

910013 B
GADR 55
VOLUME 1

CONSOLIDATED DESIGN REPORT FOR THE MODEL FSV-1 SHIPPING CASK

CONFIGURATIONS A, B, C, D

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 **GENERAL ATOMICS**

**910013 B
GADR 55
VOLUME 1**

**CONSOLIDATED DESIGN REPORT
FOR THE MODEL FSV-1 SHIPPING CASK**

CONFIGURATIONS A, B, C, D

ISSUE/RELEASE SUMMARY

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

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			ENGINEERING	QA	PROJECT	
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MAR 14 1996 2 RELEASED	B	R.O. Meyer 3/13/96				Released by CN 006486 (Roll 2306)

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GADR 55
Volume I

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 decontamination. 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 <u>46,025</u> 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, contaminated hardware and optional temporary shield tube	3720 lb	2970 lb ^(b)	Impact limiter not used; inner container 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 <u>46,025</u> 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 <u>46,025</u> lb.

(a)National Lead Company drawing.

(b)Including optional shield tube.

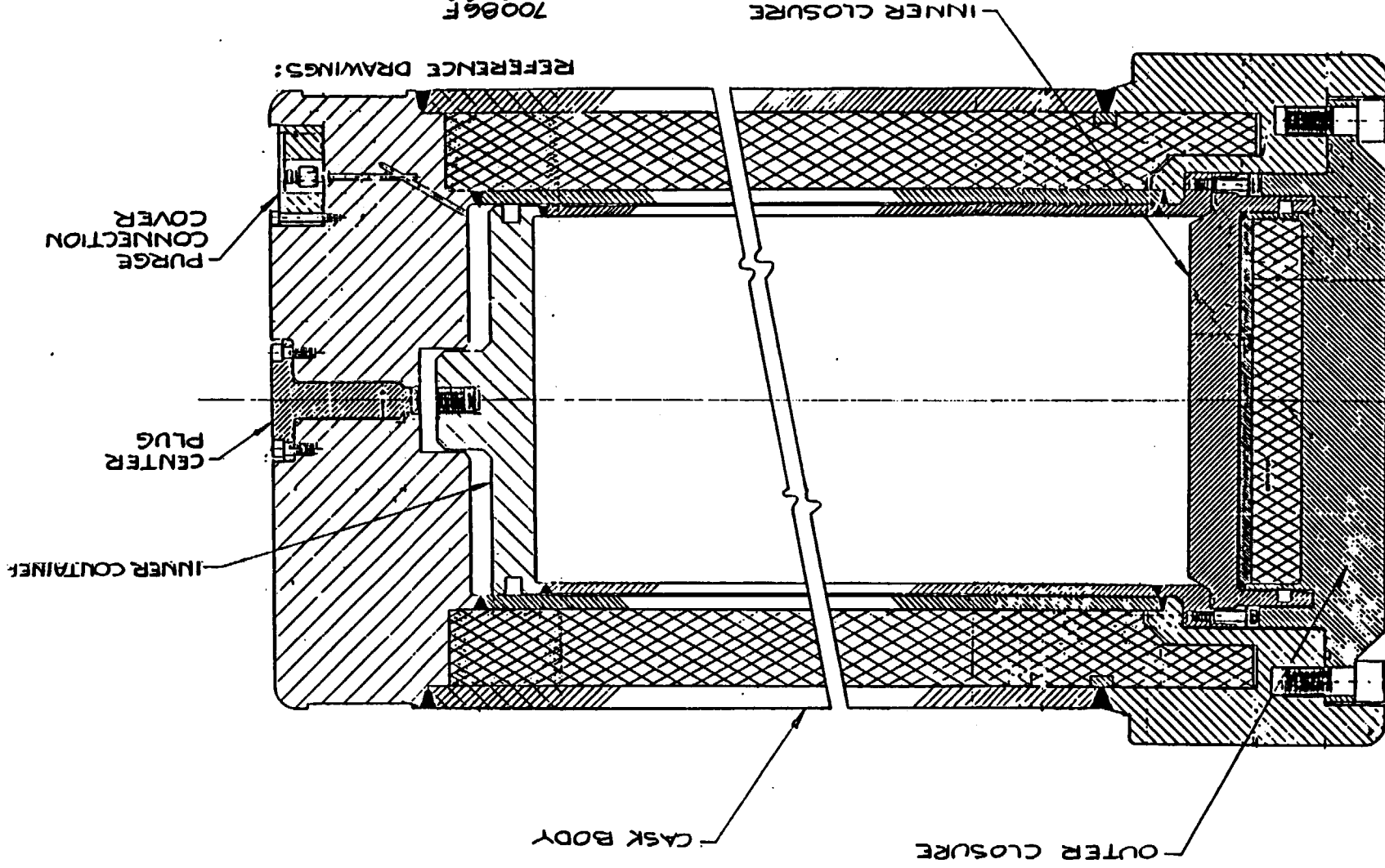
(c)General Atomica drawing.

MODEL FSV-1 CASK
CONFIGURATION A

FIGURE 1-1

70086F
1501-003C
70296F

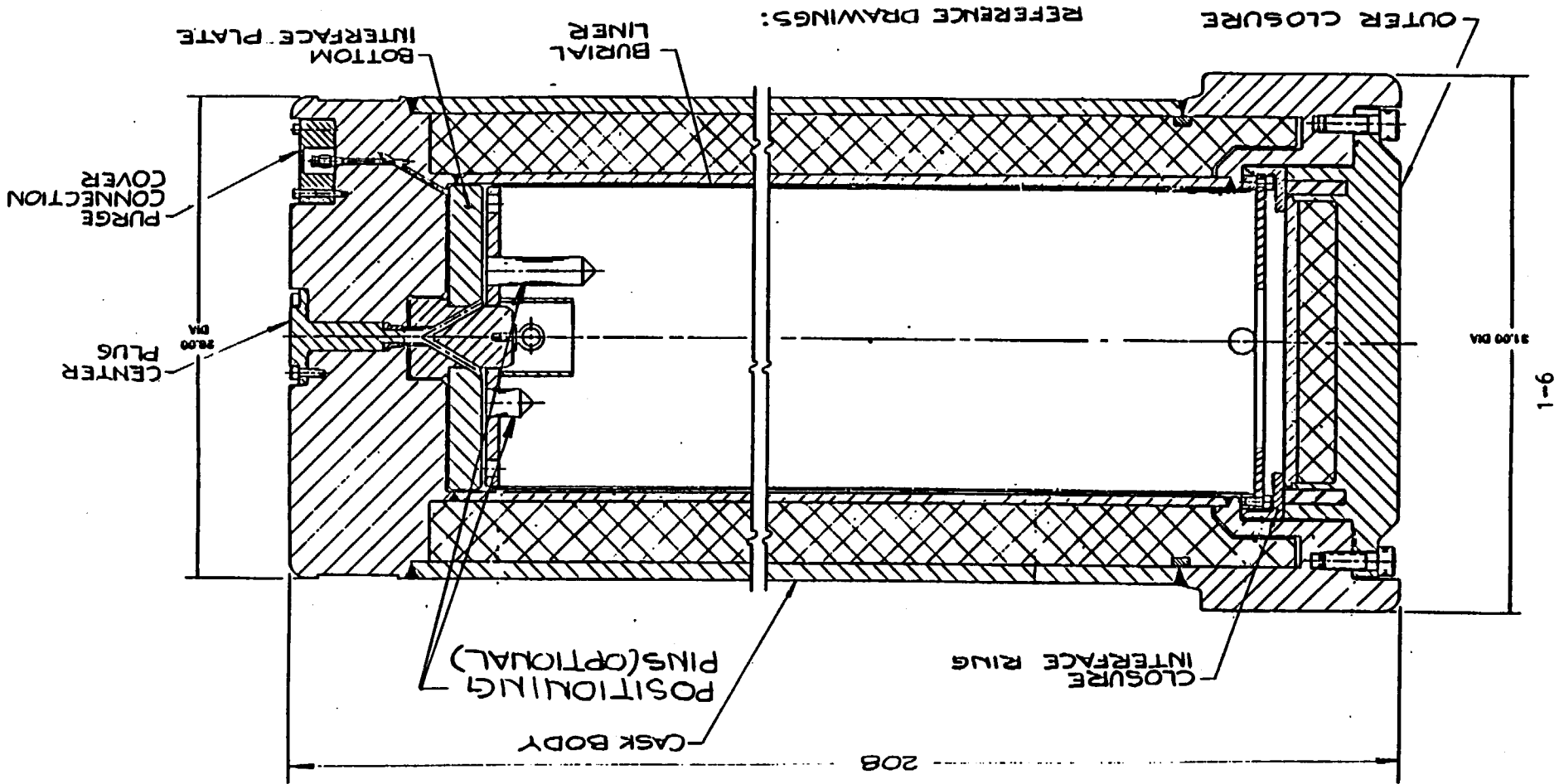
REFERENCE DRAWINGS:



