{2}	ASME Code Table I-6.0
Poisson's Ratio	0.29	Mark's Handbook page 5-6
Mass Density	0.000732 lb-sec^2/in.^{4}	Mark's Handbook page 6-10
Yield Strength	34320 lb/in.^{2}	Figure 2-3
Hardening Modulus	224720 lb/in.^{2}	Figure 2-3
Stainless Steel 304		
Young's Modulus	27.9×10^6 lb/in.^{2}	ASME Code Table I-6.0
Poisson's Ratio	0.305	Mark's Handbook page 5-6
Mass Density	0.000749 lb-sec^2/in.^{4}	Mark's Handbook page 6-10
Yield Strength	27,000 lb/in.^{2}	Figure 2-2
Hardening Modulus	196,000 lb/in.^{2}	Figure 2-2

$$UTS = 70500 \text{ lb/in}^2$$

$$E_h \cong \frac{(70500-34320)}{.161} \approx 224700 \text{ lb/in}^2$$
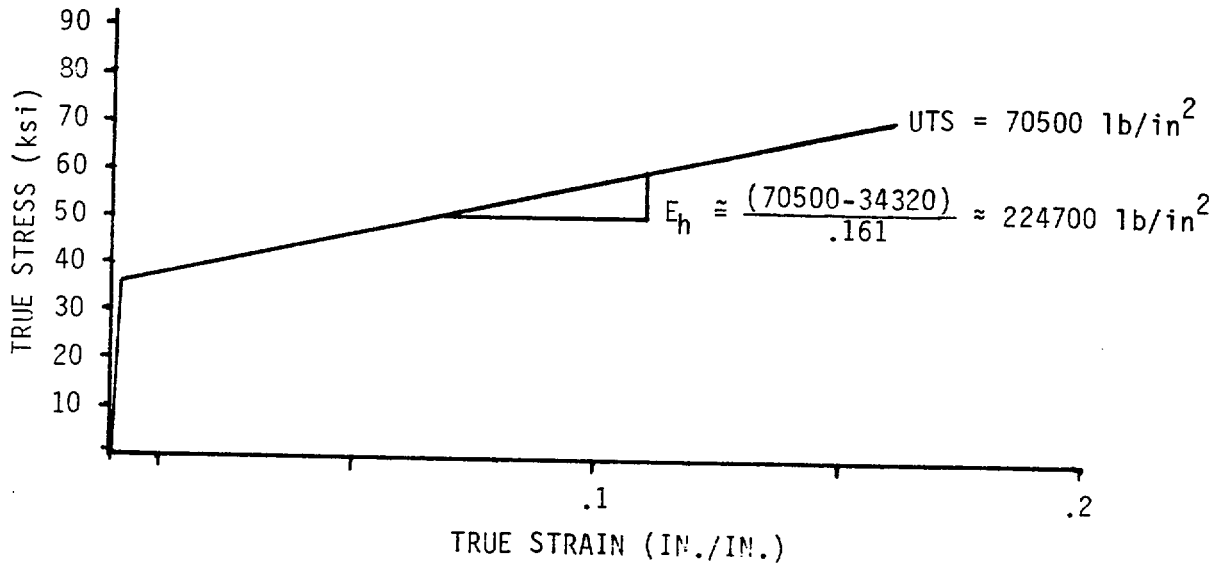
FIGURE 2-3 TRUE STRESS VS. TRUE STRAIN CURVE
FOR ASTM A-36 CARBON STEEL

2.7.3.5 <u>Finite Element Model</u>. The finite element model of the burial liner cover and shield ring assembly is shown in Fig. 2-4. Figure 2-4a shows the overall model and Fig. 2-4b shows a detail of the cover shield ring area. The 1/2-in. x 1/2-in. impact ring is attached to the 1/2-in. thick cover plate by 1/4-in. fillet welds. The 2 x 4 in. shield ring is attached to the cover plate by 1/4-in. goove welds with 1/4-in. fillet reinforcements. Slide lines are specified along the gaps between the rings and the cover plate. A slide line is specified between the bottom of the cover plate and the top of the burial liner flange. The bolts are not included in the analysis. A final slide line was used to represent the contact between the bottom of the liner flange and the support from the cask body. The sliding interfaces are shown in Fig. 2-5.

The support point of the cask was assumed to stop instantaneously at the time the stress wave reached the support flange, allowing the burial liner to be conservatively analyzed independently from the overall cask. A portion of the burial liner cylinder was included in the model to prevent reflections from the cut-off point from interfering with the solution in the cover plate and shield ring area. The impact is essentially over in the flange area in about 0.25 milliseconds. The stress wave will travel about 140 in. in this period of time. Ninety inches of burial liner was included in the model allowing greater than 0.85 milliseconds before the solution at the flange is perturbed.

In order to account for the slot in the cover plate, an equivalent axisymmetric model was constructed. The area of the slot was converted into an equivalent circular hole of equal area. The weight of the lifting bar was uniformly distributed over the central plate from a radius of 2.4 in. to 6 in. This approximation gives a reasonable representation of the behavior of the plate. It will be shown in the results section that there is considerable margin when the plate stresses are compared to the allowable stresses.

HONDO II MODEL
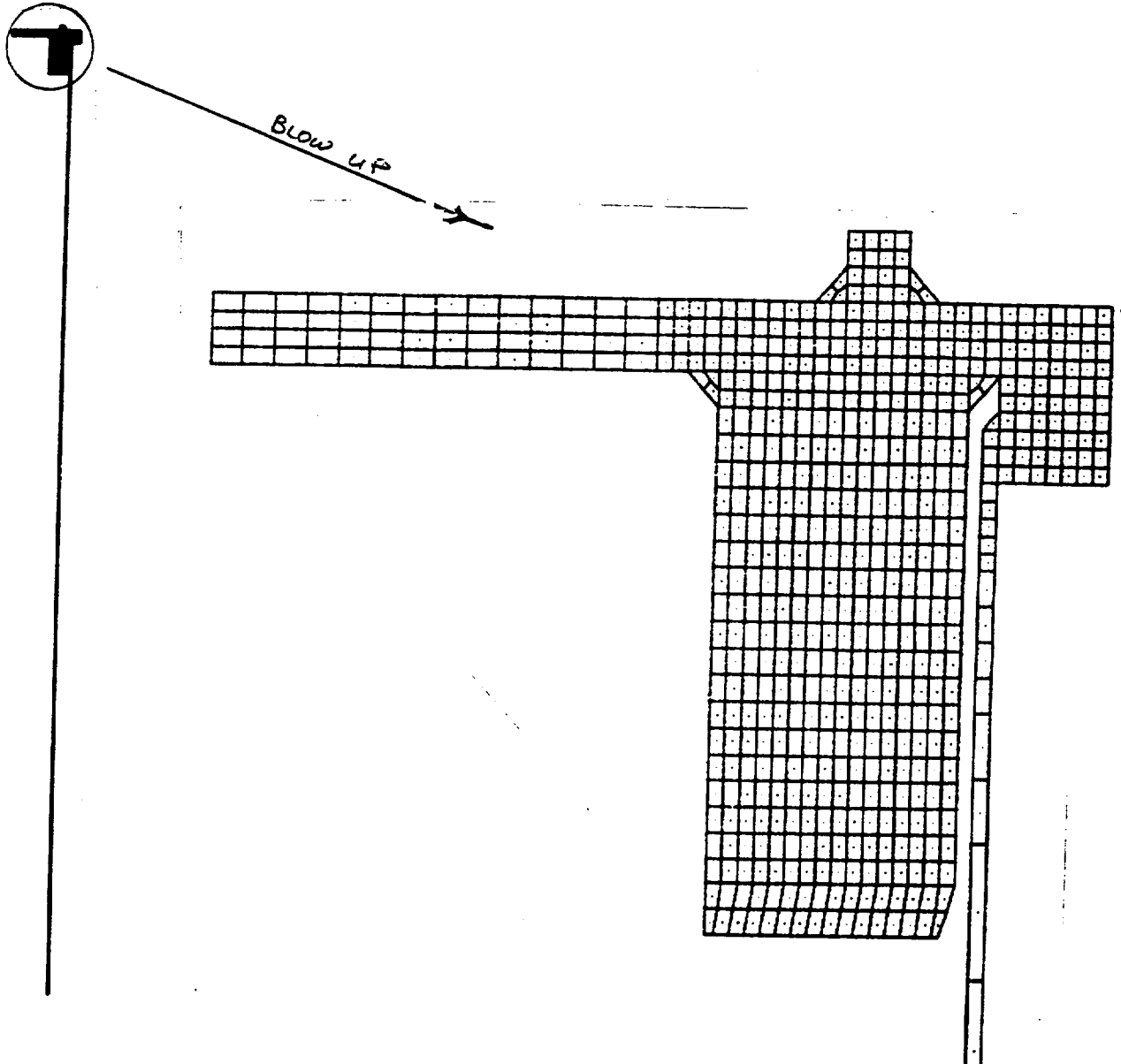
OVERALL MODEL

FIGURE 2-4a

BLOW UP AT COVER AND FLANGE

FIGURE 2-4b
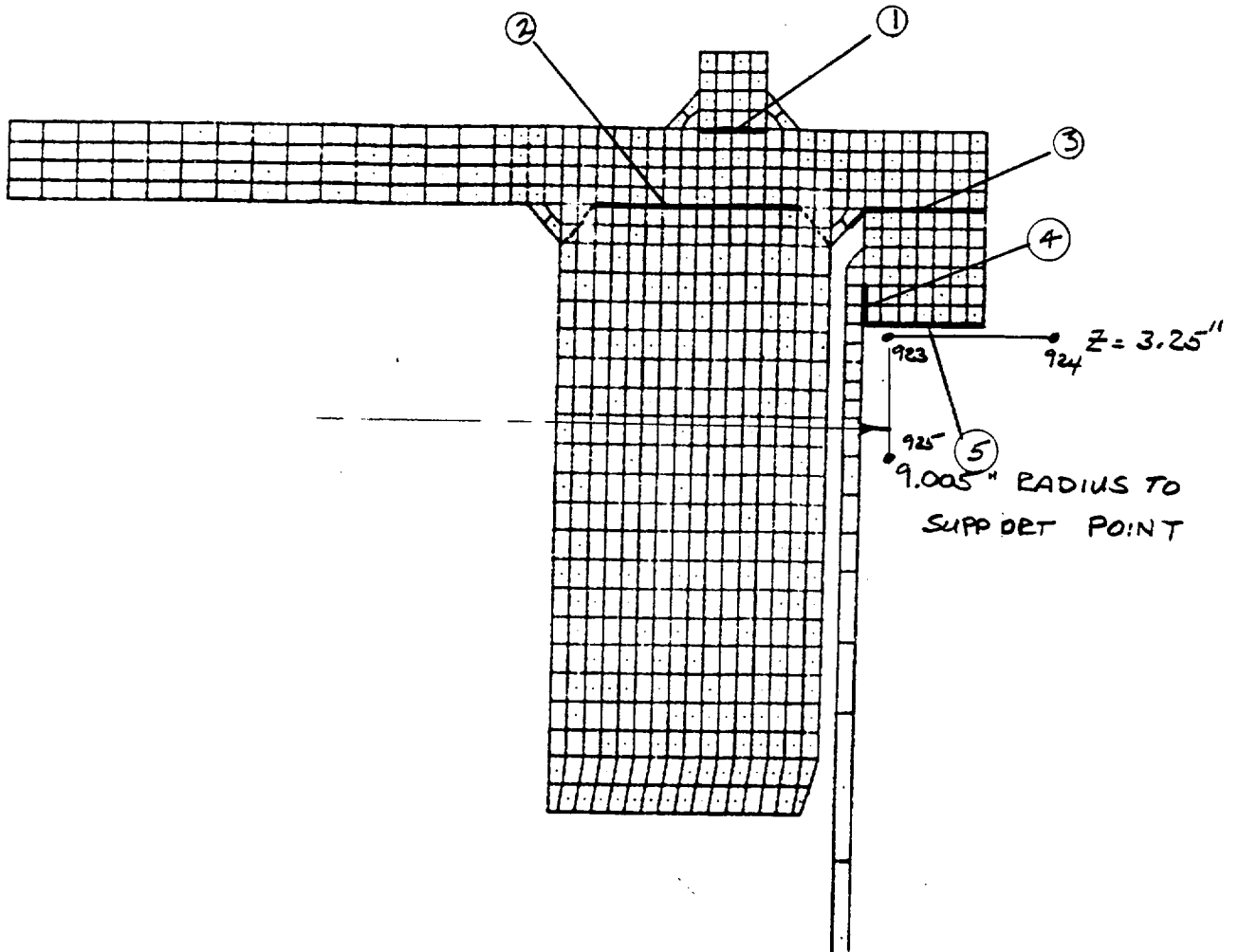
FIGURE 2-4 FINITE ELEMENT MODEL

FIGURE 2-5 SLIDING INTERFACES

2.7.3.6 <u>Results of Analysis</u>. The deformed shape of the cover plate and shield ring assembly is shown in Fig. 2-6 at 0.84 milliseconds. At this time, all of the nodes in the cover plate and shield ring have positive velicity and are rebounding. The top cover is separating from the burial liner flange and a gap is opening between the flange and the cover. The deformed shape plot allows a means of determining the critical areas in the model, i.e., those areas where the elements are highly distorted. These critical areas were confirmed by looking at time history plots of the effective stresses throughout the model. The critical areas which are examined in detail are shown in Fig. 2-7. This figure also shows the element numbers associated with each area. Figures 2-8 through 2-22 show the time history plots for the effective values of primary membrane stress and the primary membrane plus primary bending stress at the critical sections shown in Fig. 2-7.

The primary membrane stresses at the critical regions were determined by averaging the stresses through the thickness of the plate. The primary membrane plus primary bending stresses were extrapolated to the edge of the plate. This averaging was done on a stress component basis and then combined into an effective value using the equation:

$$\sigma_{eff} = \sqrt{\frac{(\sigma_r - \sigma_z)^2 + (\sigma_\theta - \sigma_z)^2 + (\sigma_r - \sigma_\theta)^2 + 6(\pi_{rz})^2}{2}}$$

Table 2-2 shows the primary membrane and the primary membrane plus primary bending stresses for the critical sections and the percentage of the allowable stress.

As may be seen from Table 2-2, the calculated stresses are significantly below the allowable stresses established for the critical sections. The cover plate and attached shield ring have been shown to satisfy criteria which
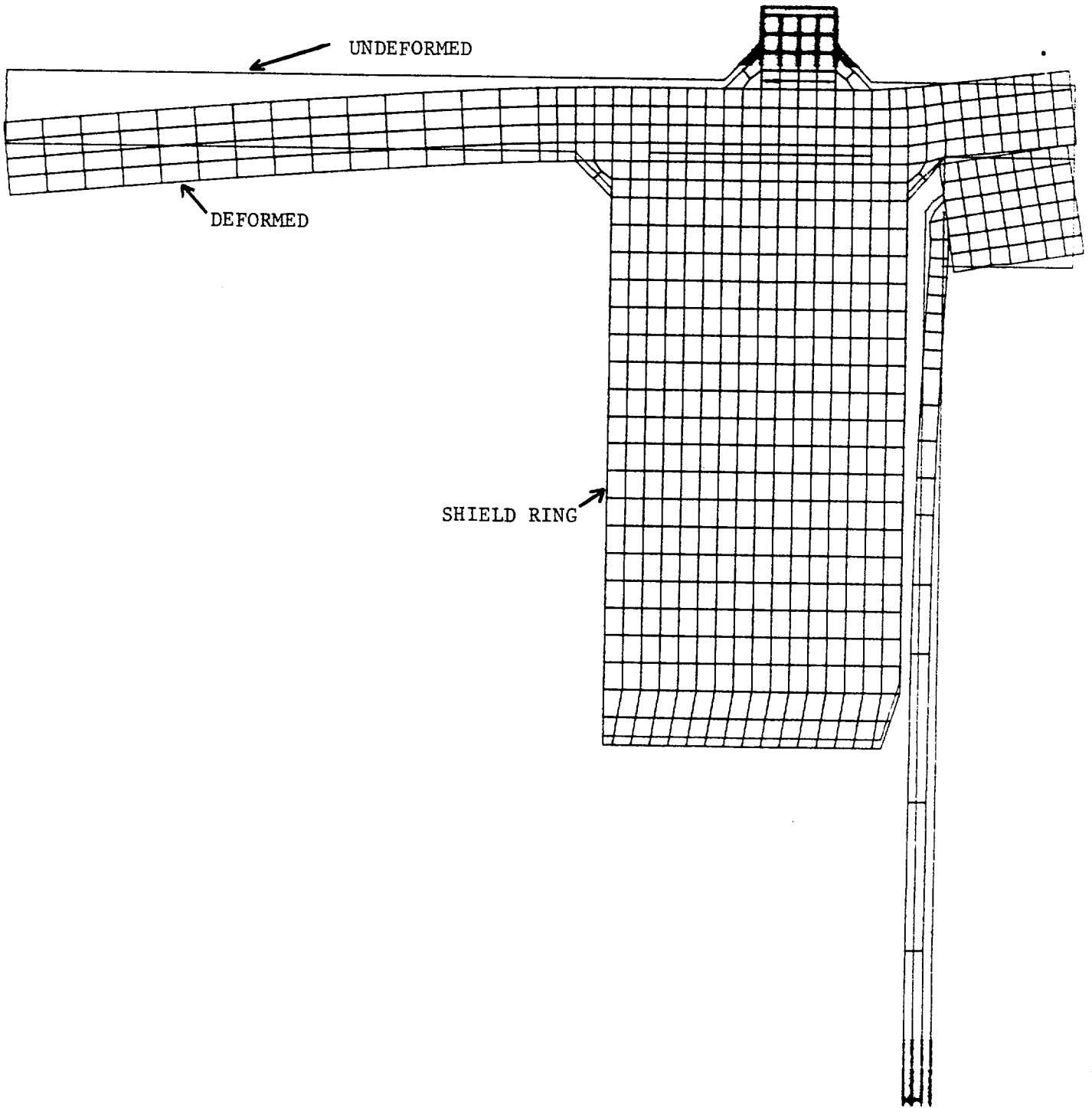
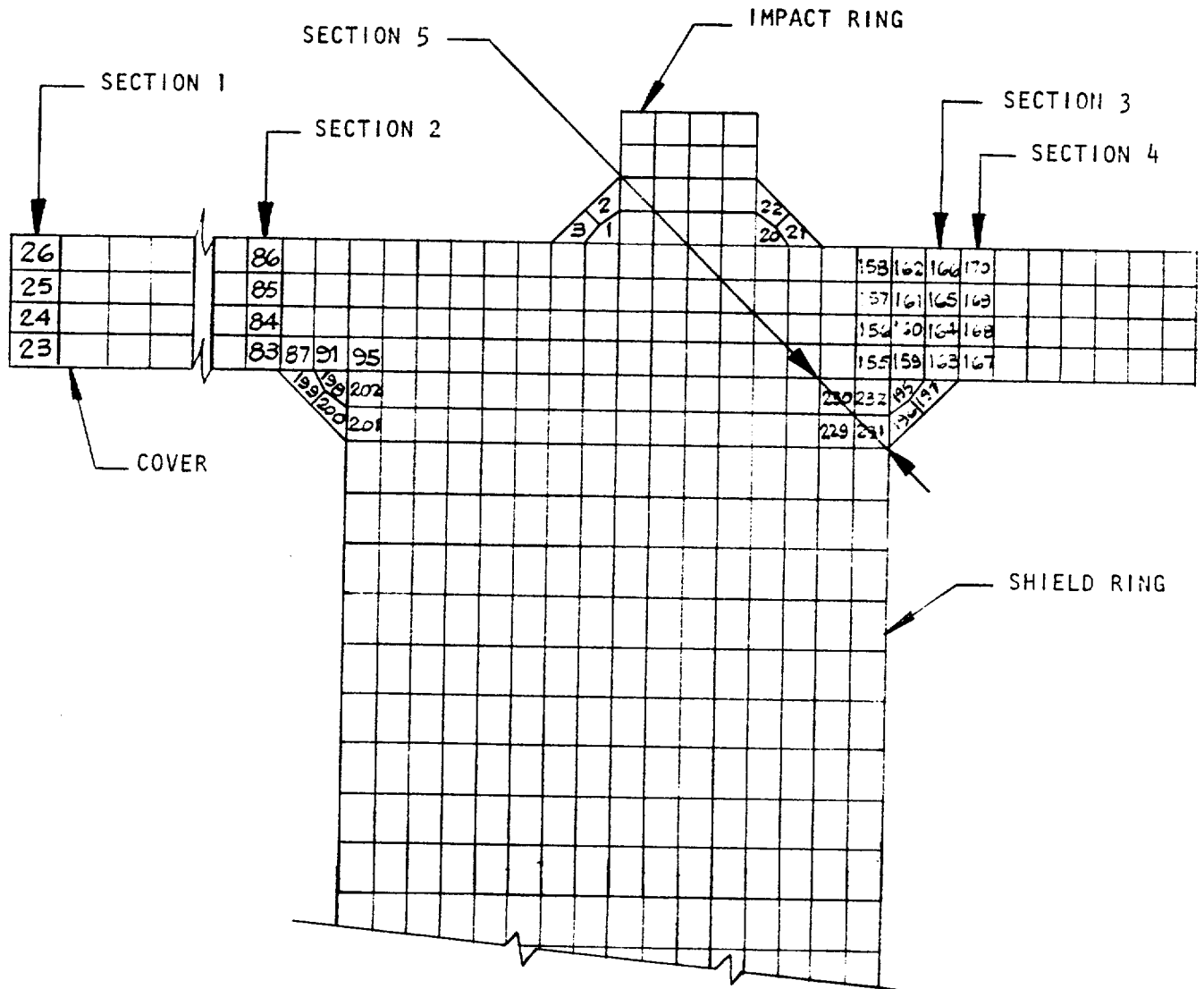FIGURE 2-6 DEFORMED SHAPE AT .84 MILLISECONDS

FIGURE 2-7 LOCATION OF SECTIONS USED
FOR PRIMARY STRESS EVALUATION

EFFECTIVE PRIMARY MEMBRANE STRESS AT SECTION 1
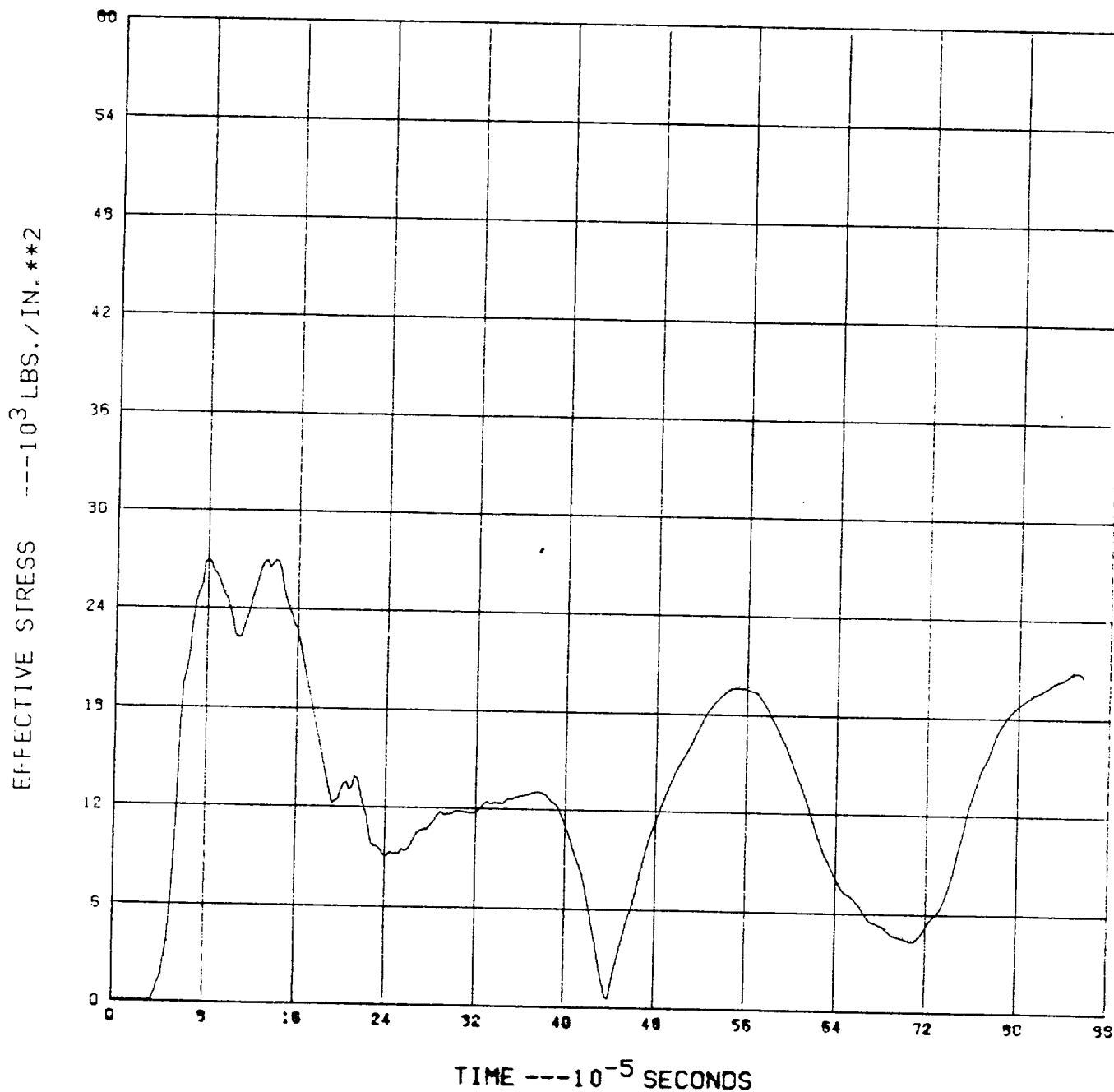