FSV-1 CONFIGURATION B
FOR IRRADIATED HARDWARE
WITH BURIAL LINER

FIGURE 1-2

700B6F
1501-003C
70296F



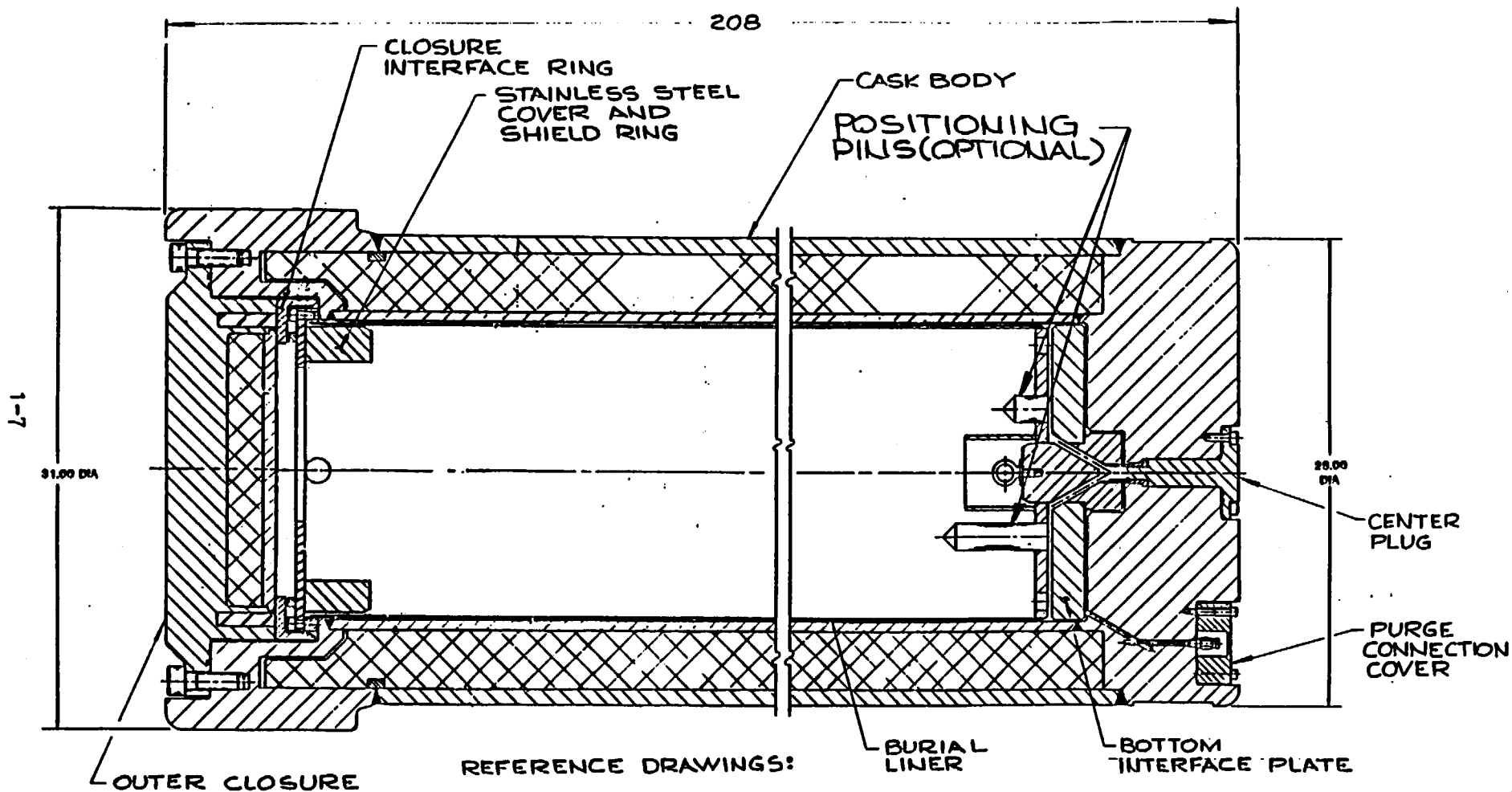


FIGURE 1-3

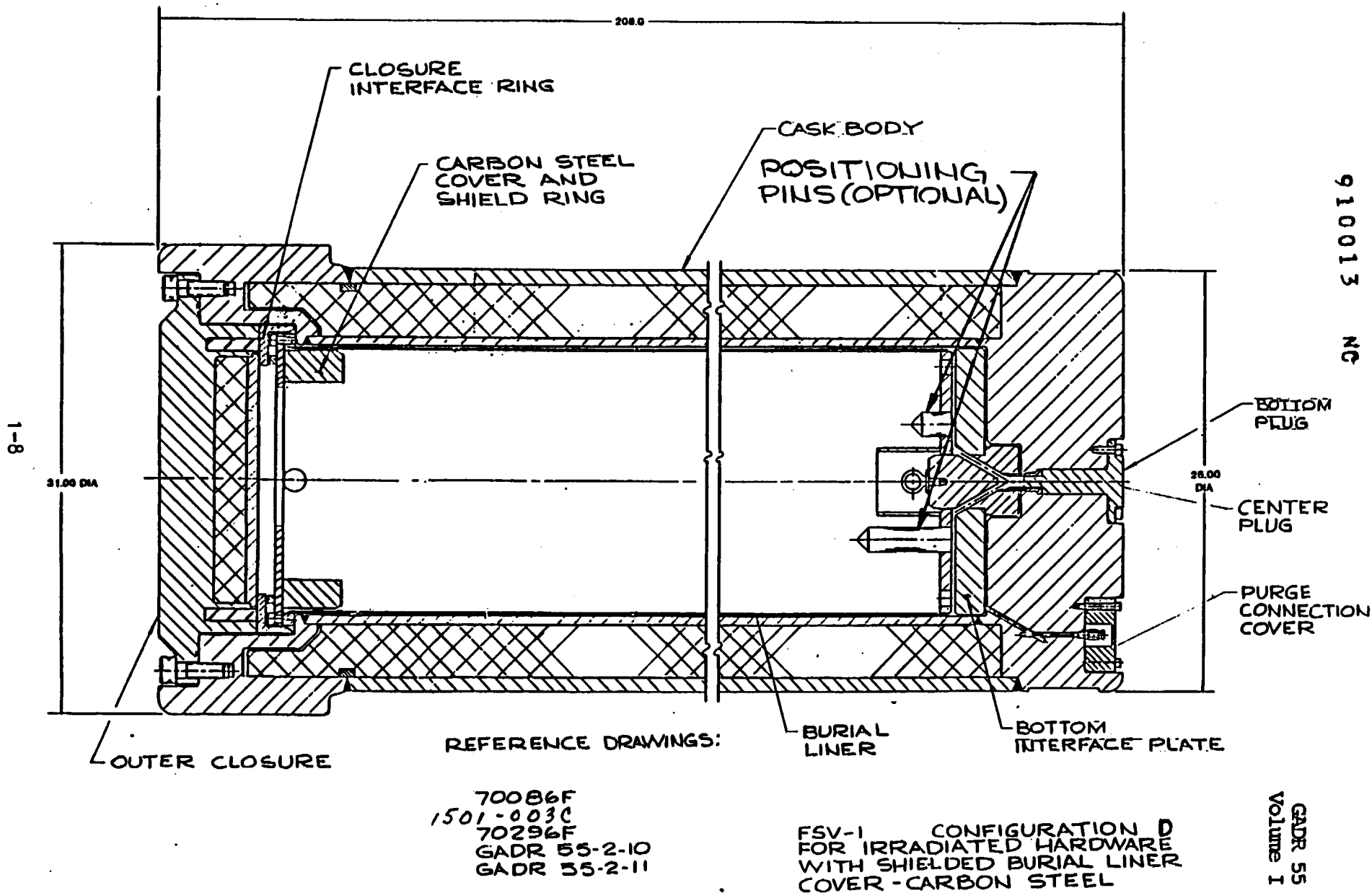


FIGURE 1-4

FIGURE WITHHELD UNDER 10 CFR 2.390

DATE	TIME	DATE	TIME	DATE	TIME
1951	11:15	1951	11:15	1951	11:15
NE NATIONAL LEAD COMPANY SPECIAL SERVICE 1500 BROADWAY		SPENT FUEL CONTAINER & SHIPPING CASK TOP SECTION		1951 11:15 1951 11:15	

FIGURE WITHHELD UNDER 10 CFR 2.390

UNIT 1		UNIT 2		UNIT 3		UNIT 4		UNIT 5		UNIT 6		UNIT 7		UNIT 8		UNIT 9		UNIT 10		UNIT 11		UNIT 12		UNIT 13		UNIT 14		UNIT 15		UNIT 16		UNIT 17		UNIT 18		UNIT 19		UNIT 20		UNIT 21		UNIT 22		UNIT 23		UNIT 24		UNIT 25		UNIT 26		UNIT 27		UNIT 28		UNIT 29		UNIT 30		UNIT 31		UNIT 32		UNIT 33		UNIT 34		UNIT 35		UNIT 36		UNIT 37		UNIT 38		UNIT 39		UNIT 40		UNIT 41		UNIT 42		UNIT 43		UNIT 44		UNIT 45		UNIT 46		UNIT 47		UNIT 48		UNIT 49		UNIT 50		UNIT 51		UNIT 52		UNIT 53		UNIT 54		UNIT 55		UNIT 56		UNIT 57		UNIT 58		UNIT 59		UNIT 60		UNIT 61		UNIT 62		UNIT 63		UNIT 64		UNIT 65		UNIT 66		UNIT 67		UNIT 68		UNIT 69		UNIT 70		UNIT 71		UNIT 72		UNIT 73		UNIT 74		UNIT 75		UNIT 76		UNIT 77		UNIT 78		UNIT 79		UNIT 80		UNIT 81		UNIT 82		UNIT 83		UNIT 84		UNIT 85		UNIT 86		UNIT 87		UNIT 88		UNIT 89		UNIT 90		UNIT 91		UNIT 92		UNIT 93		UNIT 94		UNIT 95		UNIT 96		UNIT 97		UNIT 98		UNIT 99		UNIT 100		UNIT 101		UNIT 102		UNIT 103		UNIT 104		UNIT 105		UNIT 106		UNIT 107		UNIT 108		UNIT 109		UNIT 110		UNIT 111		UNIT 112		UNIT 113		UNIT 114		UNIT 115		UNIT 116		UNIT 117		UNIT 118		UNIT 119		UNIT 120		UNIT 121		UNIT 122		UNIT 123		UNIT 124		UNIT 125		UNIT 126		UNIT 127		UNIT 128		UNIT 129		UNIT 130		UNIT 131		UNIT 132		UNIT 133		UNIT 134		UNIT 135		UNIT 136		UNIT 137		UNIT 138		UNIT 139		UNIT 140		UNIT 141		UNIT 142		UNIT 143		UNIT 144		UNIT 145		UNIT 146		UNIT 147		UNIT 148		UNIT 149		UNIT 150		UNIT 151		UNIT 152		UNIT 153		UNIT 154		UNIT 155		UNIT 156		UNIT 157		UNIT 158		UNIT 159		UNIT 160		UNIT 161		UNIT 162		UNIT 163		UNIT 164		UNIT 165		UNIT 166		UNIT 167		UNIT 168		UNIT 169		UNIT 170		UNIT 171		UNIT 172		UNIT 173		UNIT 174		UNIT 175		UNIT 176		UNIT 177		UNIT 178		UNIT 179		UNIT 180		UNIT 181		UNIT 182		UNIT 183		UNIT 184		UNIT 185		UNIT 186		UNIT 187		UNIT 188		UNIT 189		UNIT 190		UNIT 191		UNIT 192		UNIT 193		UNIT 194		UNIT 195		UNIT 196		UNIT 197		UNIT 198		UNIT 199		UNIT 200		UNIT 201		UNIT 202		UNIT 203		UNIT 204		UNIT 205		UNIT 206		UNIT 207		UNIT 208		UNIT 209		UNIT 210		UNIT 211		UNIT 212		UNIT 213		UNIT 214		UNIT 215		UNIT 216		UNIT 217		UNIT 218		UNIT 219		UNIT 220		UNIT 221		UNIT 222		UNIT 223		UNIT 224		UNIT 225		UNIT 226		UNIT 227		UNIT 228		UNIT 229		UNIT 230		UNIT 231		UNIT 232		UNIT 233		UNIT 234		UNIT 235		UNIT 236		UNIT 237		UNIT 238		UNIT 239		UNIT 240		UNIT 241		UNIT 242		UNIT 243		UNIT 244		UNIT 245		UNIT 246		UNIT 247		UNIT 248		UNIT 249		UNIT 250		UNIT 251		UNIT 252		UNIT 253		UNIT 254		UNIT 255		UNIT 256		UNIT 257		UNIT 258		UNIT 259		UNIT 260		UNIT 261		UNIT 262		UNIT 263		UNIT 264		UNIT 265		UNIT 266		UNIT 267		UNIT 268		UNIT 269		UNIT 270		UNIT 271		UNIT 272		UNIT 273		UNIT 274		UNIT 275		UNIT 276		UNIT 277		UNIT 278		UNIT 279		UNIT 280		UNIT 281		UNIT 282		UNIT 283		UNIT 284		UNIT 285		UNIT 286		UNIT 287		UNIT 288		UNIT 289		UNIT 290		UNIT 291		UNIT 292		UNIT 293		UNIT 294		UNIT 295		UNIT 296		UNIT 297		UNIT 298		UNIT 299		UNIT 300		UNIT 301		UNIT 302		UNIT 303		UNIT 304		UNIT 305		UNIT 306		UNIT 307		UNIT 308		UNIT 309		UNIT 310		UNIT 311		UNIT 312		UNIT 313		UNIT 314		UNIT 315		UNIT 316		UNIT 317		UNIT 318		UNIT 319		UNIT 320		UNIT 321		UNIT 322		UNIT 323		UNIT 324		UNIT 325		UNIT 326		UNIT 327		UNIT 328		UNIT 329		UNIT 330		UNIT 331		UNIT 332		UNIT 333		UNIT 334		UNIT 335		UNIT 336		UNIT 337		UNIT 338		UNIT 339		UNIT 340		UNIT 341		UNIT 342		UNIT 343		UNIT 344		UNIT 345		UNIT 346		UNIT 347		UNIT 348		UNIT 349		UNIT 350		UNIT 351		UNIT 352		UNIT 353		UNIT 354		UNIT 355		UNIT 356		UNIT 357		UNIT 358		UNIT 359		UNIT 360		UNIT 361		UNIT 362		UNIT 363		UNIT 364		UNIT 365		UNIT 366		UNIT 367		UNIT 368		UNIT 369		UNIT 370		UNIT 371		UNIT 372		UNIT 373		UNIT 374		UNIT 375		UNIT 376		UNIT 377		UNIT 378		UNIT 379		UNIT 380		UNIT 381		UNIT 382		UNIT 383		UNIT 384		UNIT 385		UNIT 386		UNIT 387		UNIT 388		UNIT 389		UNIT 390		UNIT 391		UNIT 392		UNIT 393		UNIT 394		UNIT 395		UNIT 396		UNIT 397		UNIT 398		UNIT 399		UNIT 400		UNIT 401		UNIT 402		UNIT 403		UNIT 404		UNIT 405		UNIT 406		UNIT 407		UNIT 408		UNIT 409		UNIT 410		UNIT 411		UNIT 412		UNIT 413		UNIT 414		UNIT 415		UNIT 416		UNIT 417		UNIT 418		UNIT 419		UNIT 420		UNIT 421		UNIT 422		UNIT 423		UNIT 424		UNIT 425		UNIT 426		UNIT 427		UNIT 428		UNIT 429		UNIT 430		UNIT 431		UNIT 432		UNIT 433		UNIT 434		UNIT 435		UNIT 436		UNIT 437		UNIT 438		UNIT 439		UNIT 440		UNIT 441		UNIT 442		UNIT 443	
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FIGURE WITHHELD UNDER 10 CFR 2.390

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FIGURE WITHHELD UNDER 10 CFR 2.390

[illegible]

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

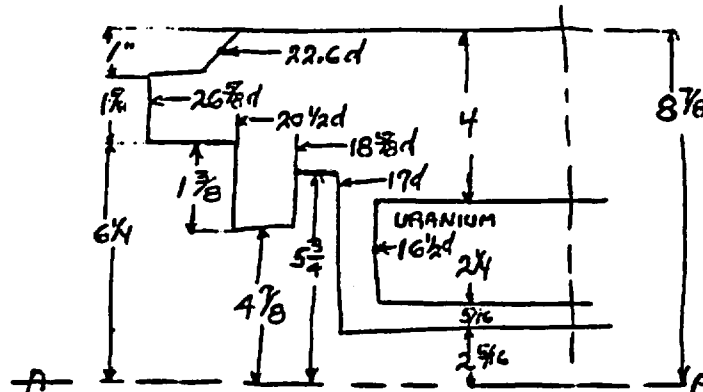
A summary of the weights of model FSV-1 Configurations A, B, C, and D is presented in Table 2-1. Model FSV-1 cask body Configurations A, B, C, D have been evaluated for the normal conditions of transport and the hypothetical accident conditions while containing a total weight of 3720 pounds. This total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

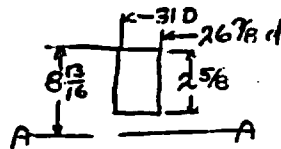
TABLE 2-1

	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

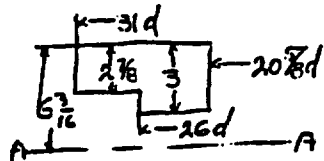
The center of gravity of Model FSV-1 Configurations A, B, C, and D is located 105.5 inches from the bottom of the package.

2.2.1.1 Cask Closure

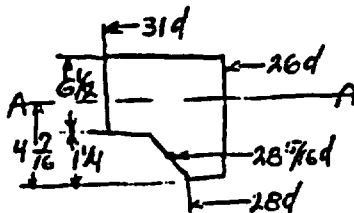
	V	y	Vy
22.6d	$400 \text{ in}^2 \times 1 = 400 \text{ in}^3$	$\times -8 \frac{3}{8} =$	-3350
26 $\frac{5}{8}$ d	$560 \times 1 \frac{7}{8} = 910$	$\times -7 \frac{1}{16} =$	-6430
20 $\frac{1}{2}$ d	$330 \times 1 \frac{3}{8} = 455$	$\times -5 \frac{9}{16} =$	-2530
17 d	$227 \times 3 \frac{7}{16} = 780$	$\times -4 \frac{1}{32} =$	-3140
	$+ 2545 \text{ in}^3$		$-15,450. \text{ in}^4$
-18 $\frac{5}{8}$ d	$-273 \times \frac{7}{8} = -239$	$\times 5 \frac{9}{16} =$	$+1330$
-16 $\frac{1}{2}$ d	$-214 \times 2 \frac{1}{4} = -482$	$\times 3 \frac{3}{4} =$	$+1809$
	-721		$+3139$
	$+1824 \text{ in}^3$		$-12,311 \text{ in}^4$
	$.29$		$.29$
(35)	$W = 530 \text{ lbs}$		$M = -3580 \text{ in. lbs.}$
<u>Uranium</u>			
16 $\frac{1}{2}$ d	$+214 \times 2 \frac{1}{4} = +482$	$\times -3 \frac{3}{4} =$	-1809
	$.663$		$.663$
	330 lbs		$M = -1235 \text{ in. lbs}$
TOTAL WT.	860 LBS.		$M = -4815 \text{ in. lbs.}$
HEAD			

2.2.1.2 Cask Body

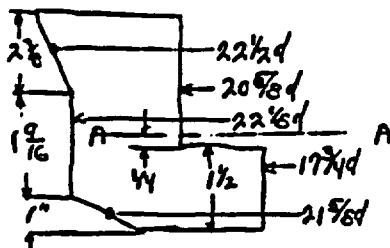
$$\begin{aligned}
 31d & + 752 \text{ in}^2 \\
 26 \frac{7}{8}d & - \frac{568}{184} \times 2 \frac{5}{8} = 483 \text{ in}^2 \times -7 \frac{1}{2} = -3622 \\
 \text{WT} & = \frac{.29}{140 \text{ lbs}} \quad M = \frac{.29}{-1050 \text{ in lbs}}
 \end{aligned}$$



$$\begin{aligned}
 31d & + 752 \text{ in}^2 \times 2 \frac{7}{8} = 2160 \text{ in}^2 \times -4 \frac{3}{4} = -10750. \\
 26d & + 530 \times \frac{1}{8} = \frac{66}{+2226} \times -3 \frac{1}{4} = -216. \\
 -20 \frac{7}{8}d & -335 \times 3 = -1005 \times -4 \frac{11}{16} = +4850 \\
 & +1221 \text{ in}^2 \quad -5616. \\
 \text{WT} & = \frac{.29}{354 \text{ lbs}} \quad M = \frac{.29}{-1628 \text{ in lbs}}
 \end{aligned}$$

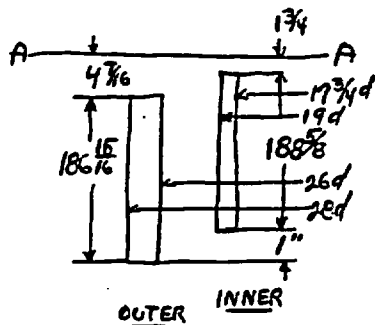


$$\begin{aligned}
 31d & 752 \text{ in}^2 \times 6.5 = 4880 \times -\frac{1}{6} = -305. \\
 28 \frac{7}{16}d & 630 \times \frac{1}{4} = \frac{787}{+5667} \times +3 \frac{13}{16} = +3000. \\
 -26d & -530 \times 7 \frac{3}{4} = -4100 \times +\frac{9}{16} = -2300 \\
 & +1567 \text{ in}^2 \quad +395. \\
 \text{WT} & = \frac{.29}{455 \text{ lbs}} \quad M = \frac{.29}{+114 \text{ in lbs}}
 \end{aligned}$$



$$\begin{aligned}
 22 \frac{1}{2}d & 397 \text{ in}^2 \times 2 \frac{7}{8} = 942 \text{ in}^2 \times -2 \frac{1}{4} = -2120 \\
 22 \frac{5}{8}d & 305 \times 1 \frac{9}{16} = 601 \times -\frac{9}{32} = -169 \\
 21 \frac{5}{8}d & 367 \times 1 = \frac{367}{+1910} \times +1 \frac{1}{4} = +458 \\
 & -1831 \\
 -20 \frac{5}{8}d & -335 \times 3 \frac{1}{6} = -1154 \times -1 \frac{15}{32} = +1670 \\
 -17 \frac{3}{4}d & -247 \times 1 \frac{1}{2} = -370 \times +1 = +370 \\
 & -1524 \quad +1300 \\
 & +386 \quad -591 \\
 \text{WT} & = \frac{.29}{112 \text{ lbs}} \quad W = \frac{.29}{-154 \text{ in lbs}}
 \end{aligned}$$

2.2.1.2 Cask Body (Continued)



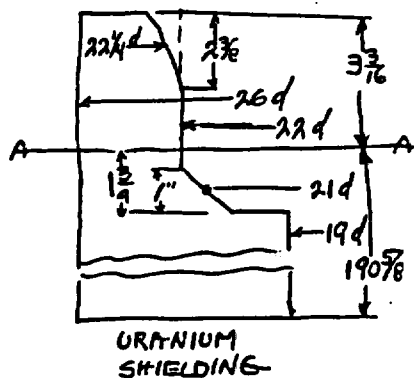
$$\begin{aligned} & \frac{28d}{-26d} = \frac{613 \text{ mi}^2}{83 \text{ mi}^2} \times 186 \frac{15}{16} = 15,500 \text{ mi}^3 \times +97 \frac{29}{32} = +1,511,000. \\ & \text{(out)} \quad \text{let } \frac{.29}{4500} \frac{26}{26} \quad M = \frac{.29}{+439,000} \end{aligned}$$

$$19d \quad 289.5 \text{ m}^2$$

$$-17d \quad -247$$

$$\frac{36.5 \text{ m}^2 \times 188.5}{1990 \text{ lbs}} = 6870 \text{ m}^3 \times +96.06 = +660,000$$

$$(runner) \quad M = \frac{29}{191,200 \text{ lbs}}$$



26d $530 \text{ m}^2 \times 199 \frac{13}{16} = 102,750 \text{ m}^2 \times 93 \frac{13}{32} = +9,635,000.$

-19d $-283.5 \times 188 \frac{7}{8} = -53,501 \times 96 \frac{7}{8} = -5,150,000$

-21d $-346 \times 1 = -346 \times 1 \frac{1}{4} = -432$

-22d $-950 \times 1 \frac{7}{8} = -594 \times -\frac{1}{2} = +19$

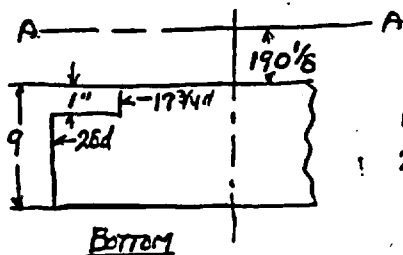
-224d $-389 \times 2 \frac{7}{8} = -925 \times -2 = +1850$

-55,365 -5,148,563

+47,385 +4,481,437

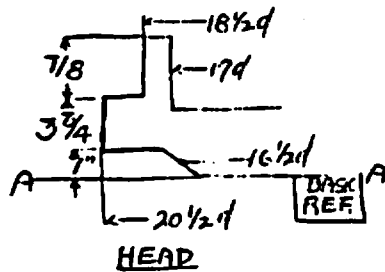
-683 -683

URANIUM m^2 32,400 lbs $M = +3,060,000''$

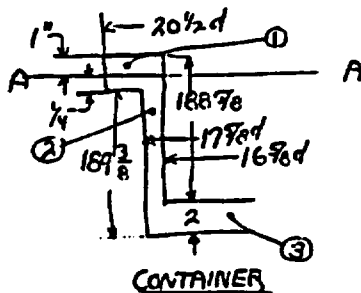


$$\begin{array}{rclcl} 174d & 247m^2 \times 1 & = 247m^3 & \times +190\frac{5}{8} & = +47,100 \\ 28d & 613 \times 8 & = \frac{4904}{+5151} & \times +195\frac{1}{8} & = \frac{+960,000}{+1,007,100} \end{array}$$

$$W = \frac{.29}{1494} \text{ lb} \quad M = \frac{.29}{292,060} \text{ lb}$$

2.2.1.3 Inner Container

$$\begin{array}{rcl}
 18\frac{1}{2}d & 268 \text{ mm}^3 \times \frac{7}{8} = 239 \text{ mm}^3 \times -5\frac{3}{16} & = -1212 \\
 20\frac{1}{2}d & 330 \times 3\frac{3}{4} = 1238 & \times -2\frac{7}{16} = -3560 \\
 16\frac{1}{2}d & 213 \times 1 = 213 & \times -\frac{1}{2} = -107 \\
 & + 1685 \text{ mm}^3 & \\
 & & -4879 \\
 17d & -227 \times 3\frac{7}{8} = -766 & \times -3\frac{15}{16} = +3010 \\
 & + 919 & \\
 & & -1869 \\
 & & -29 \\
 \text{Net} & = 266 \text{ lbs} & M = -542
 \end{array}$$



$$\begin{array}{rcl}
 Wt_1 & = \frac{330}{113} \times 1\frac{1}{4} \times 24 = 41. & \times -\frac{7}{8} = -16. \\
 Wt_2 & = \frac{244}{29} \times 187\frac{7}{8} \times 29 = 1470. & \times +93\frac{15}{16} = +138,000. \\
 Wt_3 & = 244 \times 2 \times 29 = 142. & \times 183\frac{7}{8} = +26,800 \\
 \text{Net} & = 1653 \text{ lbs} & M = +169,000 \text{ in-lb}
 \end{array}$$

CONTENTS OF
CONTAINER

$$300 \# \times 6 = 1800 \text{ lbs}$$

$$\times 93\frac{27}{32} = M = +169,000 \text{ in-lb}$$

2.2.1.4 Weight Summation

		<u>WT.</u>	<u>MOMENT</u>
	CASK	860 LBS	-4815.
	HEAD	140	-1050.
	BODY	354	-1628.
		455	+ 114.
		112	- 154.
		4500	+ 437,000.
		1990	+ 191,200
		32400	+ 3,060,000
		1494	+ 292,060

CASK-TOTAL EMPTY	42,305	+ 3,974,727
CONTAINER	266	- 542
	1653	+ 164,784
CONTENTS	1800	+ 169,000
CASK -TOTAL -LOADED	46,024 LBS	$\times [93.6] = + 4,307,969 \text{ in-LBS}$

2.2.2 Weight Calculation - Configuration B2.2.2.1 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
	Total	143 lb

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
	Total	459 lb

2.2.2.3 Closure Interface Ring

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

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

2.2.3.2 Optional Temporary Shield Tubes

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

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

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/in³

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 Stainless Steel Forgings, Type 304 per ASTM Spec. A-182, Grade F-304

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 Uranium Welds. 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 Stainless Steel Welds (Refs. 2-1 and 2-2). All stainless steel welds will be in accordance with the "Rules for Construcion 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 Seals (Refs. 2-3, 2-4, and 2-19).

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_C 40-44. 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

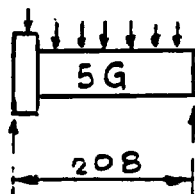
closure end of the cask body. A dedicated lifting device with a ball located on each arm is used to lift the package. The sockets are removable and can be replaced if damaged in any way.

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

$$WT = 46024 \text{ lb}$$

$$W = 5(46024) = \underline{230,120 \text{ lb}}$$

$$\text{Max at center} = \frac{Wl^2}{8} = \frac{230120}{8} (208) = 6,000,000 \text{ in. lb}$$

Assume only outer shell is stressed

$$OD = 28" \quad ID = 26"$$

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

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

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

2.5.2 External Pressure

25 psig Roark XVI Case 3c (Stability)



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. $^{\circ}$ 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.