TIME ---10^{-5} SECONDS

FIGURE 2-8 SECTION 1
PRIMARY MEMBRANE STRESS

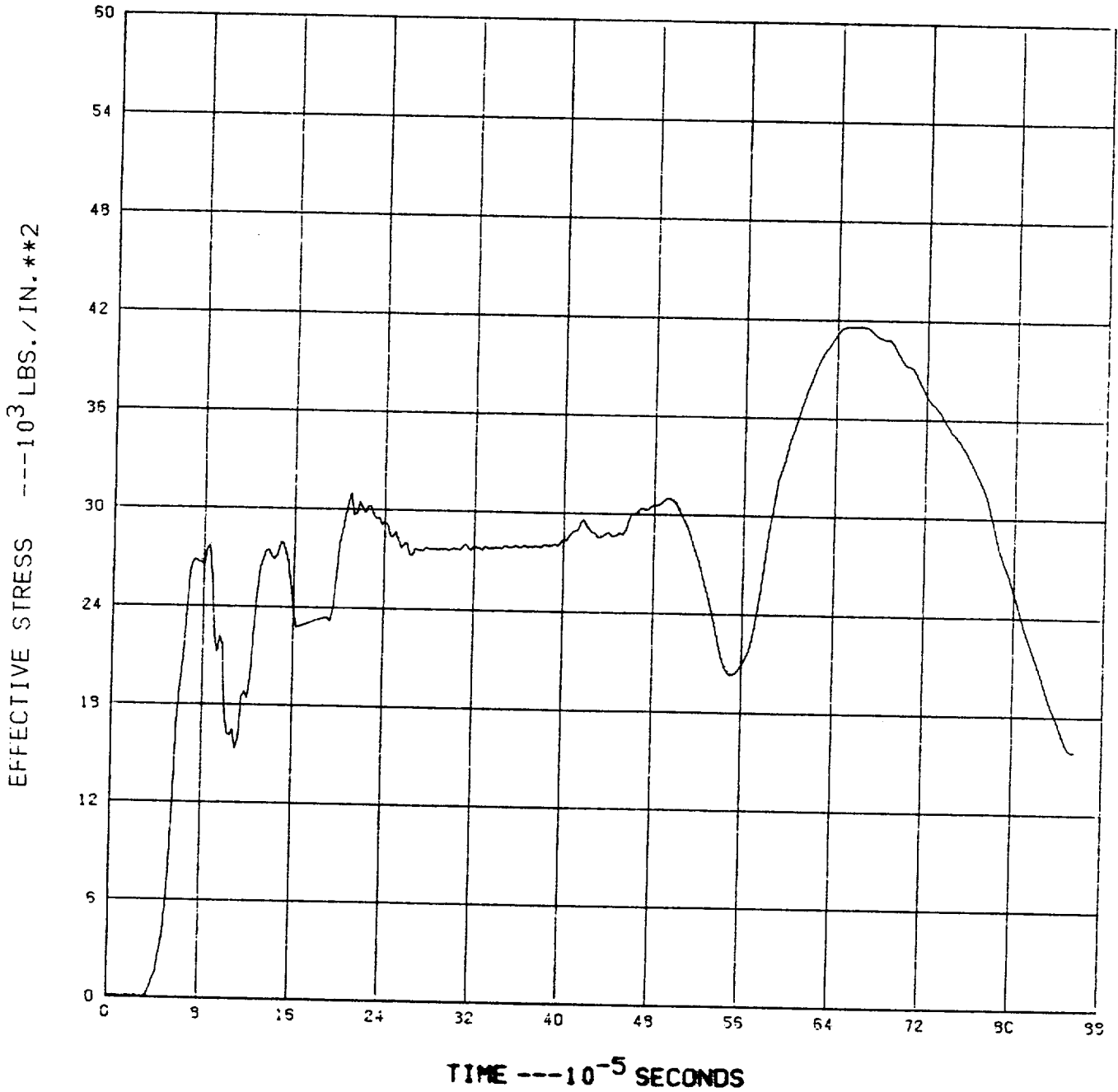## EFFECTIVE MEMBRANE + BENDING STRESS AT TOP OF SECTION 1

FIGURE 2-9 SECTION 1

PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT TOP OF PLATE
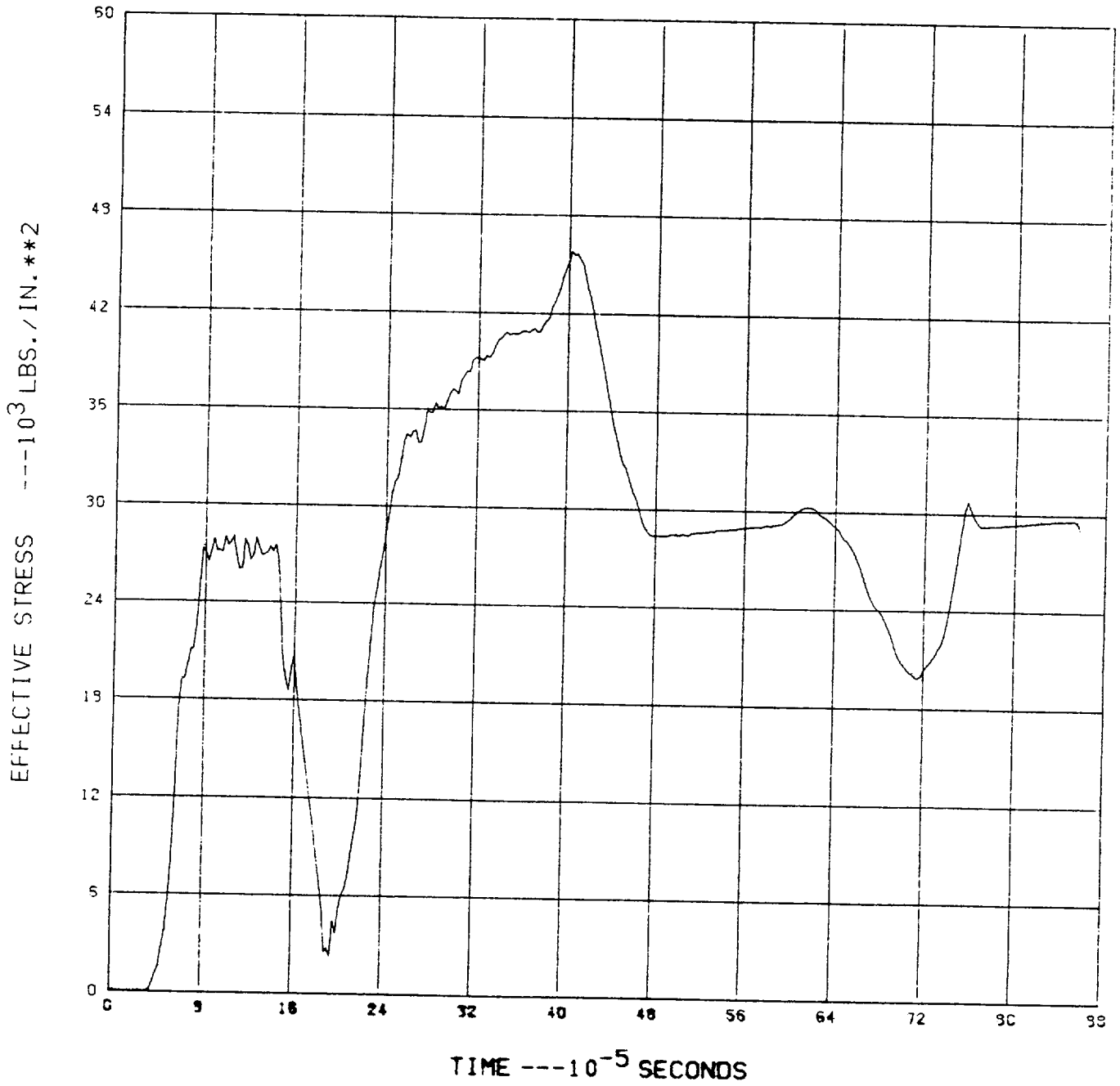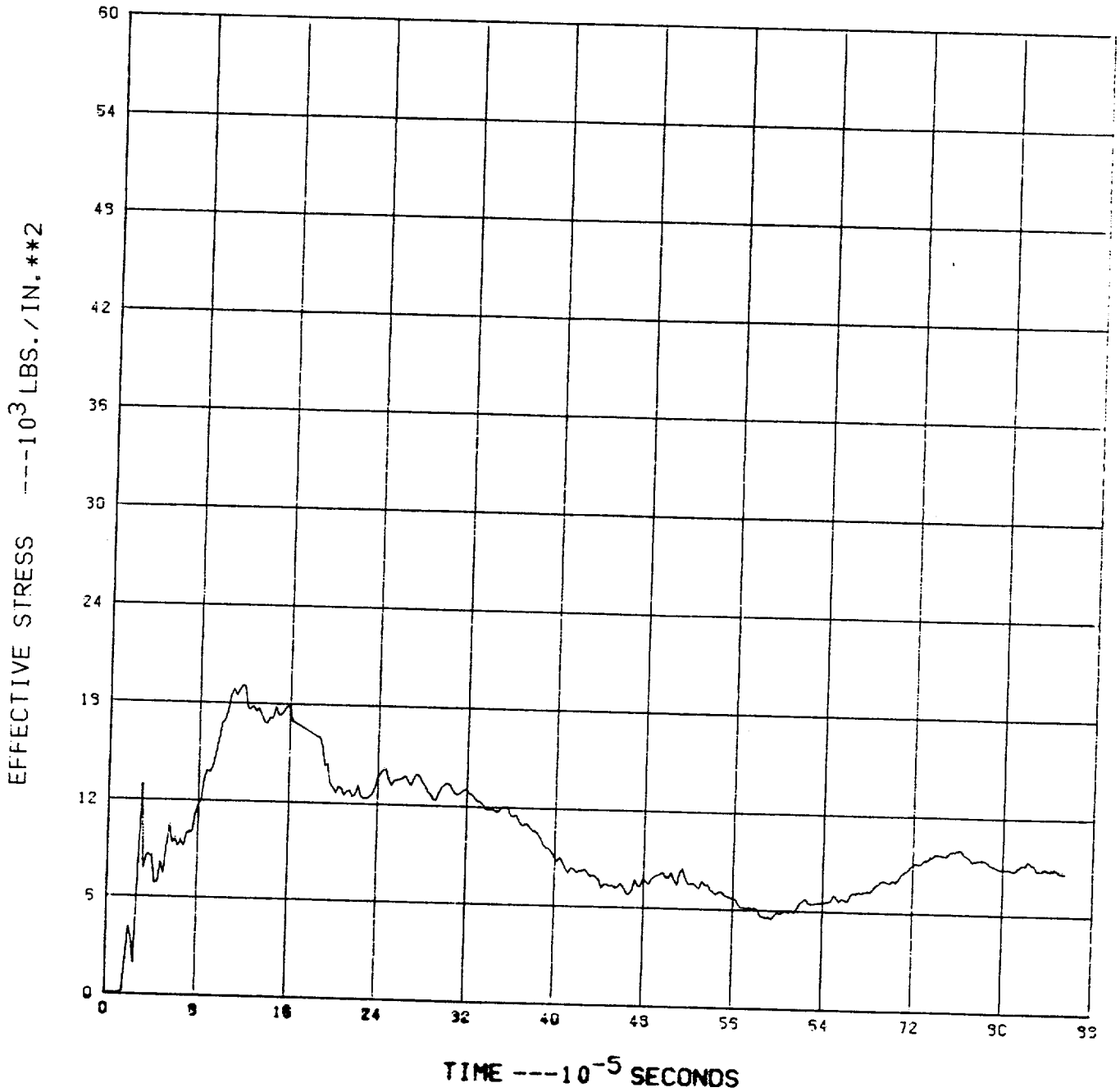
EFFECTIVE MEMBRANE + BENDING STRESS AT BOTTOM OF SECTION 1