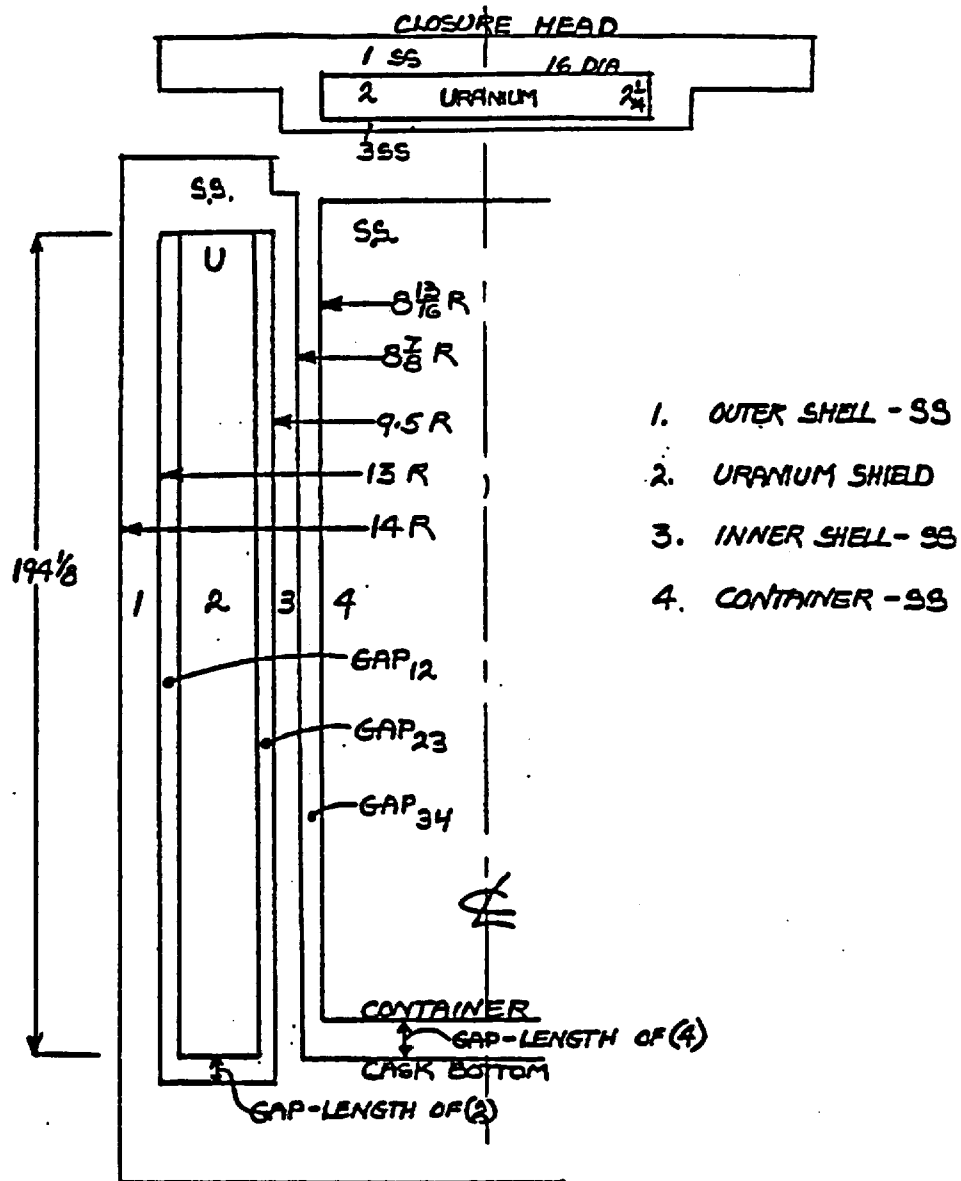


FIGURE 2-1 Location of Gaps and Materials for Model FSV-1
in Configurations A, B, C and D

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)

<u>A</u>	<u>B</u>	<u>Temp °F</u>
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

- 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	} Interferences - to be prevented by suitable mfg. clearances.
Gap 23 = -0.015 from case 2	
Gap 34 = -.0367 from case 3	
Length container = -0.309 from case 6	

Case 1

$$T_1 = 1120^\circ \quad T_2 = 430^\circ \quad T_0 = 70^\circ \quad T_u = 370^\circ$$

Gap 12 Assume $T_2 = T_1$ for max. diff.

$$\begin{aligned} dr &= (13'')(T_1 - T_0)(10.05)10^{-6} - (13'')(T_2 - T_0)(6.5)10^{-6} \\ &= 0.137 - 0.0304 = + \underline{0.1066''} \text{ SS} > \text{U gap} \end{aligned}$$

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

$$\begin{aligned} dr &= (9.5'')(T_2 - T_0)(6.5)10^{-6} - (9.5'')(T_1 - T_0)(9.6)10^{-6} \\ &= 0.0222 - 0.328 = - \underline{0.0106} \text{ U} < \text{SS interference} \end{aligned}$$

Gap 34 Assume $T_4 = 370^\circ$ from 2322 Btu/hr fuel rate

$$dr = (8-7/8")(T_3 - T_0)(9.6)10^{-6} - (8-13/16")(T_4 - T_0)(9.5)10^{-6}$$

$$= 0.0306 - 0.0251 = + 0.0055 \text{ gap}$$

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

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

$$= 2.05 - 0.455 = 1.595" \text{ expansion SS} > \text{U gap}$$

Container shows gap.

Case 2

$$T_1 = 220^\circ \quad T_3 = 340^\circ \quad T_0 = 70^\circ \quad T_4 = 370^\circ$$

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

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

$$= 0.0183 - 0.0228 = -0.0045 \text{ U} > \text{SS interference}$$

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

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

$$= 0.00925 - 0.0243 - 0.015 \text{ U} < \text{SS interference}$$

Gap 34 Assume $T_4 = 370^\circ$

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

$$= 0.0227 - 0.025 - 0.0023 \text{ interference}$$

Case 3 Cask at original dimen. $T_4 = 241^\circ \quad T_0 = 70^\circ \quad T_3 = 70^\circ$

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

$$= - 0.0367" \text{ container increase - interference}$$

Length Container

$$d1 = (187 - 5/8)(T_s - T_o) (9.7)10^{-6}$$

$$= -0.309 \text{ container increase interference}$$

Case 4

$$T_1 = 70^\circ \quad T_2 \text{ assumed} = T_s = 450^\circ \quad T_o = 70^\circ \quad T_s = 240^\circ$$

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

$$= 0 - 0.0322 = -0.0322 \text{ SS} < \text{U interference}$$

$$\text{Gap } 23 = 0$$

$$\text{Gap } 34 = \text{negligible}$$

Length Container - in time same as case 3

Case 5

$$T_1 = T_2 = T_s = 40^\circ \quad T_o = +70$$

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

$$= -0.013 + 0.0093 = 0.0037 \text{ SS} < \text{U interference}$$

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

$$= -0.0068 - 0.0095 = +0.0027 \text{ gap}$$

Case 6

$$T_1 = T_2 = -40^\circ \quad T_s = T_s = 240^\circ \quad T_o = 70^\circ$$

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

$$= -0.0068 - 0.0147 = -0.0218 \text{ interference}$$

2.6.1.6 Analysis of Uranium Shielding in Cask ClosureCase 1

Assume same gradients as given for cask.

$$T_1 = 1120^\circ \quad T_2 = 340^\circ \quad T_0 = 70^\circ$$

$$\Delta t_s = 1050^\circ \text{ for SS space expansion}$$

$$\Delta t_u = 430 - 70 = 360^\circ \text{ for U expansion from original machined dimension}$$

$$\begin{aligned} \text{Length } dl &= (16") [\Delta t_s 10.05 - \Delta t_u 6.5] 10^{-6} \\ &= 16 [1050(10.05) - 360 (6.5)] 10^{-6} \\ &= 16 (11000 - 2340) 10^{-6} = 0.1384 \text{ gap} \end{aligned}$$

Thickness - also negligible

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

Case 2

$$T_1 = 220^\circ \quad T_2 = 340^\circ \quad T_0 = 70^\circ$$

$$\Delta t_s = 150^\circ$$

$$\Delta t_u = 340 - 70 = 270^\circ$$

$$\begin{aligned} \text{Length } dl &= 16 [\Delta t_s (9.47) - \Delta t_u (6.5)] 10^{-6} \\ &= 16 [150 (9.47) - 270 (6.5)] 10^{-6} \\ &= 16 (1420 - 1750) 10^{-6} \end{aligned}$$

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

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

Case 3

No diff. exp.

Case 4

No diff exp.

Case 5

$$\text{Temp. drop} = -40 - (+70) = -110^{\circ}$$

$$\begin{aligned} \underline{\text{Length}} \text{ dl} &= 16 (-110) (9.11 - 6.5) 10^{-6} \\ &= -4600 (10^{-6}) = 0.0046 \text{ interference} \end{aligned}$$

Case 6

No diff exp.

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 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.


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

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

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

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

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



$l = 208 \text{ in.}$

$$\begin{aligned} \text{Frequency} &= \frac{3.55}{\sqrt{\frac{5}{384} \frac{Wl^3}{EI}}} = \frac{3.55}{\sqrt{\frac{5}{384} \frac{(46,500)(208)^3}{29 \times 10^6 (25,321)}}} \\ &= \underline{41.2} \quad \text{cycles per second (cps)} \end{aligned}$$

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 Internal Pressure 50 psig Roark XIII - Case 1

See section 2.5.2 for inner container description

Hoop Stress

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

Meridional Stress

$$S_1 = \frac{PR}{2t} = \frac{862}{2} = 431 \text{ psi}$$

Radial Displacement

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

$$= \frac{8.62}{30(10^6)} [862 - 0.3 (431)2] = \frac{6300}{30(10^6)} = 0.00021 \text{ inches ok}$$

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

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

$$\rho = \frac{1b/in^3}{386} \quad v_1^2 = [44 \text{ ft/sec} \times 12]^2 = 278,784 \text{ (in/sec)}^2$$

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	$\rho = 0.097/386 = 0.000251$	$\rho v_1^2 = 70$
Steel (incl SS)	$\rho = 0.290/386 = 0.0007512$	= 209
Copper	$\rho = 0.332/386 = 0.00086$	= 240
Lead	$\rho = 0.41/306 = 0.001062$	= 296
Uranium	$\rho = 0.683/386 = 0.001769$	= 493

Typical Dynamic Properties

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

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

$$\epsilon_1 = \frac{P_1 - 40,000}{825,000}$$

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 End Drop. 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

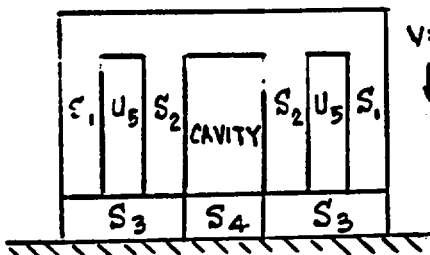
U = Uranium cylinder

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 , and S_2

S_4 = part of S.S. plate stressed by impact from container, etc



<u>Area</u>	<u>Weight</u>
S ₁ 86.3 in. ²	5,125 lb (incl. 635 lb head)
S ₂ 36.8 in. ²	2,260 lb (incl. 270 lb head)
U _s $\frac{250.9 \text{ in.}^2}{374 \text{ in.}^2}$	$\frac{32,400 \text{ lb}}{39,785 \text{ lb}}$
S _s 378.5 in. ²	990 lb
S _c 248.5 in. ²	5,249 lb (1,919 container, 1,800 contents)
Total	<u>46,024 lb</u>

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 S₁, U_s, and S₂, the assumption of striking a rigid mass is conservative. Actually, they strike against an intermediate mass S_s, with some consequential reduction in stress.

S_s, on its under-surface, does strike a rigid mass, and counts S₁, U_s, and S₂ only as added load, considered as an equivalent weight of steel, added to the height of S_s as a cylinder.

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

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

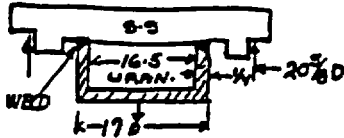
For S_1 $F_1 = 86.3 \text{ in}^2 \times 53,800 \text{ psi} = 4,640,000 \text{ lb}$
 S_2 $F_2 = 36.8 \text{ in}^2 \times 53,800 \text{ psi} = 1,975,000 \text{ lb}$
 U_1 $F_3 = 250.9 \text{ in}^2 \times 90,000 \text{ psi} = 22,581,000 \text{ lb}$

G's Developed

$G = F/wt$

S_1 $4,640,000/5125 = 905 \text{ G}$
 S_2 $1,975,000/2260 = 875 \text{ G}$
 U_1 $22,581,000/32,400 = 697 \text{ G}$
 S_3 $29,196,000/39,785 = 735 \text{ G}$ 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

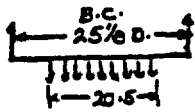
$$= \pi/4 (17^2 - 16.5^2) = 12.5 \text{ in.}^2$$

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

Stress in weld = $264,000/12.5 = \underline{21,100 \text{ psi}}$

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

$$\begin{aligned} \text{Max } S_r = S_t &= \frac{-3W}{2\pi m t^2} \left[m + (m+1) \log_e a/r_o - (m-1) \frac{r_o^2}{4a^2} \right] \\ &= \frac{-3(632,000)}{628(10/3)4^2} \left\{ 10/3 + 13/3 \log_e \frac{12.56}{10.25} - 7/3 \frac{10.25^2}{4 \times 12.56^2} \right\} \\ &= 5650 [10/3 + 13/3 (0.230) - 3.9] = 5650 (0.31) \\ &= \underline{1750 \text{ psi}} \text{ ok } 30,000 \text{ psi yield strength} \end{aligned}$$

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

$$\begin{aligned} \text{Max } S_r &= \frac{3W}{2\pi t^2} \left(1 - \frac{r_o^2}{2a^2} \right) = \frac{3(632,000)}{628(2.625)^2} \left(1 - \frac{10.25^2}{2(12.56)^2} \right) \\ &= 43,700 (1 - 0.334) = \underline{29,200 \text{ psi}} \text{ ok } 30,000 \text{ psi yield strength} \end{aligned}$$

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

$$\begin{aligned}\text{Area in shear} &= 2\pi rt = \pi (20.625) 2.625 \\ &= 170.5 \text{ in.}^2\end{aligned}$$

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

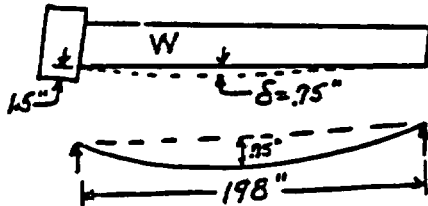
$$\begin{aligned}F_{SU} &= 0.60 (70,000 \text{ psi}) \\ &= 42,000\end{aligned}$$

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

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.