TIME --- 10^{-5} SECONDS

FIGURE 2-10 SECTION 1

PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT BOTTOM OF PLATE

EFFECTIVE PRIMARY MEMBRANE STRESS AT SECTION 2

FIGURE 2-11 SECTION 2
PRIMARY MEMBRANE STRESS

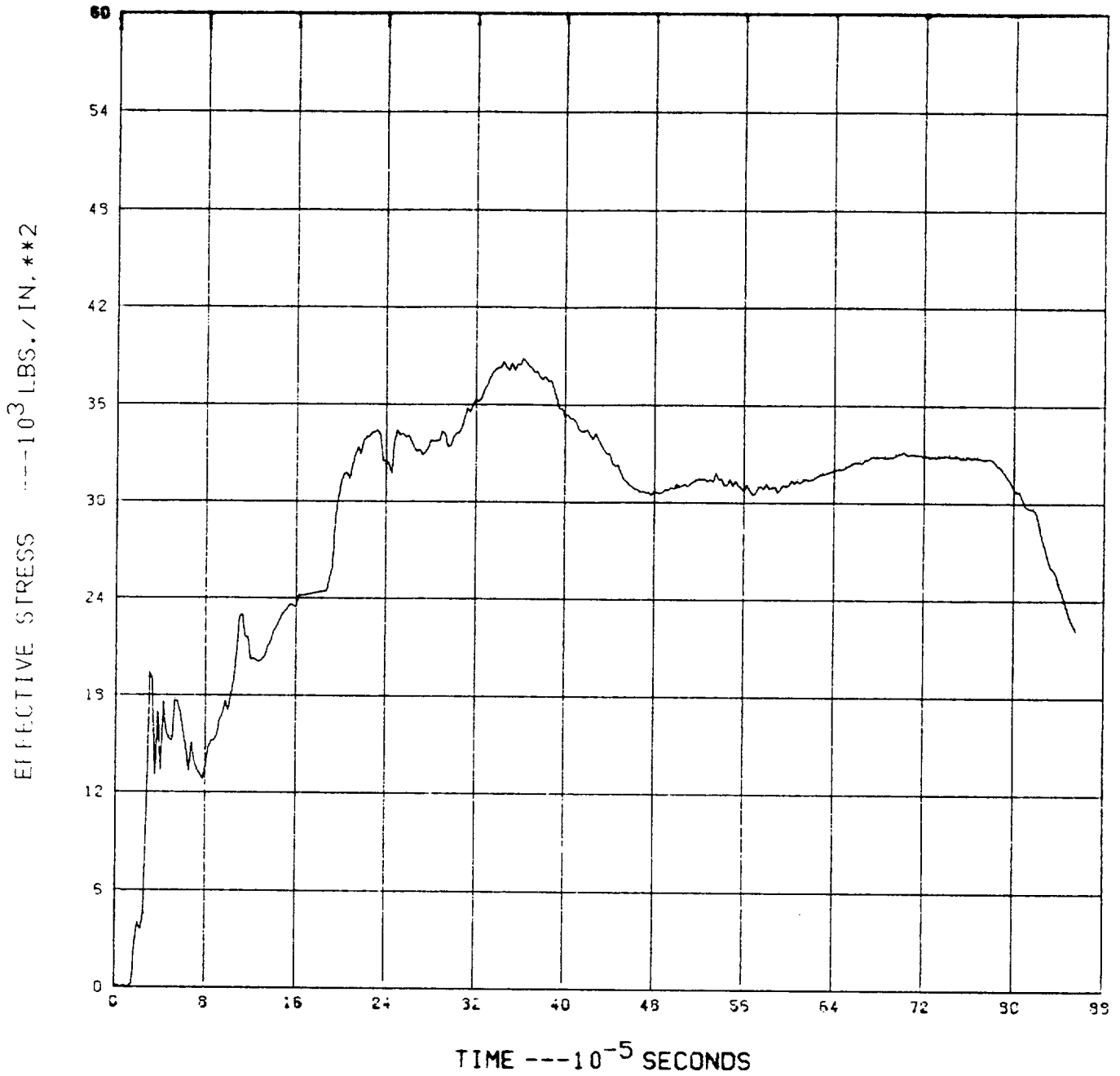## EFFECTIVE MEMBRANE + BENDING STRESS AT TOP OF SECTION 2

FIGURE 2-12 SECTION 2

PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT TOP OF PLATE

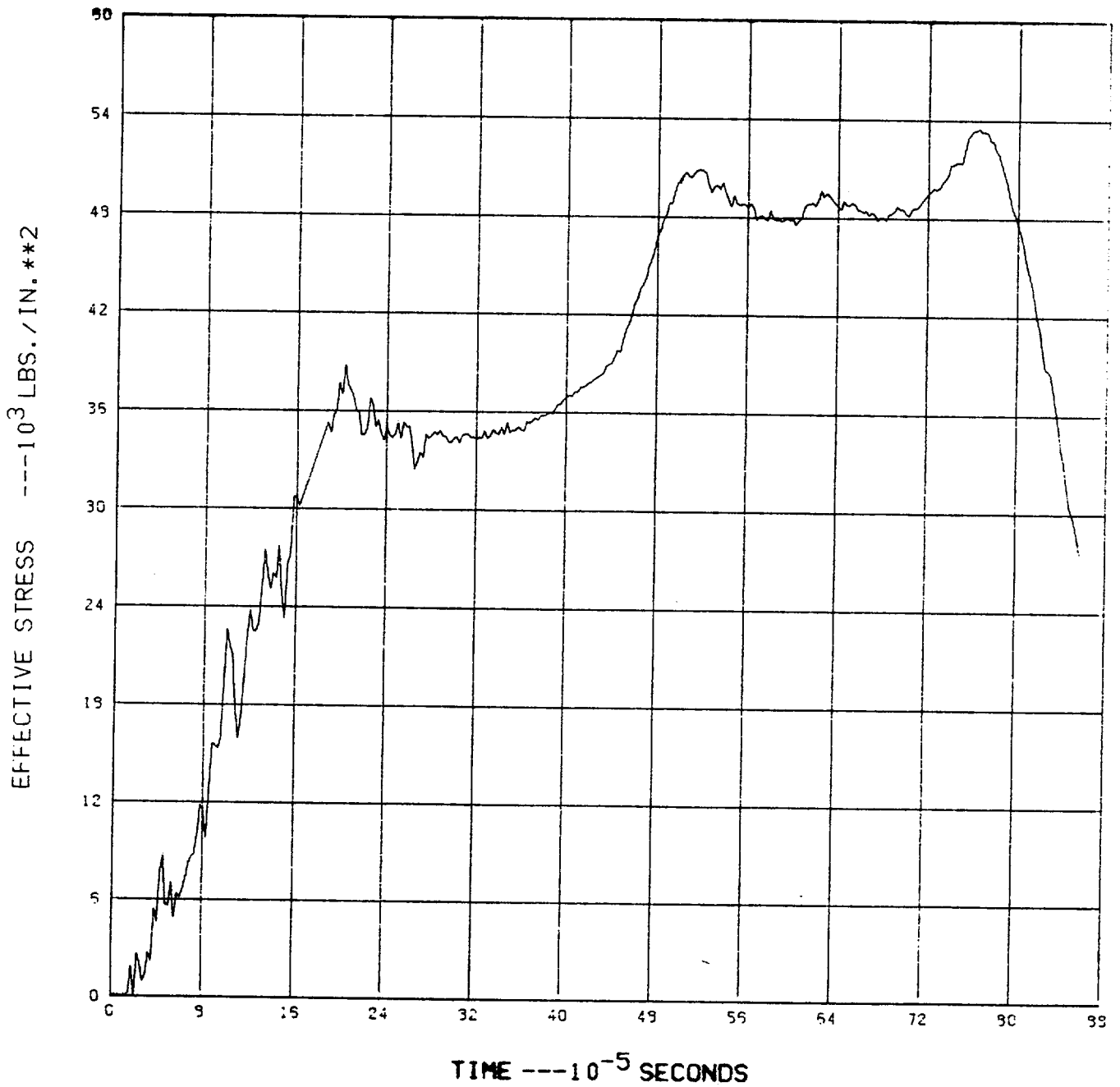EFFECTIVE MEMBRANE + BENDING STRESS AT BOTTOM OF SECTION 2

FIGURE 2-13 SECTION 2

PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT BOTTOM OF PLATE
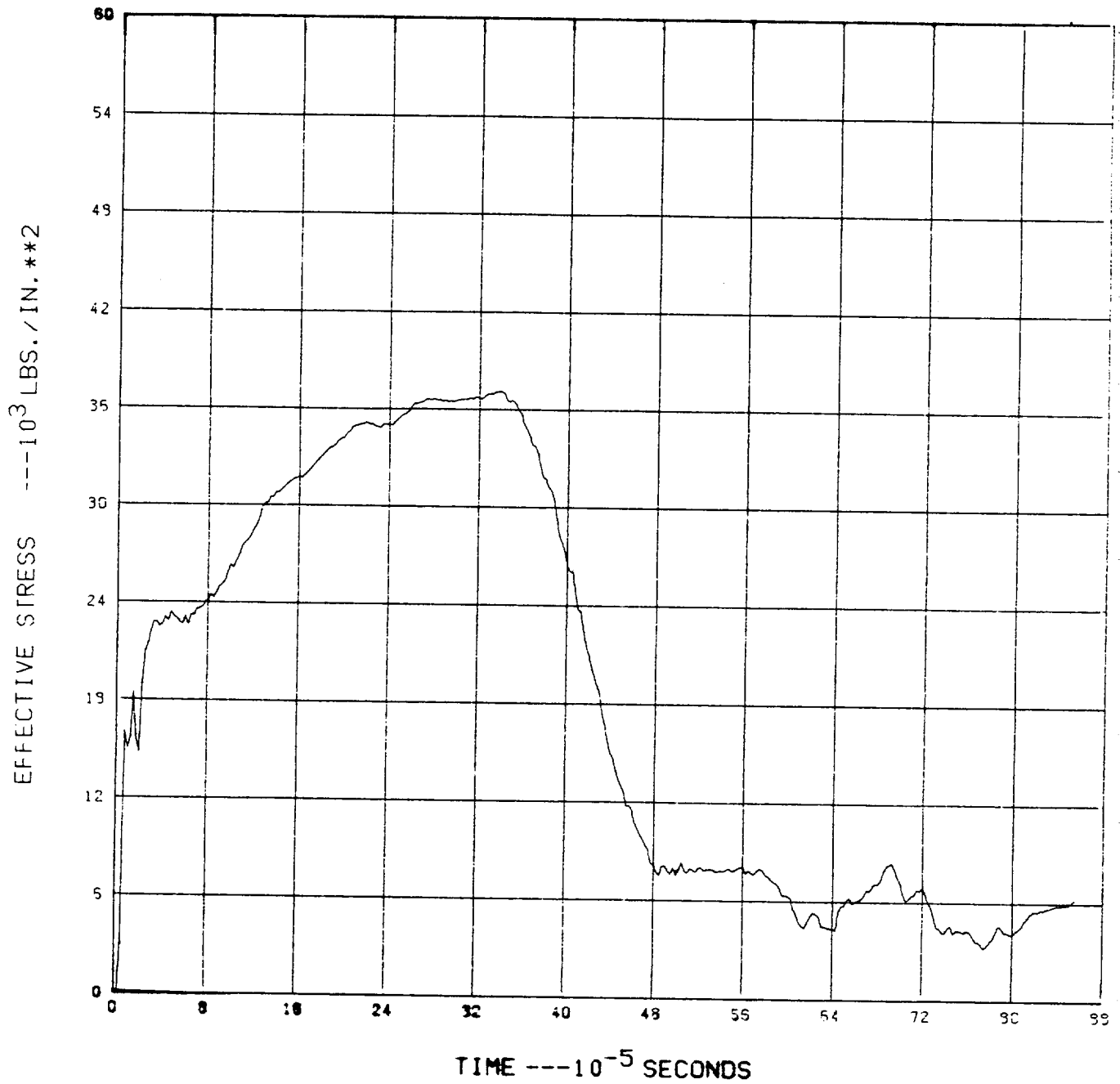
EFFECTIVE PRIMARY MEMBRANE STRESS AT SECTION 3