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

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

$$(W \cdot g) = 215 I$$

Let $W = 46,024 \text{ lb}$ = 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.

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

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

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

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

$$\text{Now } (W \cdot g) = 215 (\Sigma I)$$

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

$$g = \frac{3,880,000}{46,024} = 83.8 \text{ g}$$

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

and

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

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

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

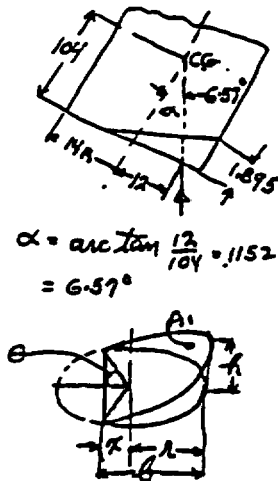
All these stresses are momentary peaks and local S_3 for uranium is less than that sustained in the test.

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.



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

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

$$\frac{h}{b} = 0.1152$$

$$r^3 = 14^3 = 2744$$

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

Assume center of impact 2 in. from edge, at $R = 12$ in.

Total weight impacting on SS bottom plate = 46024 lb

KE = $(46024)(360 \text{ in.}) = \underline{16,600,000 \text{ in. lb}}$

Mean flow stress for SS. (Approximately)

$$P = 53,800 \text{ psi}$$

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

This is the calculated ungula volume

$$\therefore \text{Vol} = 317 [] = 308 \quad [] = \frac{308}{317} = 0.972$$

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

and $\theta = 1.745$ radians

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

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

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

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

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

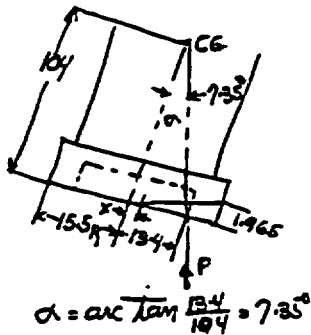
$$F_{\max} = A_1 P_1 = (375.9)(53,800) = \underline{20,200,000 \text{ lb}}$$

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

Component along axis

$$(438 \text{ G})(\cos 6.57^\circ) = 438 (0.99343) = \underline{435 \text{ G}}$$

Top Corner (Method similar to bottom impact analysis)



Assume center of impact at edge of closure, $R = 13.4$ in.

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

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

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

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

$$\text{Let } \theta = 89^\circ, \text{ then } [] = 0.99985 - 0.333 - (1.5533 \times 0.0174) = 0.639$$

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

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

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

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

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

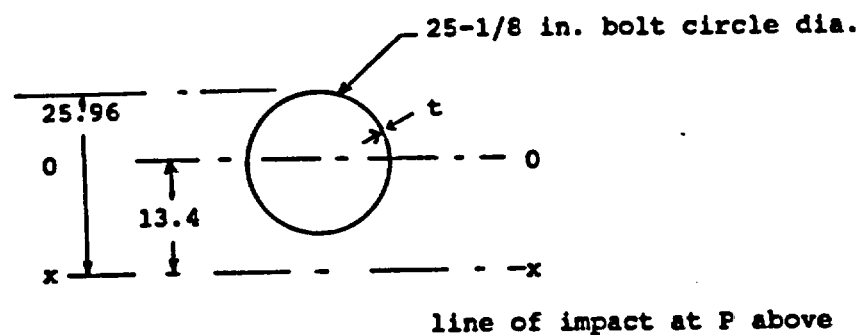
$$F_{\max} = A_1 P_1 = (369) 53,800 = 19,850,000 \text{ lb}$$

$$= F/WT = 19,850,000/46,024 = 430 \text{ G vertically}$$

Component along axis

$$(430\text{G})(\cos 7.35^\circ) = 430 (0.99178) = 426.5\text{G}$$

BOLT LOADS



Let A = Tensile area of 24 bolts $(1-1/4 - 7) = 24 \times 0.9684 \text{ in.}^2$

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

= 25-1/8 in.

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

$$I_{\infty} = \frac{\pi}{4} (R^4 - r^4) = \frac{\pi}{4} (12.71^4 - 12.42^4) = 1808 \text{ in.}^4$$

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

Weights 860 closure Moment about axis xx
 1919 container $M = (4579 \text{ lb})(426.5 \text{ G}) \times 13.4 \text{ in.} =$
 1800 contents 26,169,443 in. lb
 4579 lb Total

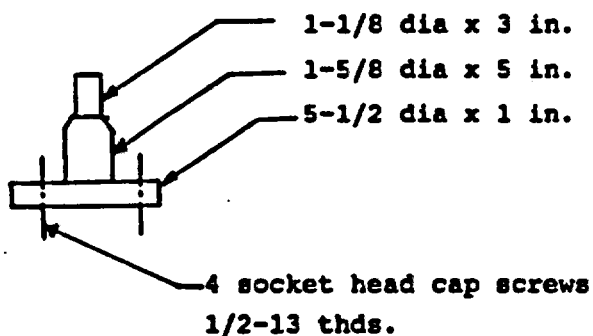
$$S_x = \frac{My}{I_{xx}} = \frac{26,169,433 \times 25.96}{5980} = \underline{113,605 \text{ psi}}$$

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

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

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

Bottom Plug



$$\begin{aligned} \pi/4(1.125)^2 \times 3 &= 3.0 \text{ in.}^3 \\ \pi/4(1.625)^2 \times 5 &= 10.4 \text{ in.}^3 \\ \pi/4(5.5)^2 \times 1 &= \underline{23.8 \text{ in.}^3} \\ 37.2 \text{ in.}^3 \times .284 &= \\ \underline{10.57 \text{ lbs}} \end{aligned}$$

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.

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

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.

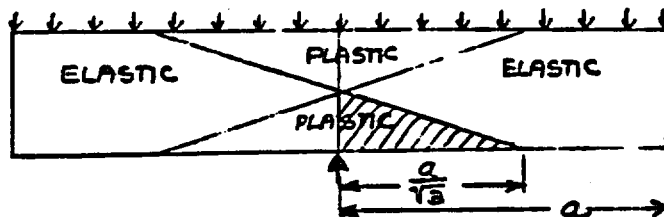
Energy is, however, absorbed in the three concentric shells of the cask in bending, with the possible formation of a plastic hinge.

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
59.1	43,500 in. lb	3-1/2 in. uranium
<u>6.8%</u>	<u>5,030 in. lb</u>	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



$$\begin{aligned} a &= \frac{190}{2} \\ r_0 &= 13 \\ r_1 &= 9.5 \\ H &= \frac{9}{\sqrt{3}} = 55 \end{aligned}$$

The plastic volume = 4 x volume of shaded ungula

$$\begin{aligned} &= 4 \left(\frac{2}{3} \right) (r_0^2 - r_1^2) H = 4 \left(\frac{2}{3} \right) (13^2 - 9.5^2) 55 \\ &= 11,520 \text{ in.}^3 \end{aligned}$$

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

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

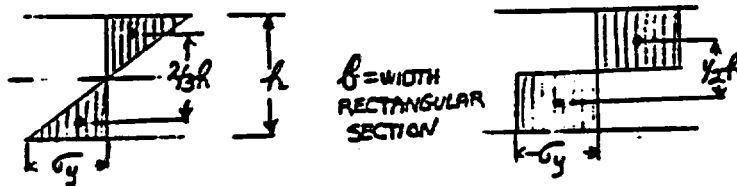
The total energy of the 40-in. fall is

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

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

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

$$\therefore M_P = \frac{3}{2} M_E$$

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

For the circular section of the uranium cylinder

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

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

$$\begin{aligned} M_p &= \sigma_p Z_p \\ &= (30,000)(1860) = 55,800,000 \text{ in. lb} \end{aligned}$$

But

$$U_B = M\theta$$



or

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

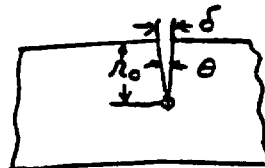
$$\theta = \frac{1,514,400}{55,800,000} = 0.0271 \text{ radians} = \underline{1.56^\circ}$$

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

$$(0.59 (1.56^\circ)) = 0.925^\circ \theta$$

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 40% elongation o.k. No Rupture

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 Evaluation of FSV-1 Configuration C

2.7.3.1 Discussion. 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 Evaluation Criteria. 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: ($P_1 + P_b$) 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 quadrilateral 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.

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

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

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 from 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

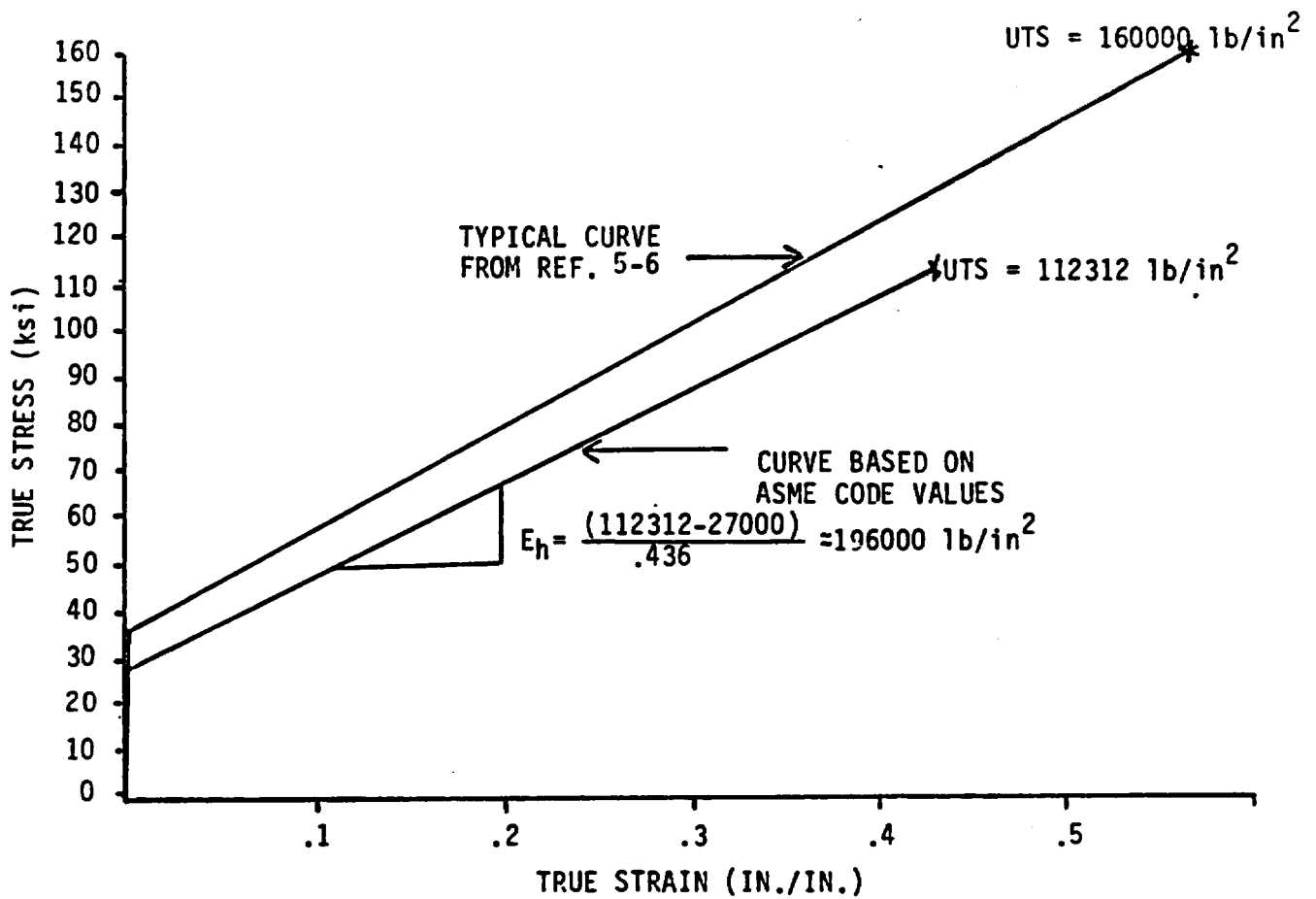


FIGURE 2-2 TRUE STRESS VS. TRUE STRAIN
FOR 304 STAINLESS STEEL

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
<u>Carbon Steel ASTM A36</u>		
Young's Modulus	$27.8 \times 10^2 \text{ lb/in.}^2$	ASME Code Table I-6.0
Poisson's Ratio	0.29	Mark's Handbook page 5-6
Mass Density	$0.000732 \text{ lb-sec}^2/\text{in.}^4$	Mark's Handbook page 6-10
Yield Strength	34320 lb/in.^2	Figure 2-3
Hardening Modulus	224720 lb/in.^2	Figure 2-3
<u>Stainless Steel 304</u>		
Young's Modulus	$27.9 \times 10^6 \text{ lb/in.}^2$	ASME Code Table I-6.0
Poisson's Ratio	0.305	Mark's Handbook page 5-6
Mass Density	$0.000749 \text{ lb-sec}^2/\text{in.}^4$	Mark's Handbook page 6-10
Yield Strength	$27,000 \text{ lb/in.}^2$	Figure 2-2
Hardening Modulus	$196,000 \text{ lb/in.}^2$	Figure 2-2