FIGURE 2-14 SECTION 3
PRIMARY MEMBRANE STRESS

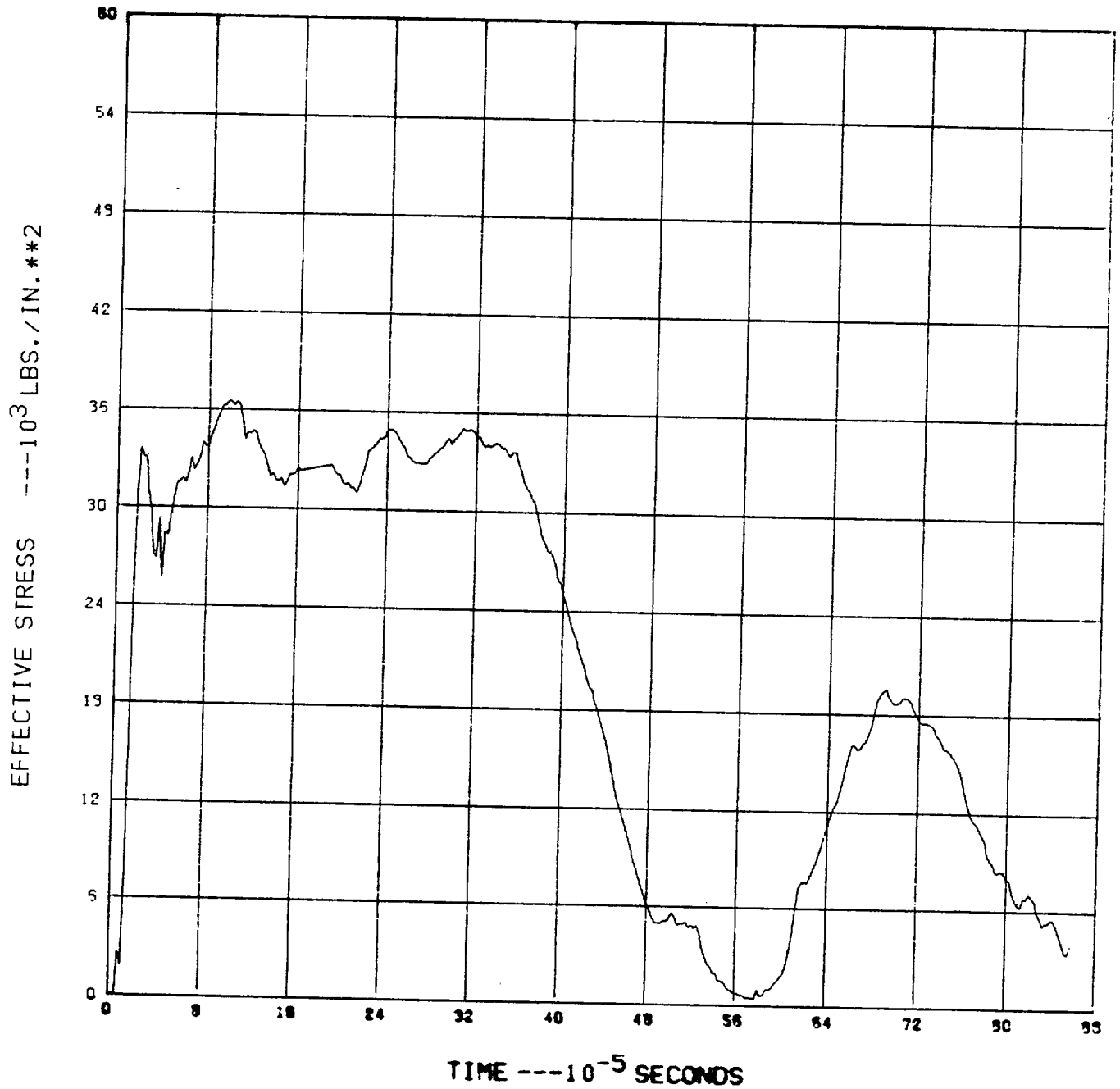## EFFECTIVE MEMBRANE + BENDING STRESS AT TOP OF SECTION 3

FIGURE 2-15 SECTION 3

PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT TOP OF PLATE

EFFECTIVE MEMBRANE + BENDING STRESS AT BOTTOM OF SECTION 3
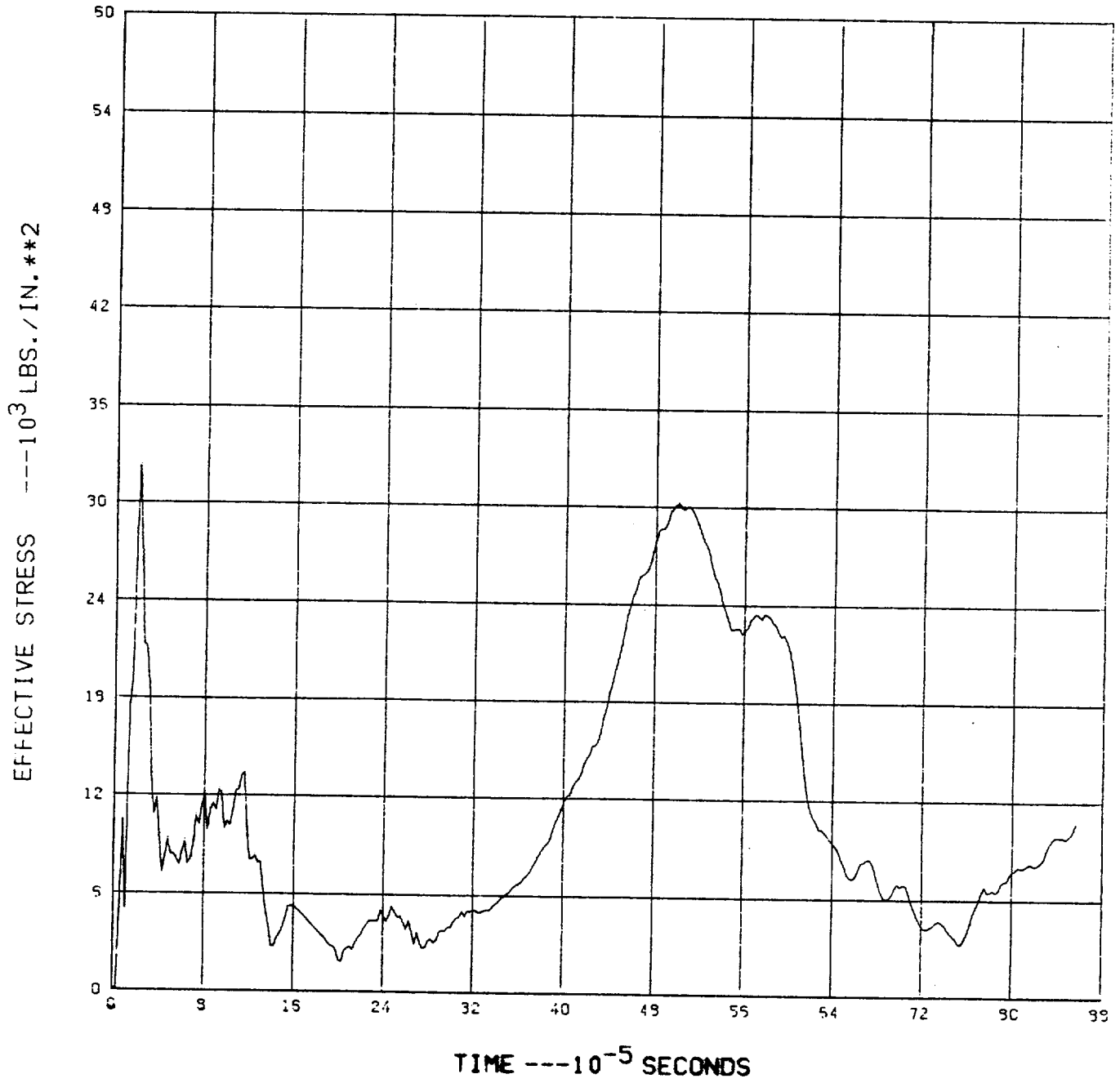

TIME ---10^{-5} SECONDS

FIGURE 2-16   SECTION 3

PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT BOTTOM OF PLATE

EFFECTIVE PRIMARY MEMBRANE STRESS AT SECTION 4

FIGURE 2-17 SECTION 4
PRIMARY MEMBRANE STRESS

EFFECTIVE MEMBRANE + BENDING STRESS AT TOP OF SECTION 4

TIME ---10^{-5} SECONDS

FIGURE 2-18 SECTION 4

PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT TOP OF PLATE

FIGURE 2-19 SECTION 4

PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT BOTTOM OF PLATE
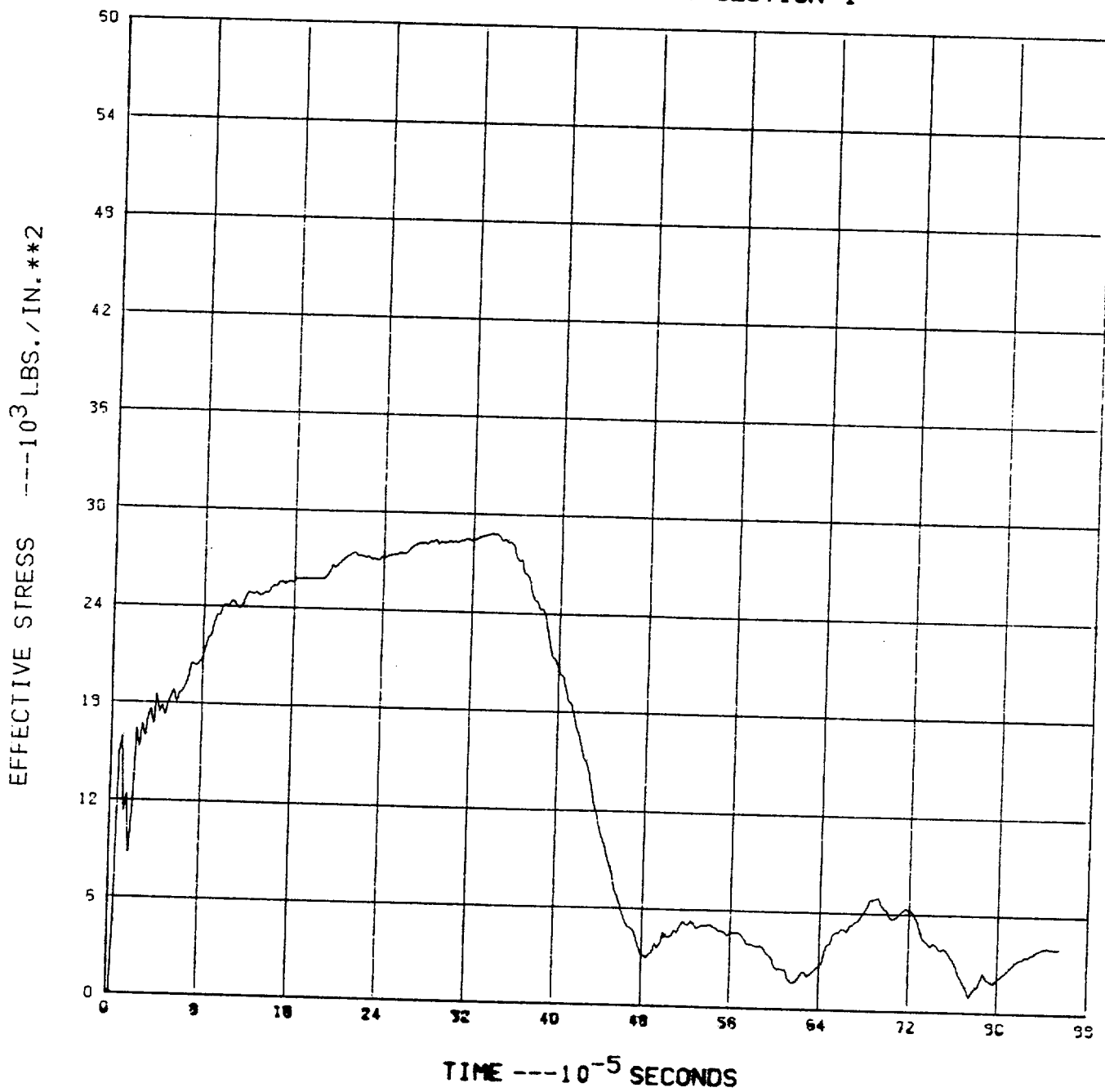
EFFECTIVE PRIMARY MEMBRANE STRESS AT SECTION 5
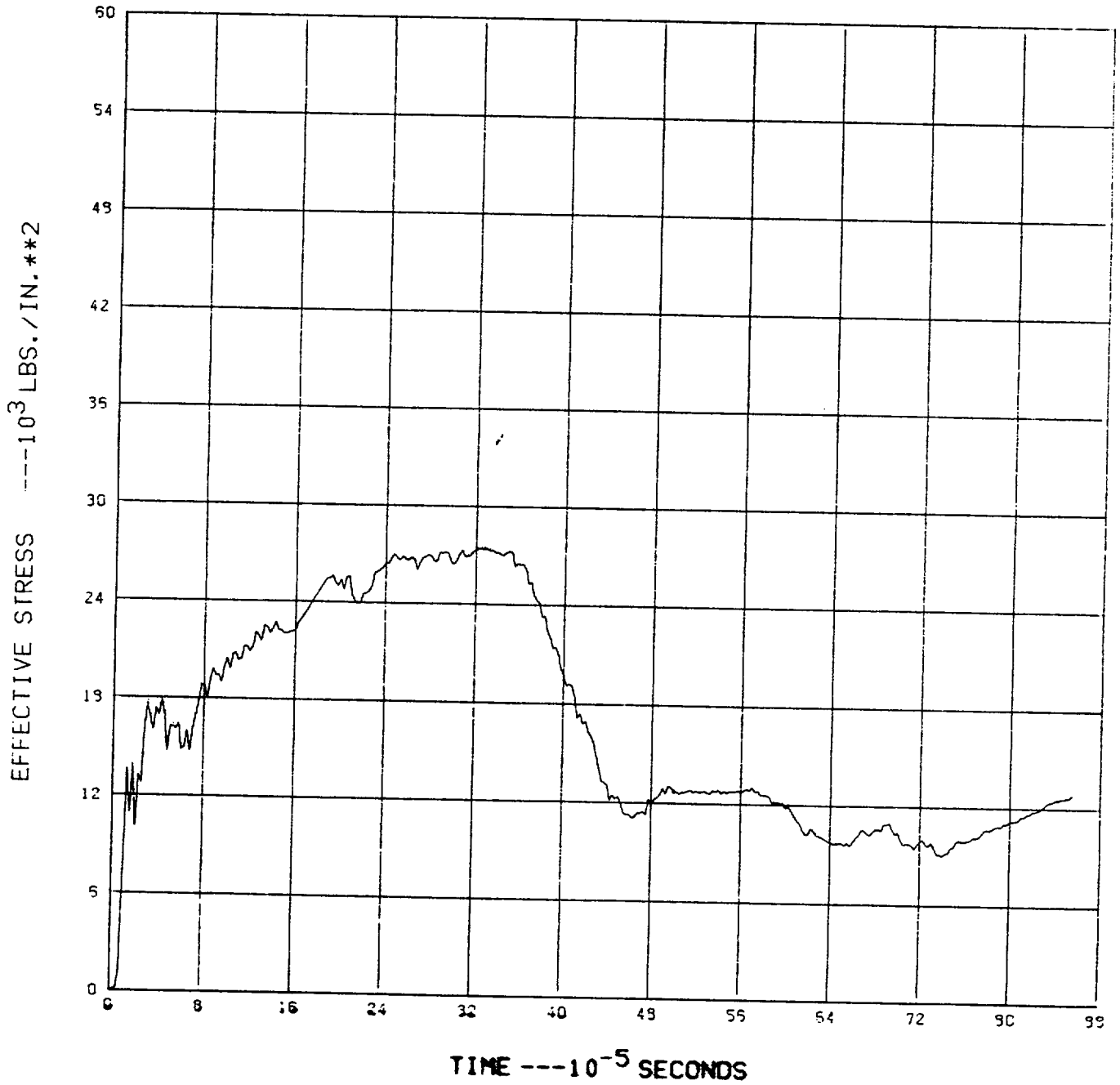
FIGURE 2-20 SECTION 5
PRIMARY MEMBRANE STRESS
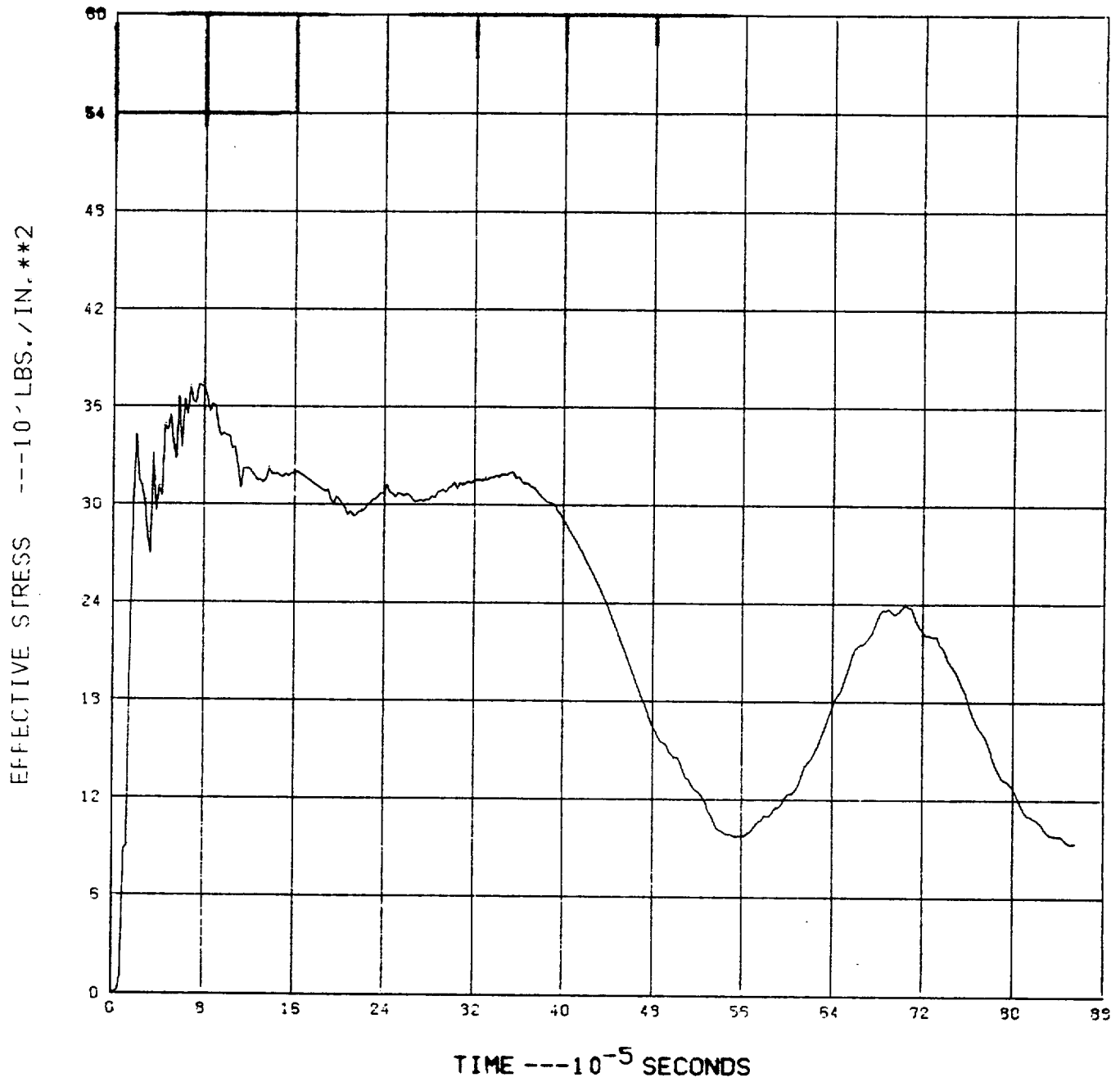
EFFECTIVE MEMBRANE + BENDING STRESS AT TOP OF SECTION 5