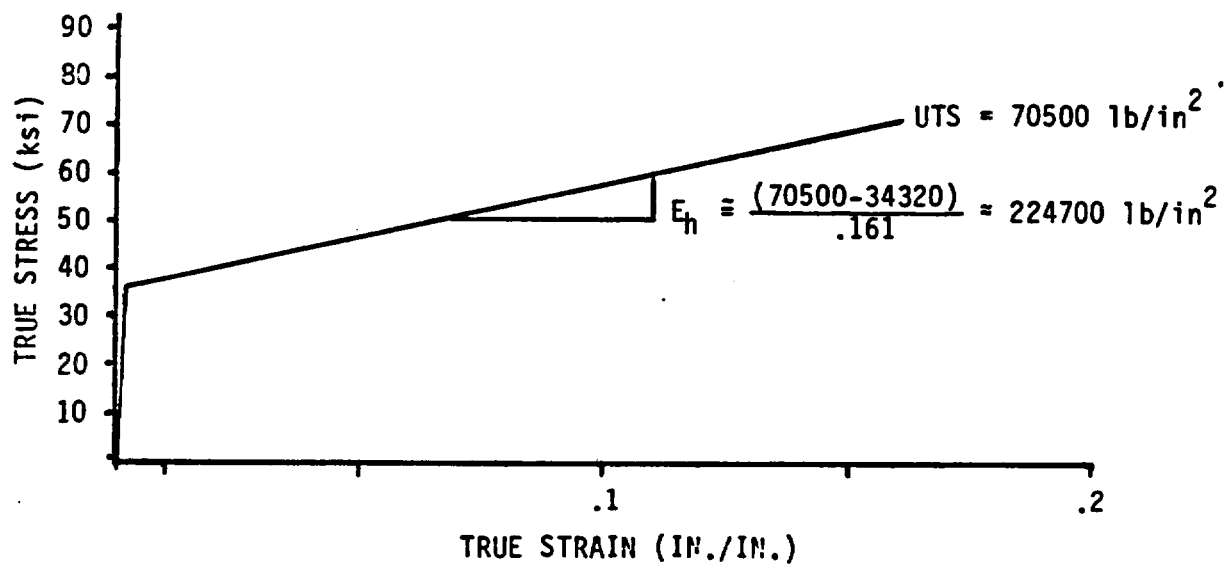


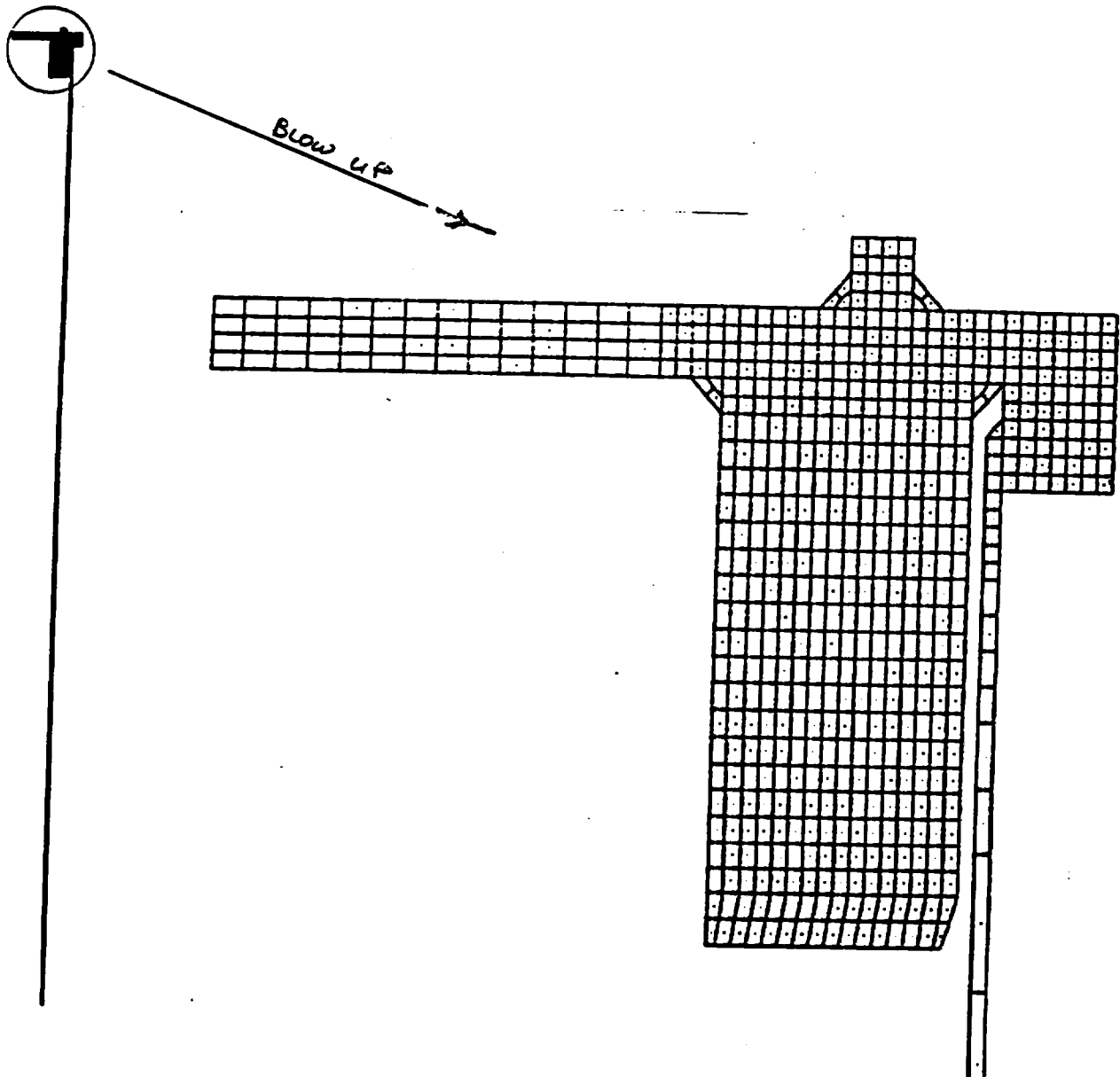
FIGURE 2-3 TRUE STRESS VS. TRUE STRAIN CURVE
FOR ASTM A-36 CARBON STEEL

2.7.3.5 Finite Element Model. 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. groove 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

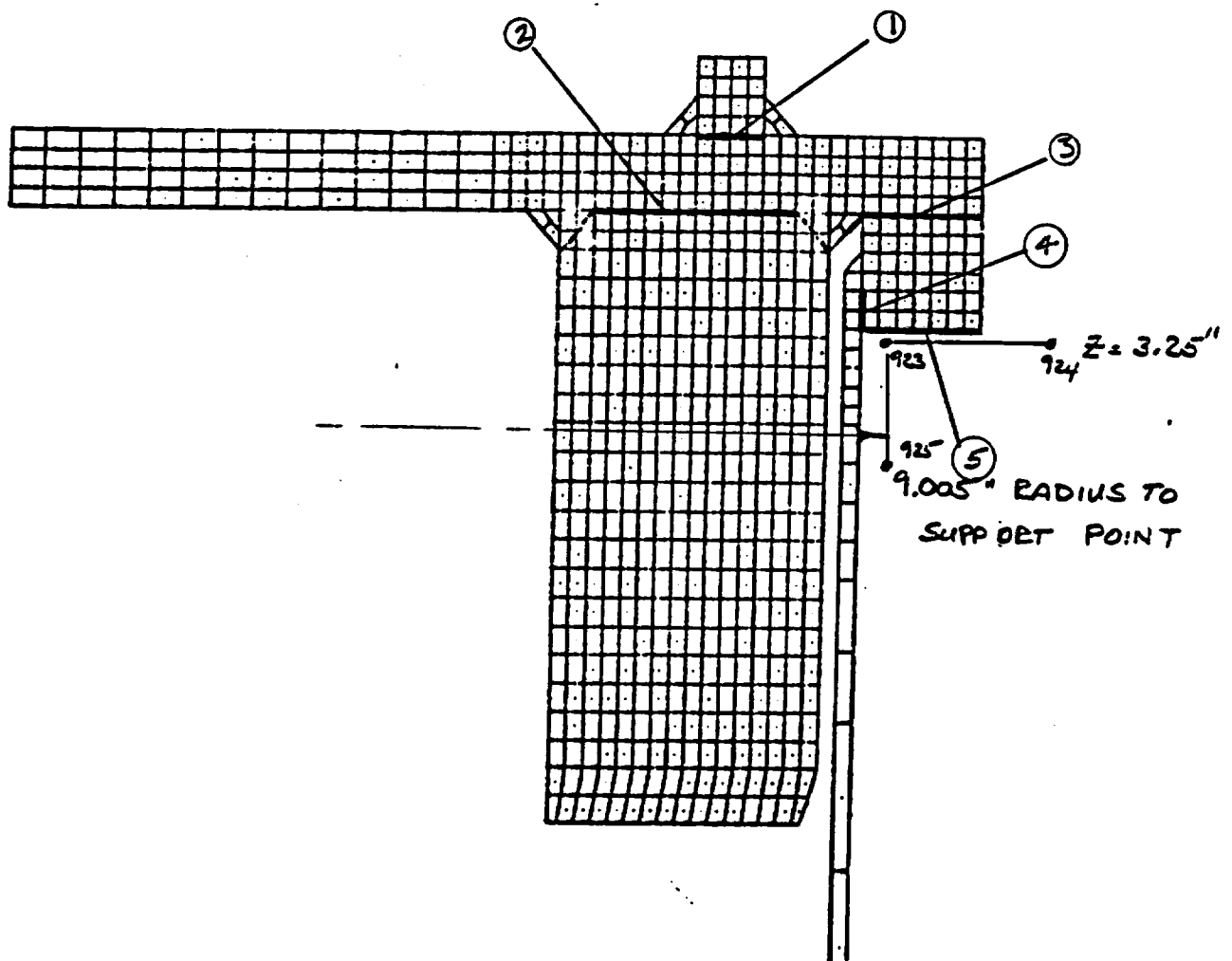


FIGURE 2-5 SLIDING INTERFACES

2.7.3.6 Results of Analysis. 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 velocity 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:

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

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

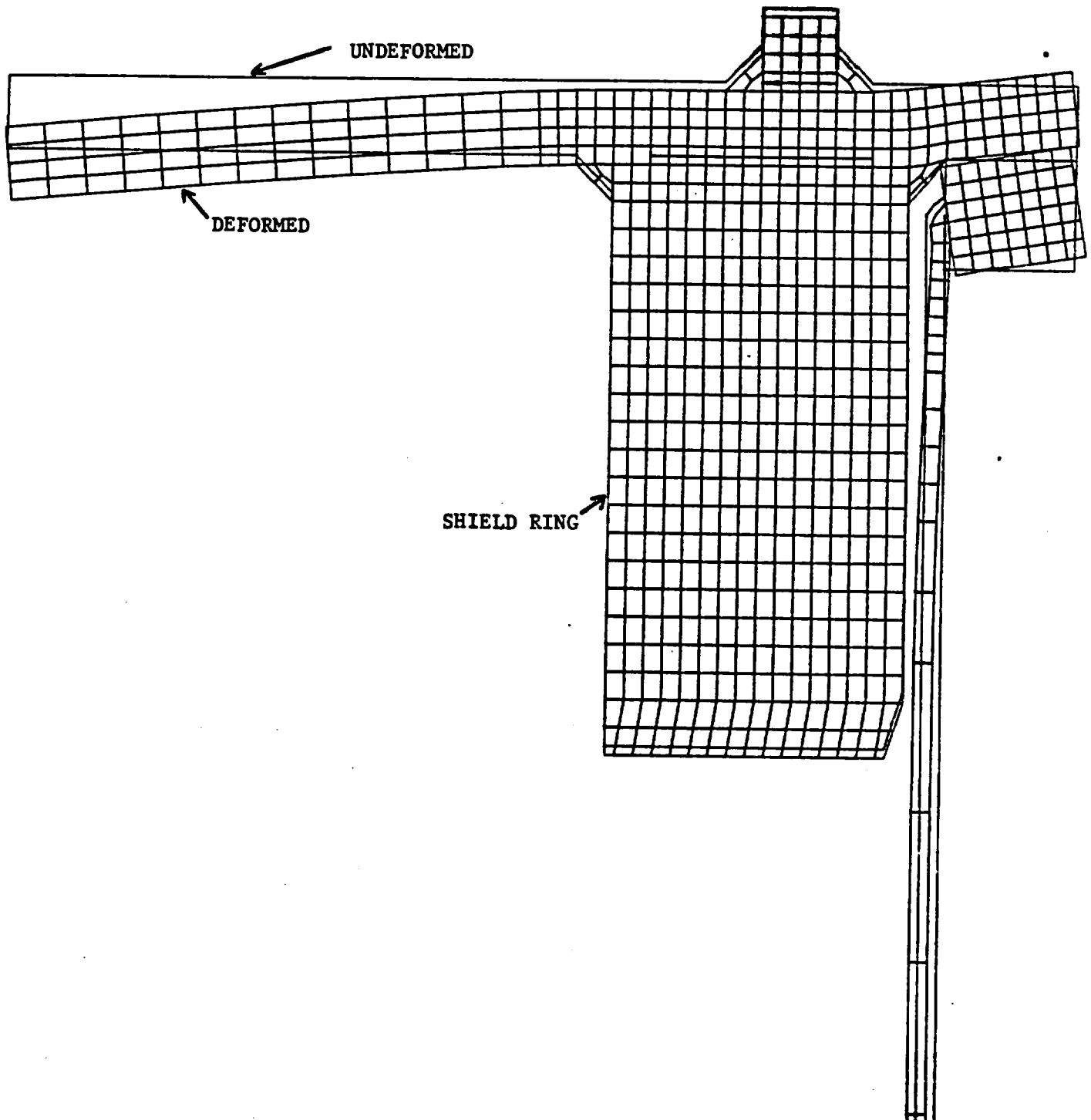


FIGURE 2-6 DEFORMED SHAPE AT .84 MILLISECONDS

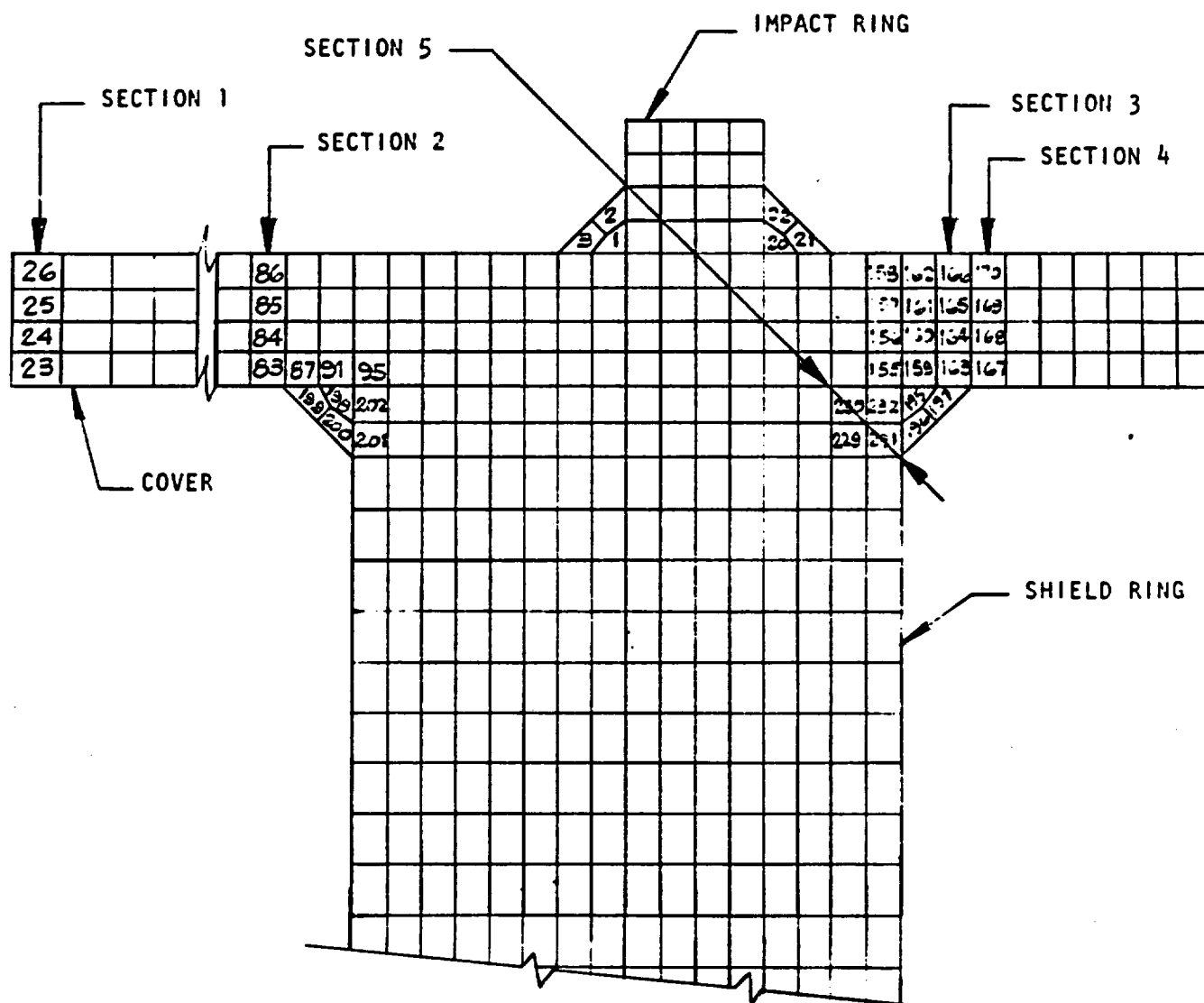
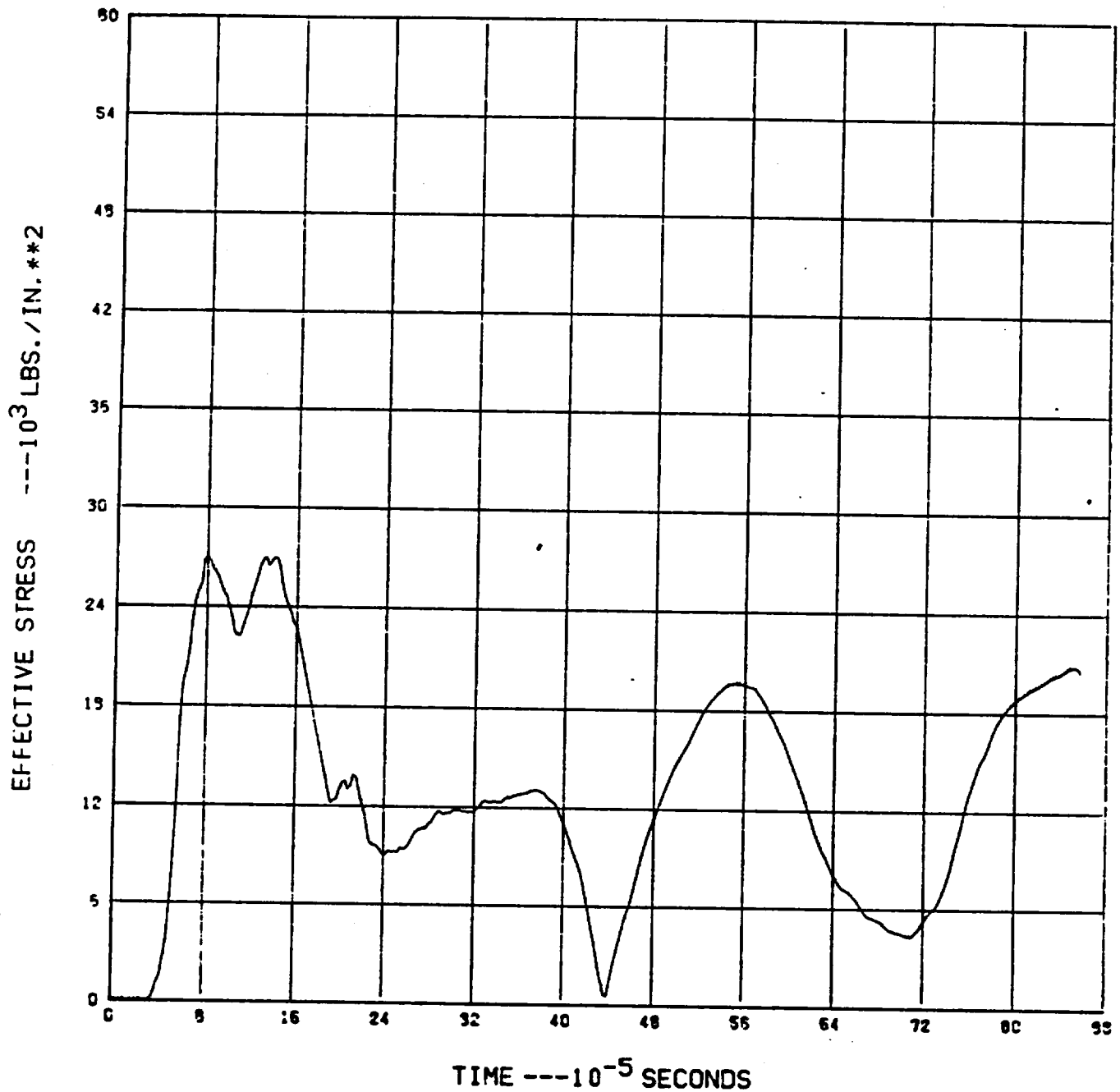


FIGURE 2-7 LOCATION OF SECTIONS USED
FOR PRIMARY STRESS EVALUATION

EFFECTIVE PRIMARY MEMBRANE STRESS AT SECTION 1

FIGURE 2-8 SECTION 1
PRIMARY MEMBRANE STRESS

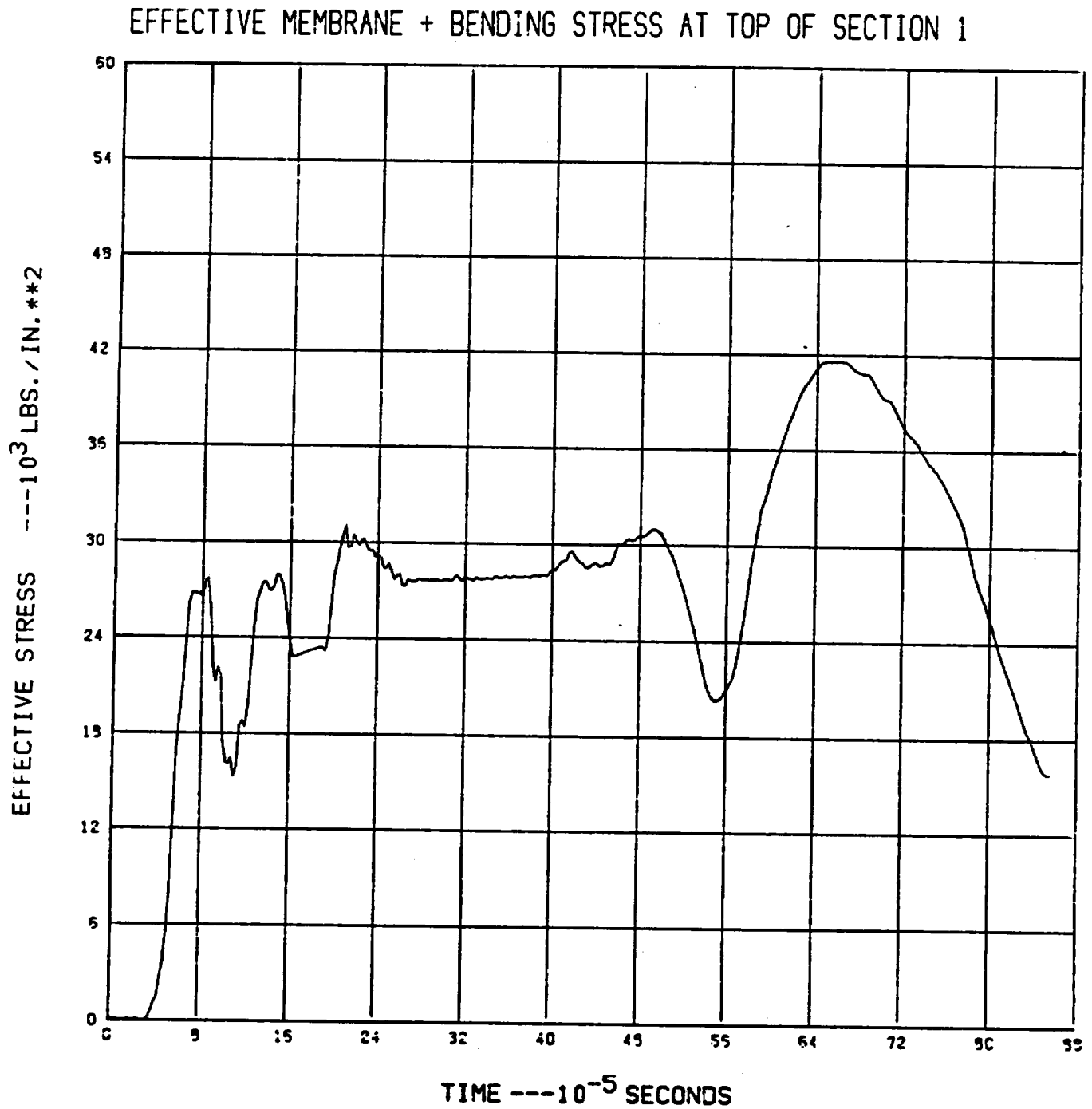


FIGURE 2-9 SECTION 1
PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT TOP OF PLATE