TIME ---10^{-5} SECONDS

FIGURE 2-21 SECTION 5

PRIMARY MEMBRANE PLUS PRIMARY BENDING
AT INSIDE EDGE OF WELD

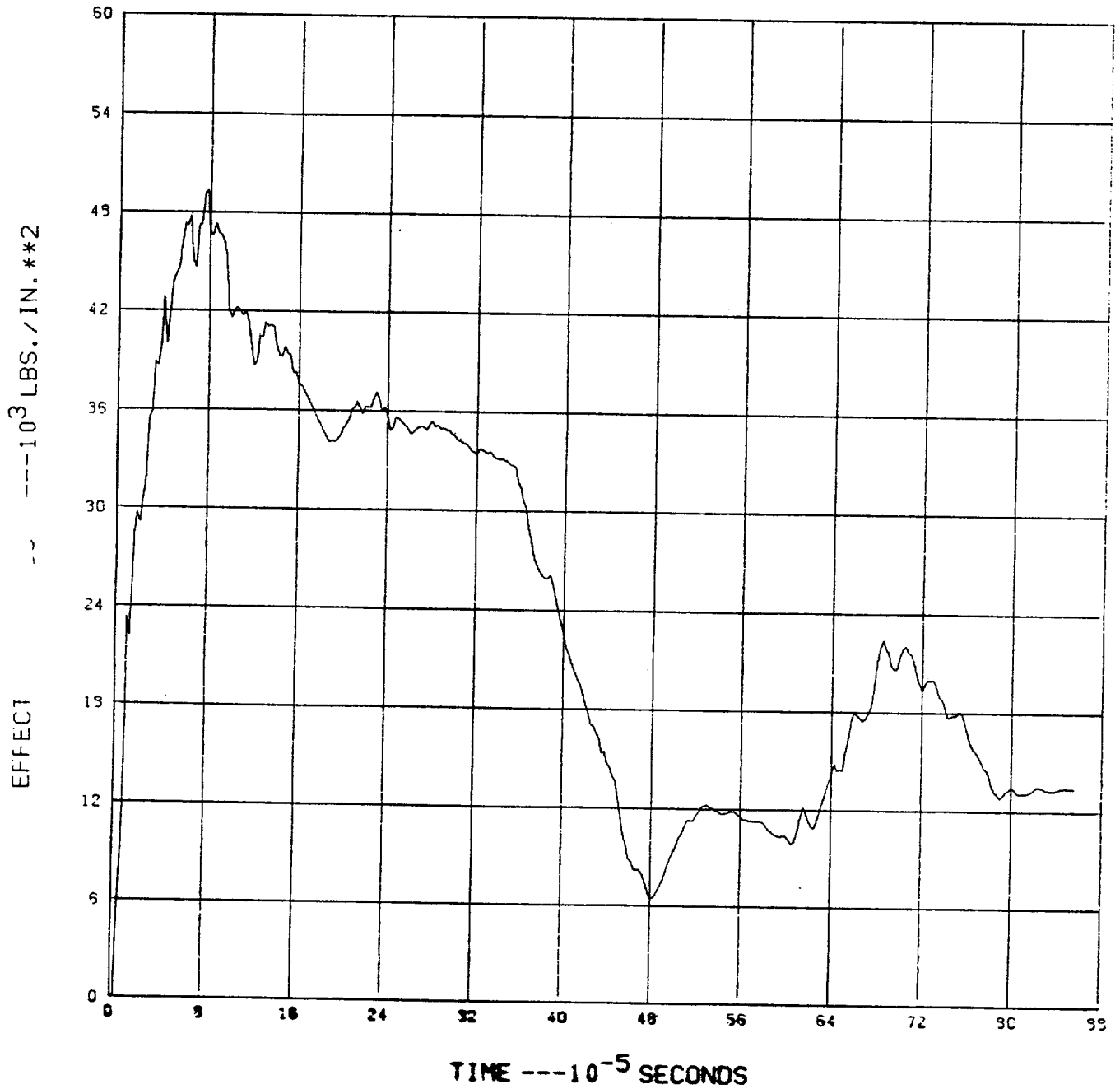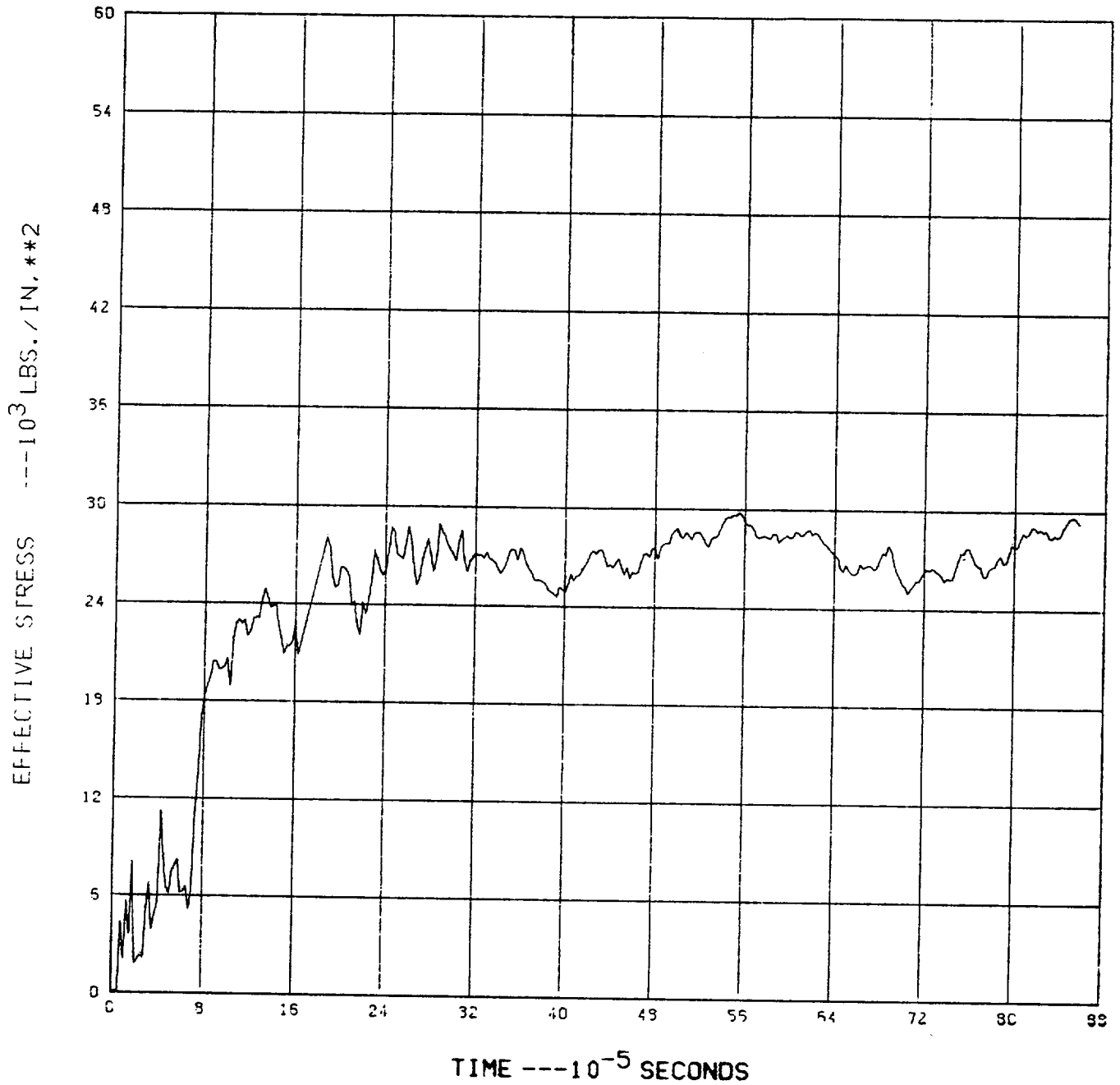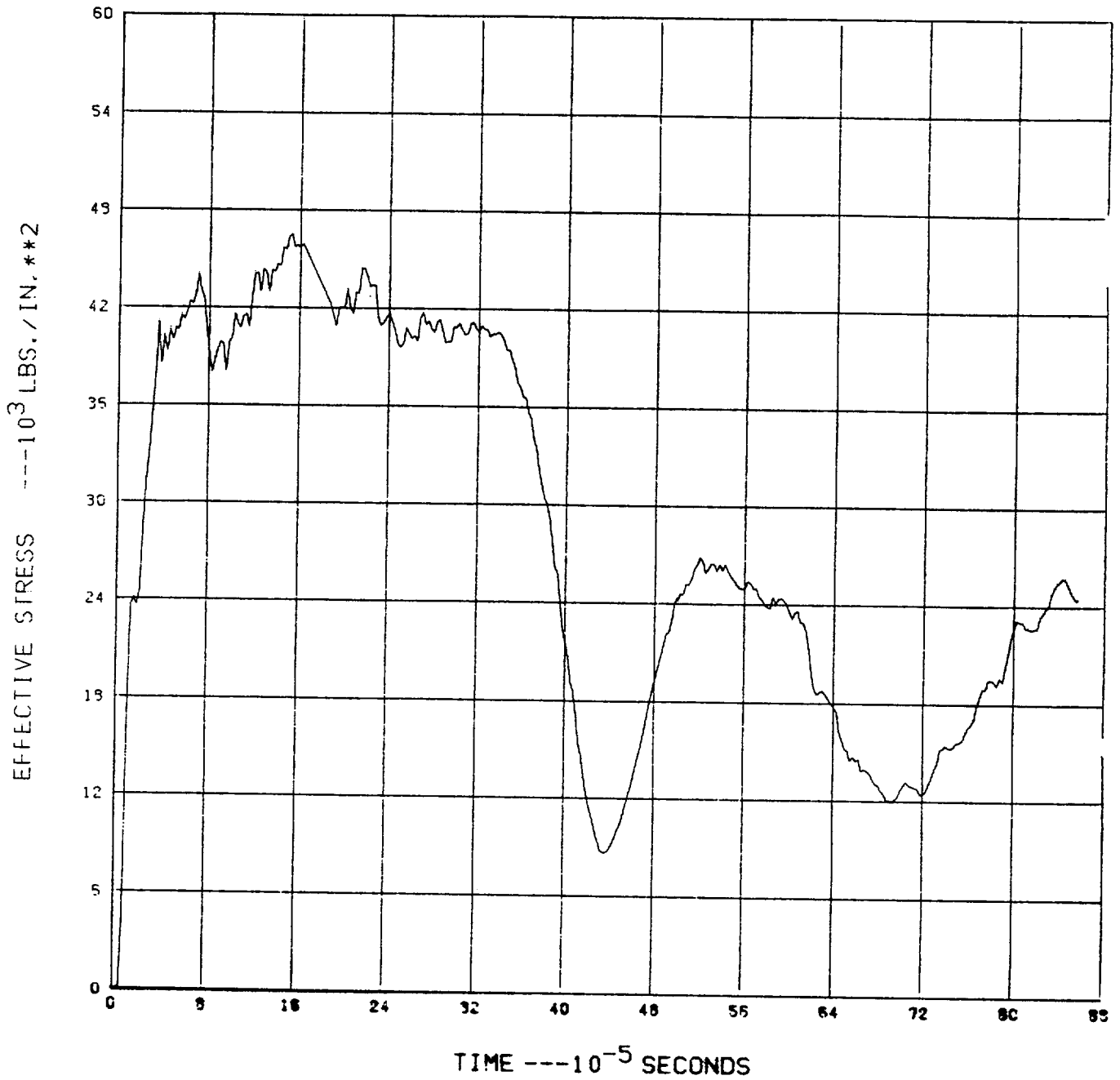## EFFECTIVE MEMBRANE + BENDING STRESS AT BOTTOM OF SECTION 5

FIGURE 2-22 SECTION 5

PRIMARY MEMBRANE PLUS PRIMARY BENDING
AT OUTSIDE EDGE OF WELD

TABLE 2-2
PRIMARY STRESSES

Location	Effective Stress (psi)	Time of Max Stress (ms)	% of Allowable
Section 1 Center of Plate			
General Membrane*	26,975	0.14	49
Membrane + Bending Top+	41,615	0.65	53
Membrane + Bending Bottom	45,806	0.403	58
Section 2 Inner Edge of Ring			
General Membrane	19,079	0.118	34
Membrane + Bending at Top	38,851	0.363	49
Membrane + Bending at Bottom	53,501	0.765	68
Section 3 Plate at Flange			
General Membrane	37,112	0.340	67
Membrane + Bending at Top	36,496	0.098	46
Membrane + Bending at Bottom	32,235	0.020	41
Section 4 Plate at Flange			
General Membrane	28,822	0.338	52
Membrane + Bending at Top	37,303	0.075	47
Membrane + Bending at Bottom	49,310	0.078	63
Section 5 Outer Weld			
General Membrane	27,435	0.323	49
Membrane + Bending at Top	29,884	0.555	38
Membrane + Bending at Bottom	46,525	0.153	59

*Membrane Allowable = S_y + 1/3 (S_{ut} - S_y) = 55,437 psi

(See page 5-2 for criteria and 5-3 for application)

+Membrane + Bending Allowable 0.7 S_{ut} = 78,618 psi

ensure that the shield ring will remain in place and be functional following the 30-ft hypothetical accidental drop.

2.7.3.7.   Flat Top End Drop

In the flat top end drop, the typical burial liner contents (control rods) will bear upon the shield ring. An impact ring 1/2 in. x 1/2 in. in cross section is provided on the outside of the burial liner cover to transmit all of the load from the burial liner and contents to the interface ring attached to the cask closure. As analyzed in Section 2.7.1.3, the cask closure is designed to absorb loads from a mass well in excess of the four control rods and burial liner.

2.7.4   Evaluation of the Optional Temporary Shield Tubes

2.7.4.1   Discussion.   The following sections provide structural analyses which demonstrate that the optional temporary shield tubes withstand the hypothetical accident conditions and maintain shielding effectiveness. The evaluation for hypothetical drop conditions in Section 2.7.1 shows that the maximum longitudinal and transverse cask decelerations during the corner drops is less than those for the end and side drops. Thus, only these two drop orientations are selected for evaluation.

As shown in drawing GADR 55-2-14 in Chapter 1, the optional temporary shield tubes are austenitic stainless steel. The sidewalls of the shield tubes are 6 in. diameter pipe, with either a 1-in. or 2-in. wall thickness. The bottom is 4 in. thick, and the lid is 5 in. thick. The lid is fastened with 1/2-13 UNC bolts and 3/8-in. locating pins which are also stainless steel.

2.7.4.2.   Evaluation Criteria.   Although the temporary shielding is not a containment boundary component, for conservatism the design criteria from USNRC Regulatory Guide 7.6, "Design Criteria for Structural Analysis of

Shipping Cask Containment Vessels," will be used. Applying these criteria, the allowable stress intensities become:

Primary membrane $2.4 \, S_m = 40$ ksi.

Membrane plus bending $3.6 \, S_m = 60$ ksi.

for $S_m = 16.7$ ksi. Based on Table I-1-2 of the ASME Code (Ref. 2-17), 16.7 is the minimum value of S_m for the austenitic stainless steels to be used for the optional shield tubes. This value of S_m is allowable for temperatures less than 300°F, and is consistent with the thermal results presented in Section 3.4.

Because the shielding is austenitic stainless steel, brittle fracture is not a concern.

2.7.4.3. End Drop. The temporary shielding will be analyzed for a 735 g deceleration during the end drop, the same deceleration calculated for the impacting end of the cask in Section 2.7.1.1.
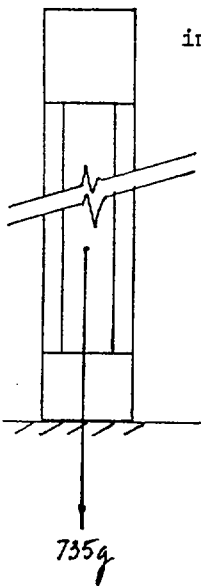
Sidewalls

$$\sigma = \frac{WG}{A}$$

where W = canister weight

G = 735 g's

A = cross sectional area of shielding sidewalls

For the 1-in.-thick shielding:

$$A = \frac{\pi}{4} [(6 \text{ in.})^2 - (4 \text{ in.})^2] = 15.7 \text{ in.}^2$$

$$W_{sidewall} = A \, (144 \text{ in.}) \left(0.285 \, \frac{\text{lb}}{\text{in.}^3}\right) = 644 \text{ lb}$$

735 g

$$A_{end} = \frac{\pi}{4} (6 \text{ in.})^2 = 28.3 \text{ in.}^2$$

$$W_{end} = A_{end} (5 \text{ in.}) \left(0.285 \frac{lb}{in.^3}\right) = 40 \text{ lb}$$

Therefore:

$$\sigma = \frac{(644 + 40) \text{ lb } (735 \text{ g})}{15.7 \text{ in.}^2} = 32.0 \text{ ksi} < 40 \text{ ksi, F.S.} = 1.25$$

For the 2-in.-thick shielding:

$$A = \frac{\pi}{4} [(6 \text{ in.})^2 - (2 \text{ in.})^2] = 25.1 \text{ in.}^2$$

$$W_{sidewall} = A (144 \text{ in.}) \left(0.285 \frac{lb}{in.^3}\right) = 1,030 \text{ lb}$$

Therefore:

$$\sigma = \frac{(1030 + 40) \text{ lb } (735 \text{ g})}{25.1 \text{ in.}^2} = 31.3 \text{ ksi} < 40 \text{ ksi, F.S.} = 1.28$$

Ends
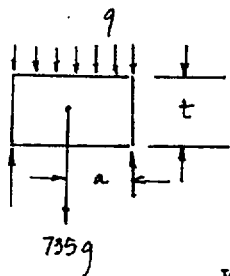
For the maximum plate stress, assume simply supported:

$$\sigma_b = \frac{6M}{t^2}$$

where $M = q \frac{a^2 (3 + \nu)}{16}$ (Ref. 2-18, p. 363)

$t_1 = 5$ in., $t_2 = 4$ in.

$\nu = 0.3 =$ Poisson's ratio

$q = g \frac{(W_{end})}{(A_{end})}$

$a = 3$ in.

For the 5-in.-thick lid:

$$q = \frac{(735 \text{ g}) (40 \text{ lb})}{28.3 \text{ in.}^2} = 1039 \frac{\text{lb}}{\text{in.}^2}$$

$$M = \frac{\left(1039 \frac{\text{lb}}{\text{in.}^2}\right) (3 \text{ in.})^2 (3.3)}{16} = 1,929 \frac{\text{in.} - \text{lb}}{\text{in.}}$$

$$\sigma_b = \frac{6 (1929 \text{ lb})}{(5 \text{ in.})^2} = 0.5 \text{ ksi} \ll 60 \text{ ksi, F.S.} = 120$$

For the 4-in.-thick bottom plate:

$$q = \frac{(735 \text{ g}) (32 \text{ lb})}{28.3 \text{ in.}^2} = 831 \frac{\text{lb}}{\text{in.}^2}$$

$$M = \frac{\left(831 \frac{\text{lb}}{\text{in.}^2}\right) (3 \text{ in.})^2 (3.3)}{16} = 1542 \frac{\text{in.} - \text{lb}}{\text{in.}}$$

$$\sigma_b = \frac{6 (1542 \text{ lb})}{(4 \text{ in.})^2} = 0.6 \text{ ksi} \ll 60 \text{ ksi, F.S.} = 100$$

2.7.4.4 <u>Side Drop</u>. During the side drop, the sidewall of the optional temporary shield tube is uniformly supported and decelerates at 84 g's, the maximum cask deceleration during the side drop calculated in Section 2.7.1.2.

From Ref. 2-18, p. 231:
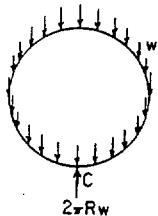
$$M_{max} = \frac{3}{2} WR^2 \text{ at pt. C} \tag{1}$$

$$T = T_A U + V_A Z + LT_T \tag{2}$$

$$V = T_A Z + V_A U + LT_V \tag{3}$$

where W = weight per unit inch circumference

R = radius = 3 in.

$T_A = \dfrac{WR}{2}$

$V_A = 0$

$LT_A = -WRXZ$

$LT_V = -WRXU$

X = angular distance to $C = \pi$ radiaus

$U = \cos X = -1$

$Z = \sin x = 0$

For the 1-in.-thick shielding, assuming a 1-in. depth:

$$w = \frac{\text{weight}}{\text{circumference}} \times G = \frac{(15.7 \text{ in.}^2)(1 \text{ in.})\left(0.285 \ \dfrac{\text{lb}}{\text{in.}^3}\right)(84 \text{ g})}{\pi \ (6 \text{ in.})} = 20 \ \frac{\text{lb}}{\text{in.}}$$

$$T_A = \frac{\left(20 \ \dfrac{\text{lb}}{\text{in.}}\right)(3 \text{ in.})}{2} = 30 \text{ lb}$$

$LT_T = 0$

$LT_V = -\left(20 \ \dfrac{\text{lb}}{\text{in.}}\right)(3 \text{ in.})(\pi)(-1) = 188.5 \text{ lb}$

Substituting into equations (1) through (3):

$$M_{max} = \frac{3}{2} \left(20 \ \frac{\text{lb}}{\text{in.}}\right)(3 \text{ in.})^2 = 270 \text{ in.-lb per 1-in. depth}$$

$T = (30 \text{ lb})(-1) + 0 + 0 = -30 \text{ lb}$

$V = 0 + 0 + 188.5 \text{ lb} = 188.5 \text{ lb}$

Then:

$$\sigma_m = \frac{T}{A} + \frac{V}{A} = \frac{30 \text{ lb} + 188.5 \text{ lb}}{1 \text{ in.}^2} = 0.2 \text{ ksi} \ll 40 \text{ ksi}, \ F.S. = 200$$

$$\sigma_m + \sigma_b = 0.2 + \frac{6M}{t^2} = \frac{6 \ (270 \text{ in.-lb})}{(1 \text{ in.})^2} = 1.8 \text{ ksi} < 60 \text{ ksi}, \ F.S. = 33$$

where A is the cross-sectional area of the tube at pt. C, the point of impact, assuming a 1-in. depth.

For the 2-in.-thick shielding, assuming a 1-in. depth:

$$w = \frac{\text{weight}}{\text{circumference}} \times G = \frac{(25.1 \text{ in.}^2)(1 \text{ in.})\left(0.285 \frac{lb}{\text{in.}^3}\right)(84 \text{ g})}{\pi (6 \text{ in.})} = 32 \frac{lb}{\text{in.}}$$

$$T_A = \frac{\left(32 \frac{lb}{\text{in.}}\right)(3 \text{ in.})}{2} = 48 \text{ lb}$$

$$LT_T = 0$$

$$LT_V = -\left(32 \frac{lb}{\text{in.}}\right)(3 \text{ in.})(\pi)(-1) = 302 \text{ lb}$$

Substituting into equations (1) through (3):

$$M_{max} = \frac{3}{2}\left[32 \frac{lb}{\text{in.}}\right](3 \text{ in.})^2 = 432 \text{ in.-lb per 1-in. depth}$$

$$T = 48 \text{ lb } (-1) + 0 + 0 = -48 \text{ lb}$$

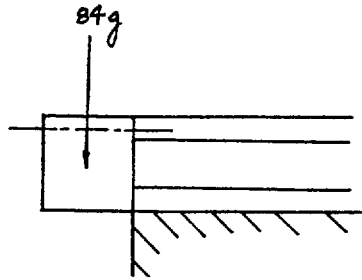$$V = 0 + 0 + 302 \text{ lb} = 302 \text{ lb}$$

Then:

$$\sigma_m = \frac{T}{A} + \frac{V}{A} = \frac{48 + 302}{2 \text{ in.}^2} = 0.2 \text{ ksi} \ll 40 \text{ ksi, F.S.} = 200$$

$$\sigma_m + \sigma_b = 0.2 + \frac{6M}{t^2} = \frac{6\left(432 \text{ in.} - \frac{lb}{\text{in.}}\right)}{(2 \text{ in.})^2} = 0.8 \text{ ksi} \ll 60 \text{ ksi, F.S.} = 75$$

Bolts and Locating Pins

Since the temporary shielding is supported uniformly on all sides, there will be no stress on the bolts or locating pins during a side drop. However, for conservatism, the lid will be assumed to be supported during a side drop as follows.

Since the locating pins have a tighter fit than the bolts, they will react shear.

$$\tau_{pin} = \frac{W_{lid}G}{2A_\sigma} \text{ (shear)}$$

where   G = 84 g's

A_σ = stress area of a 3/8-in. pin = 0.1104 in.^{2}

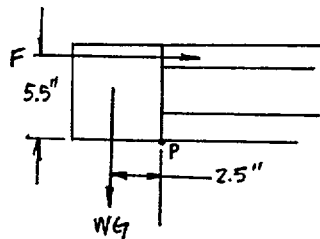3 = number of pins

W_{lid} = 40 lb

$$\tau_{pin} = \frac{(40 \text{ lb}) (84)}{2 (0.1104 \text{ in.}^2)} = 15.2 \text{ ksi}$$

$$\sigma = 2\tau = 30.4 \text{ ksi} < 40 \text{ ksi, F.S.} = 1.3$$

The bolts will react the moment which tends to rotate the lid. Conservatively assuming that only one of the three bolts reacts this moment:

$$\Sigma M_p = 0 = (40 \text{ lb})(84 \text{ g})(2.5 \text{ in.}) - F(5.5 \text{ in.})$$

$$F = 1.53 \text{ kips}$$

$$\sigma_{bolt} = \frac{F}{A}$$

where A = stress area at a 1/2-13 UNC bolt = 0.1419 in.^{2}

$$= \frac{1.53 \text{ kips}}{0.1419 \text{ in.}^2} = 10.8 \text{ ksi} < 40 \text{ ksi, F.S.} = 3.7$$

Welds

Again, since the optional temporary shield tube is supported uniformly on all sides, there will be no stress on the welds. However, for conservatism, the welds will be assumed to support the end during side deceleration as follows:

$$\tau = \frac{W_{end}}{A_w}$$

where   G = 84 g's

$$A_w = \frac{\pi}{4} [(6 \text{ in.})^2 - (5.5 \text{ in.})^2] = 4.5 \text{ in.}^2$$

$$W_{end} = 32 \text{ lb}$$

Therefore:

$$\tau = \frac{(32 \text{ lb})(84 \text{ g})}{4.5 \text{ in.}^2} = 0.6 \text{ ksi}$$

$$\sigma = 2\tau = 1.2 \text{ ksi} < 40 \text{ ksi, F.S.} = 33.$$

2.7.4.6 Summary of Results. The results of the structural evaluation for the optional temporary shield tubes are summarized in Table 2-3. As shown in the table, all of the stresses calculated are below the allowables of Regulatory Guide 7.6.

2.8 APPENDIX

2.8.1 Low Temperature Uranium Bar Drop Tests

2.8.1.1 Discussion. In an effort to obtain a preliminary indication of the
ductility properties of uranium at subzero temperatures, a series of drop test
was scheduled and conducted at National Lead Company's Albany branch. Samples
of unalloyed depleted uranium were tested along with various samples of low
alloys to serve as a basis for comparison.

Various diameters of as cast depleted uranium round bar were used;
however, each round bar was cut to length such that the length to diameter
ratio was 8 to 1 - this being approximately the same as the L/D ratio of the
depleted uranium in Model FSV-1 Configurations A, B, C and D.

Most drops were conducted from a height of 29 feet, 3 inches onto
essentially unyielding surfaces of either steel or concrete. Two samples,
also from a height of 29 feet 3 inches, were dropped on a sharp edged fulcrum.
Temperature at the top of drop of all samples ranged between -55°F to -60°F.

Results of these preliminary drop test indicated that the unalloyed
depleted uranium exhibited good ductility properties at low temperatures.
Furthermore, it wasn't until the third 29 feet 3 inches drop of a test
specimen that ductility failure occurred. A 1-3/16" diam by 9-1/2 inch long
unalloyed depleted uranium round bar was dropped on a concrete impact surface
with no visible failure resulting. The same test specimen was then dropped
from the same height of 29 feet 3 inches and at the same temperature of -60°F
on a steel plate. This time a slight bend in the bar was noted. On the third
test, the specimen was dropped on the sharp edge of a steel angle. Along with
a greater bend in the bar, it was noted that a small crack developed opposite
the side of impact.

2.8.1.2 Results

Test No.	Specimen Drop No.	Material (As-Cast)	Diameter (in.)	Attitude at Impact	Impacting Surface	Test Results
1	1	U-2% Mo	1-1/4	Horizontal	Concrete	No failure
2	1	Unalloyed	1-3/16	Horizontal	Concrete	No failure
3	1	Unalloyed	1-3/16	Horizontal	Concrete	No failure
4	1	U-1% Mo, 1% Nb	0.6	Horizontal	Concrete	No failure
5	1	U-1% Mo, 1% W	0.6	45° Corner Drop	Concrete	No failure
6	1	U-1% Nb, 1% W	0.6	45° Corner Drop	Concrete	No failure
7	1	U-1% Ta, 1% W	0.6	Horizontal	Concrete	No failure
8	1	Unalloyed	0.6	Horizontal	Concrete	No failure
9	1	Unalloyed	0.6	Horizontal	Concrete	No failure
10	2	Unalloyed	1-3/16	Horizontal	Stl Plate	Slight Bend
11	2	Unalloyed	1-3/16	Horizontal	Stl Plate	Slight Bend
12	2	U-2% Mo	1-1/4	Horizontal	Stl Plate	No failure
13	2	U-1% Mo, 1% W	0.6	Horizontal	Stl Plate	No failure
14	2	Unalloyed	0.6	Horizontal	Stl Plate	Slight Bend
15	3	Unalloyed	1-3/16	Horizontal	90° Corner	Bent & Cracked
16	3	Unalloyed	1-3/16	Slight Angle Corner Drop	90° Corner	Bent

Test Conditions

Height of drop   29'3"

Temperature of specimens   -55°F to -60°F

Material Condition   As cast

Material Configuration   Round bars, L/D = 8:1

2.8.2 Low Temperature 1/8 Scale Uranium Shell Impact Tests

2.8.2.1 Purpose of Tests. Experimental knowledge is required of the effects
of impact loads as used in casks with uranium shielding and having relatively
large L/D ratios. The experimental test specimen was subjected to several

puncture tests to determine the amount of deformation and to observe the surface condition of the uranium in the impacted areas.

2.8.2.2 Material Used. An as cast 0.2% - 0.3% molybdenum-uranium cylindrical shell, 1/8 the size of the actual shielding in the shipping cask, was used for this series of tests.

Comparative dimensions and weights:

	1/8 Scale Cylindrical Shell	Full Size Uranium Shielding
Outside diameter	3-1/4"	26"
Inside diameter	2-3/8"	19"
Length	24-3/8"	194"
Weight	64 lb	32,800 lb

2.8.2.3 Test Procedures. Tests were conducted using the indoor drop test facilities of NLC, Wilmington branch.

An iron-constantan thermocouple was attached to the outer surface of the uranium cylindrical shell. The thermocouple extension leads were connected to a calibrated pyrometer indicator.

The test specimen was then submerged in a solution of acetone and dry-ice until it reached a temperature of approximately -60°F. With the thermocouple still attached, the uranium was connected to a quick release mechanism which, in turn, was attached to an overhead crane. The entire assembly was moved over the impact area which consisted of a 3/4 inch wide carbon steel fulcrum 4 inch deep and 12 inch long welded to a 12 inch x 12 inch x 3/4 inch carbon steel base plate. This weldment was resting on a steel anvil pad supported by a concrete foundation. With the use of a scaled line and plumb bob, the test

specimen was raised to a height of 40 inches over the fulcrum. Its long axis
was positioned level and perpendicular to the long axis of the fulcrum so that
the center of gravity of the shell would impact against the fulcrum.

When the pyrometer indicator measured a surface temperature of -40°F the
solenoid on the release mechanism was actuated, pulled a release pin and
allowed the test specimen to free fall on the fulcrum.

This same test was conducted at 60 inches, 80 inches, 100 inches and 120
inches. Measurements were taken after each test of the outside diameter,
inside diameter, length and angle of bend.

2.8.2.4 Results. Results of all free fall fulcrum tests proved negative.
The test specimen experienced no dimensional changes or deformations.

The 120 inches drop test had a rebound after impact of 31-1/4 inches
above the fulcrum--the test specimen remained level during the rebound.
Although no damage occurred to the uranium, the 3/4 inch fulcrum received an
indentation approximately 1/4 inch deep.

2.8.3 Uranium Weld Joint Study

2.8.3.1 Discussion. Successful completion of side puncture tests on a
uranium casting scale model has prompted a review of impact condition analyses
and has also allowed consideration of a type of joint for the uranium sections
which permits a greatly reduced depth of welding.

The previous calculations for side wall puncture assumed the onset of
plastic hinge deformation at 30,000 psi, and required that 83% of the kinetic
energy of the 40 in. drop had to be absorbed by this mechanism. The new drop
tests proved that more than double this height of drop did not produce any
measurable permanent set. Obviously, justification of this performance

requires that the calculated instantaneous peak bending stresses be allowed to reach for higher values and still be elastic.

The effects of this new performance are now investigated in regard to accident conditions, and include the behavior of the redesigned joints under such loadings.

According to the tests, separately reported, a drop height of 120 in., with a rebound of 31-1/4 in., can be credited, conservatively, with kinetic energy proportional only to a drop of 120 - 31-1/4 = 88-3/4 in. This means that energy 2.22 times the specified 40 in. drop was successful absorbed elastically.

2.8.3.2 Puncture Side Wall - 88-3/4 in. Drop Analysis. The stainless steel outer shell, as the member directly exposed to penetration by the steel piston, is not critical. The piston merely cuts partly into the wall, due to the back-up effects of the uranium cylinder.

The uranium cylinder (as tested) is taken as a model, but calculations are now made on a full size cylinder analyzed as a separate body in an elastic drop up to 88-3/4 in.
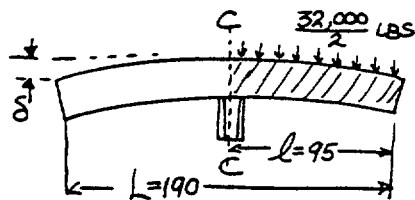
Consider 1G Load on Cantilever

$D = 26$ in., $d = 19$ in.

$\ell = 95$ in., $L = 190$ in.

Vol. $= (531-284)$ in.^{2} $(190$ in.$) = 46,930$ in.^{3} total

Wt $= 46,930 (0.683) = 32,000$ lb total

$I = 0.0491 (26^4 - 19^4) = 16,041$ in.^{4}

$Z = 16,041/13 = 1235$ in.

$$M_c = \frac{w\ell}{2} = \frac{32,000}{2} \cdot \frac{95}{2} = 760,000 \text{ in. lb}$$

$$\delta = \frac{w\ell^3}{8EI} = \frac{16,000 \, (95)^3}{8(29) \, 10^6 \, (16,041)} = 0.0369 \text{ in.}$$

$$S_b = \frac{M_c}{Z} = \frac{760,000}{1235} = 615 \text{ psi}$$

From Marks Handbook, sixth edition, page 5-44:

$$U_{cantilever} = \frac{n^2}{m} \left(\frac{K}{c}\right)^2 \frac{S^2 V}{2E} \quad \text{for uniform load}$$

$$n = 2 \quad m = 8 \quad K = \text{rad. gyr.} = \frac{\sqrt{D^2 + d^2}}{4} \quad \text{for tube section}$$

$$c = D/2$$

Therefore,

$$U_{cant.} = \frac{(2)^2}{8} \cdot \frac{4}{16} \cdot \frac{(D^2 + d^2)}{D^2} \cdot \frac{S^2 V}{2E} = \frac{D^2 + d^2}{16 D^2} \cdot \frac{S^2 V}{E} \quad \text{for 1/2 beam lgth.}$$

$$U_B = \frac{D^2 + d^2}{8D^2} \frac{S^2 V}{E} \quad \text{for full length beam as above}$$

$$= \frac{26^2 + 19^2}{8 \times 26^2} \frac{S^2(46,930)}{29(10^6)} = \frac{674 + 361}{8(676)} S^2 \frac{1615}{10^6} = \frac{310}{10^6} S^2$$

Now S = 615 psi for 1G load

$$U_B = \frac{310(615)^2}{10^6} = 117 \text{ in. lb} \quad \text{Note: } U_B \propto S^2 \propto (G's)^2$$

Line	G's	S psi	δ in.	(G's)²	U_B in. lb	Height Drop
1	1	615.	0.00369	1	117	
2	50	30,750	0.1845	2500	292,500	
3	104.5	64,300	0.386	10,920	1,280,000	40
4	156.	96,000	0.575	24,280	2,840,000	88-75

The stress of 96,000 psi is reached "instantaneously" and only at the top mid point of the beam. This peak is quite consistent with nominal properties of unalloyed cast uranium. Static compression, stress-strain curves for cast, unalloyed depleted uraniium are shown in Fig. 2-23.

The specification is limited to the values of Line 3. The joints are actually at positions B and the moment and stress is reduced to 4/9.
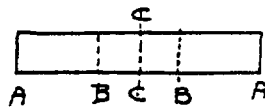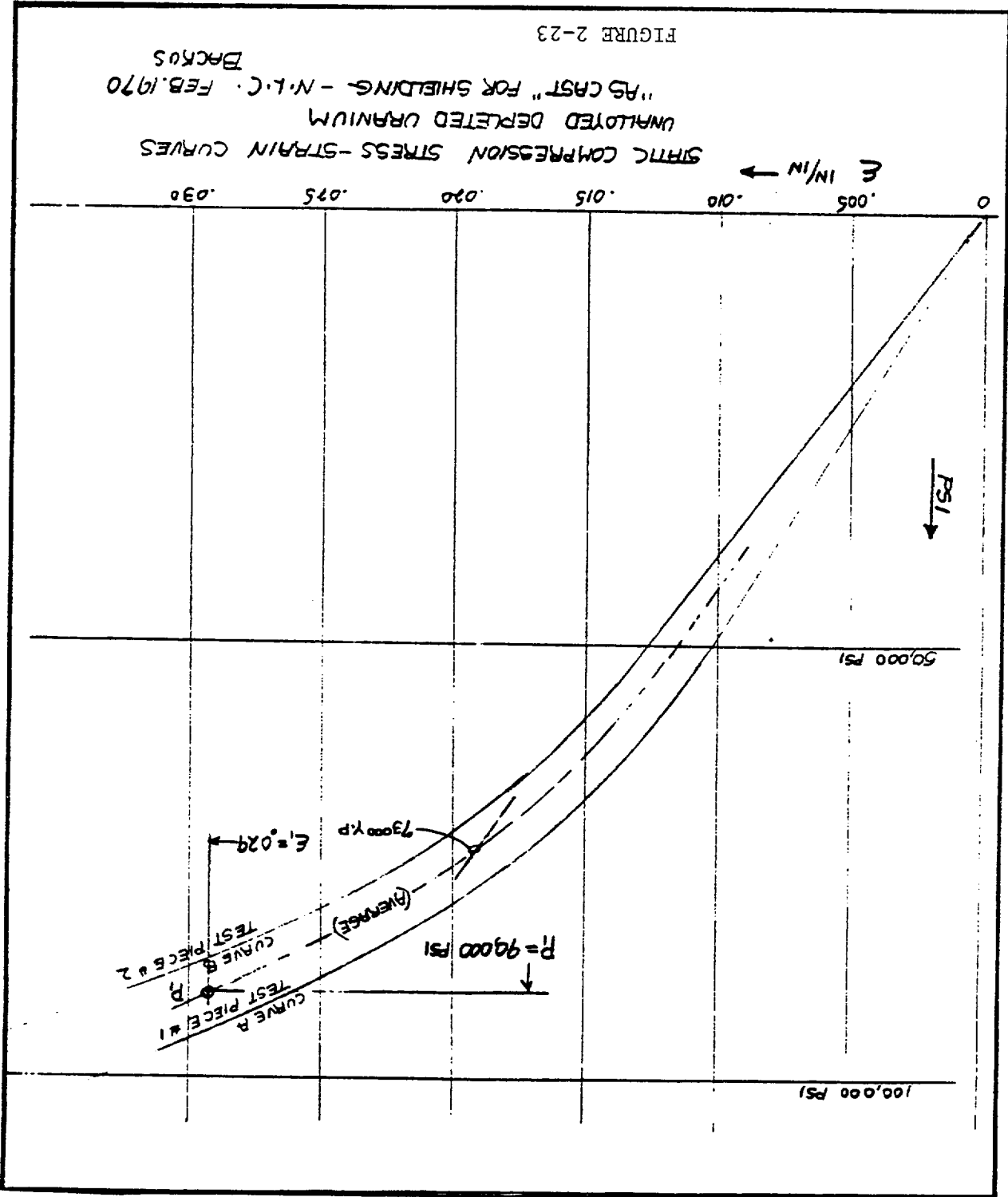
$$\frac{M_B}{M_C} = \frac{(2/3 \ Wt)(2/3\ell)}{(Wt)(\ell)} = 4/9 \quad \therefore \ S_B = \frac{4}{9} S_C = \frac{4}{9} (64,300)$$

$$= \underline{28,600 \ psi}$$

2.8.3.3 <u>Results</u>. Since the "solid" cylinder of the tests (equivalent to full depth penetration weld) has withstood at its midsection <u>96,000 psi</u>, it would be theoretically possible to reduce the depth of an outside weld for position B so that Z for the weld area is only

$$Z_{B2} = \frac{28,600}{96,000} \ 1235 = \underline{368}$$

to find the i.d. which corresponds to this Z value, let

$$Z_{B2} = 368 = \frac{0.098}{26} \ (26^4 - d^4) = 0.00378 \ [457,000 - d^4]$$

$$d^4 = \frac{1725 - 368}{0.00378} = 359,000$$

$$d = 24.5 \ in. \quad \therefore \ t = \frac{26 - 24.5}{2} = \underline{3/4\text{-in. weld req'd min.}}$$

To be quite conservative, it is desirable to <u>double this depth</u> of weld, using <u>2</u> layers of 3/4-in. penetration welds (in offset relationship), providing continuous beam strength in an outer annulus 1-1/2 in. thick out of a total of 3-1/2-in. wall. The inner 2 in. would be machined with interlocking steps (four of 1/2-in. each) providing concentric shear rings and an effective shielding pattern as shown on Fig. 2-24.
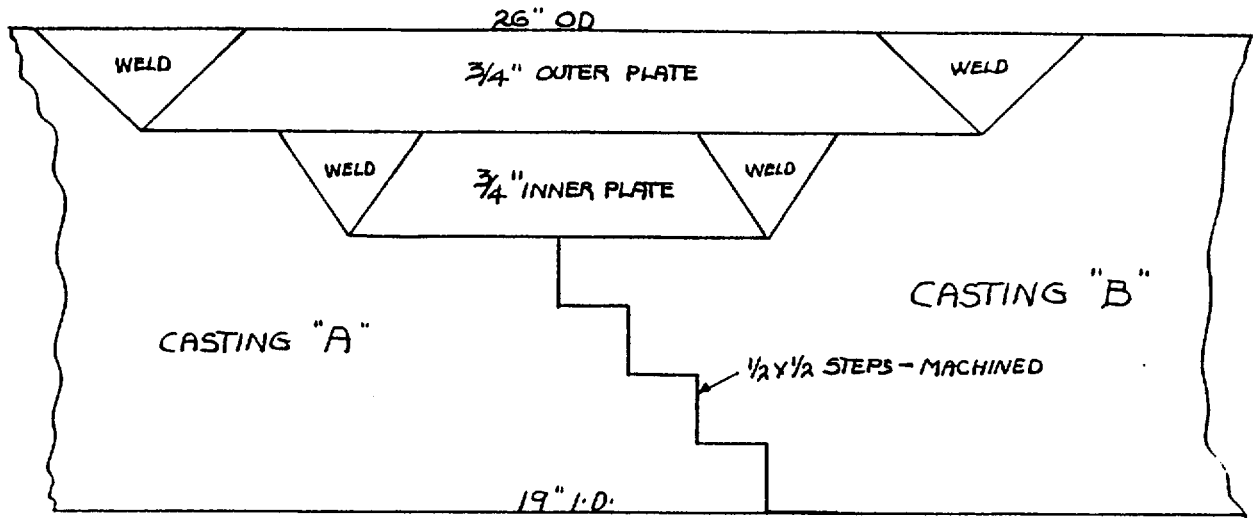
FIGURE 2-24   Depleted Uranium Joint Detail

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