EFFECTIVE MEMBRANE + BENDING STRESS AT BOTTON OF SECTION 1

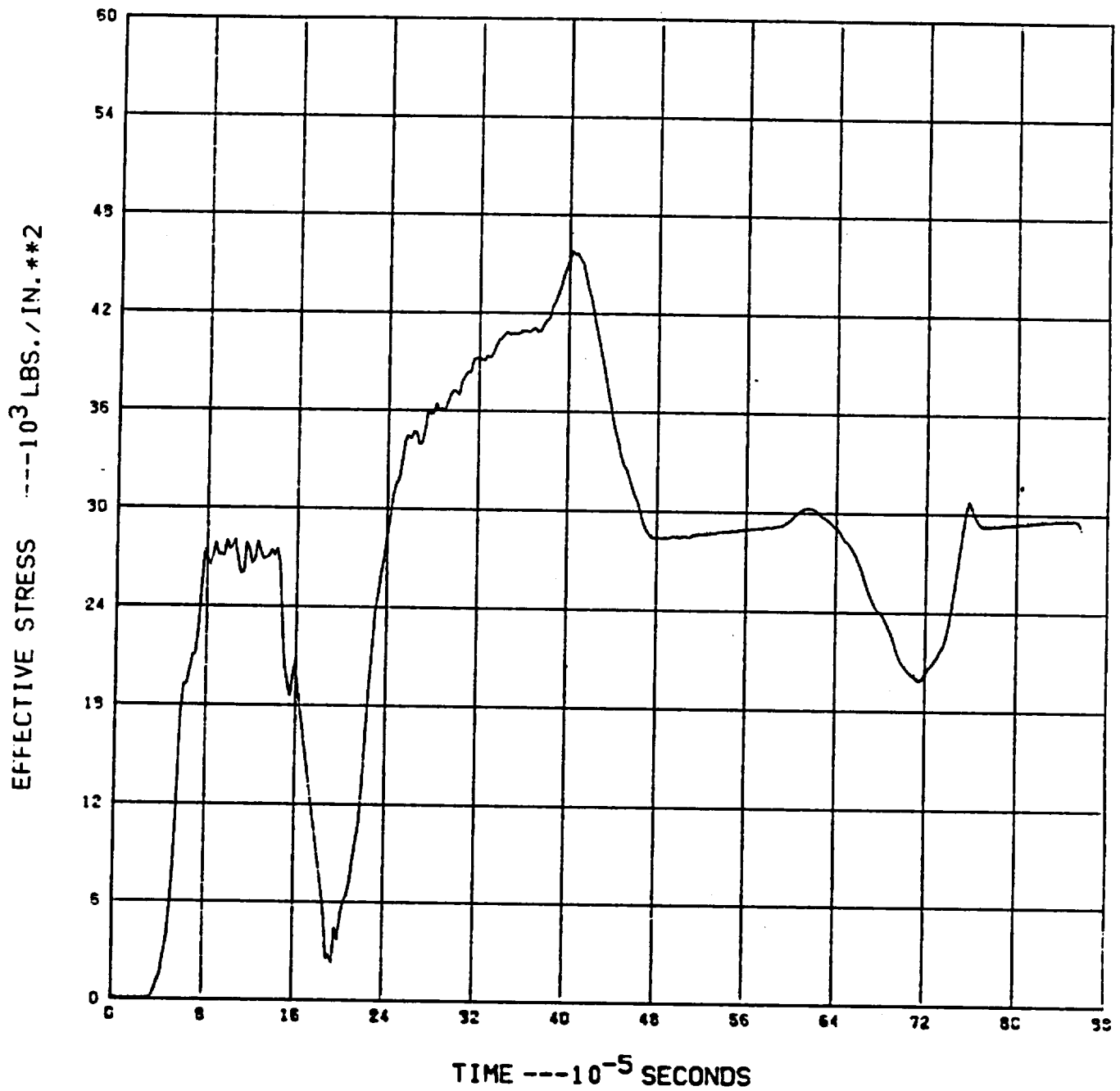
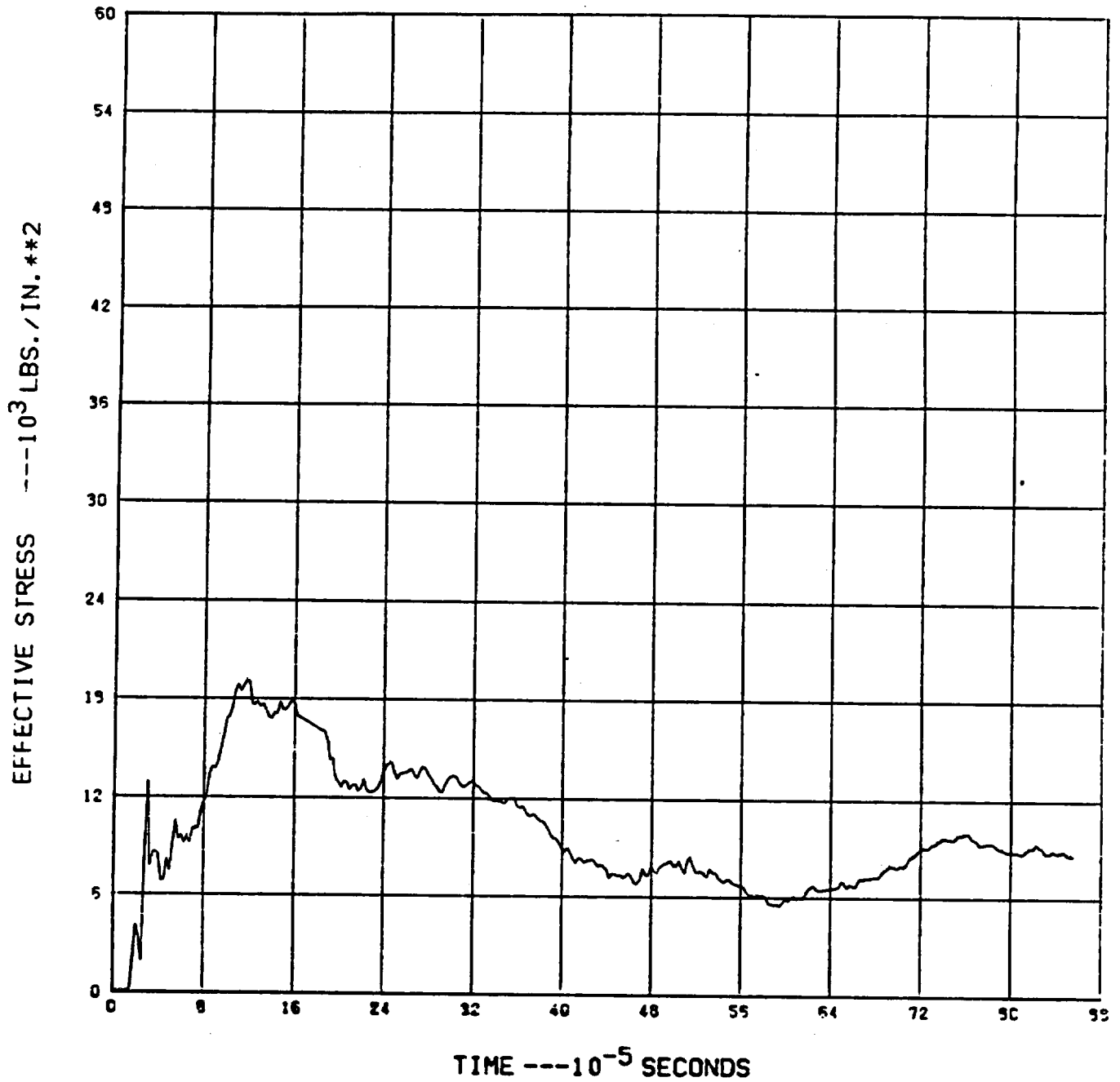


FIGURE 2-10 SECTION 1
PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT BOTTOM OF PLATE

EFFECTIVE PRIMARY MEMBRANE STRESS AT SECTION 2

FIGURE 2-11 SECTION 2
PRIMARY MEMBRANE STRESS

EFFECTIVE MEMBRANE + BENDING STRESS AT TOP OF SECTION 2

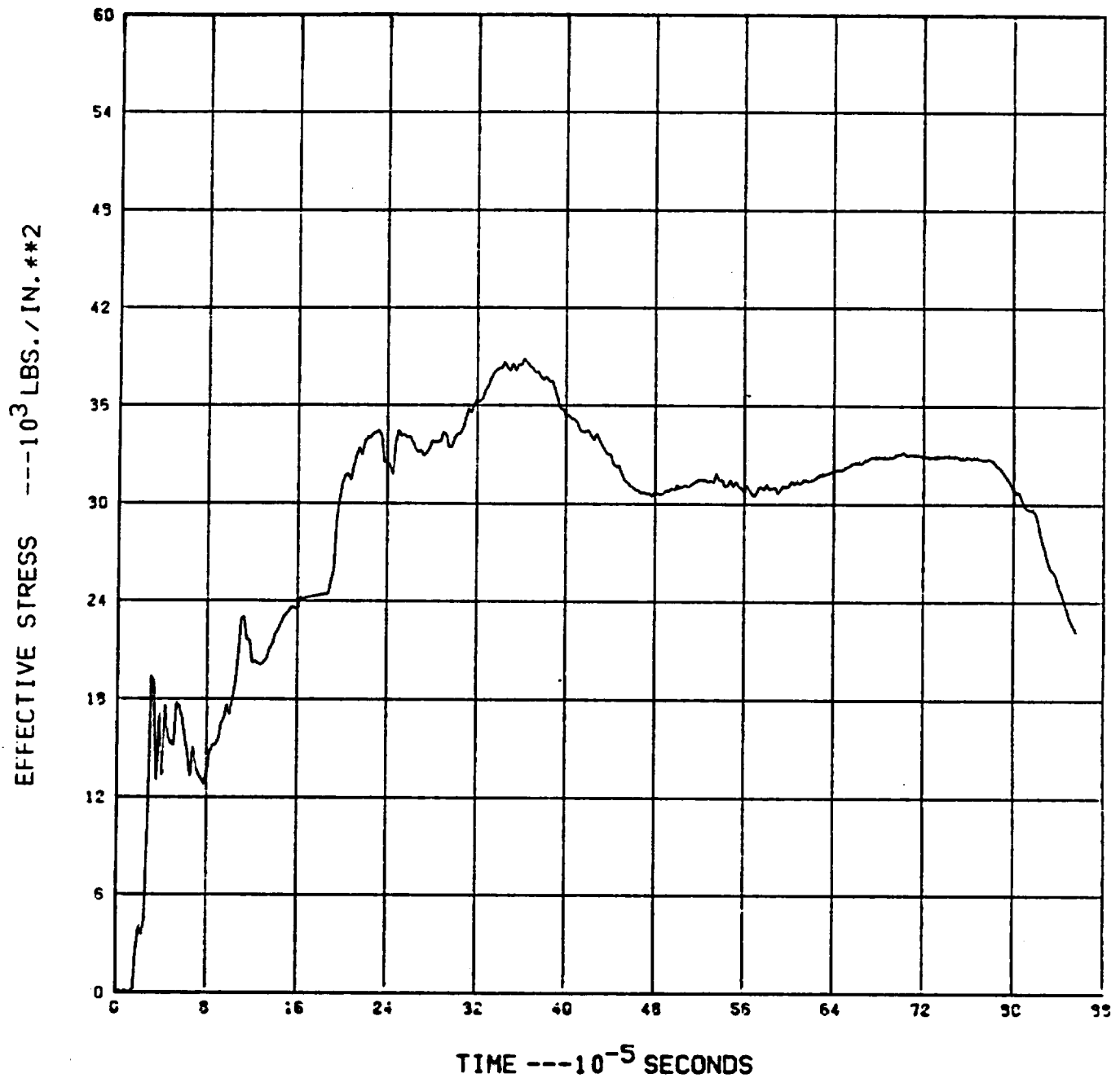


FIGURE 2-12 SECTION 2
PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT TOP OF PLATE

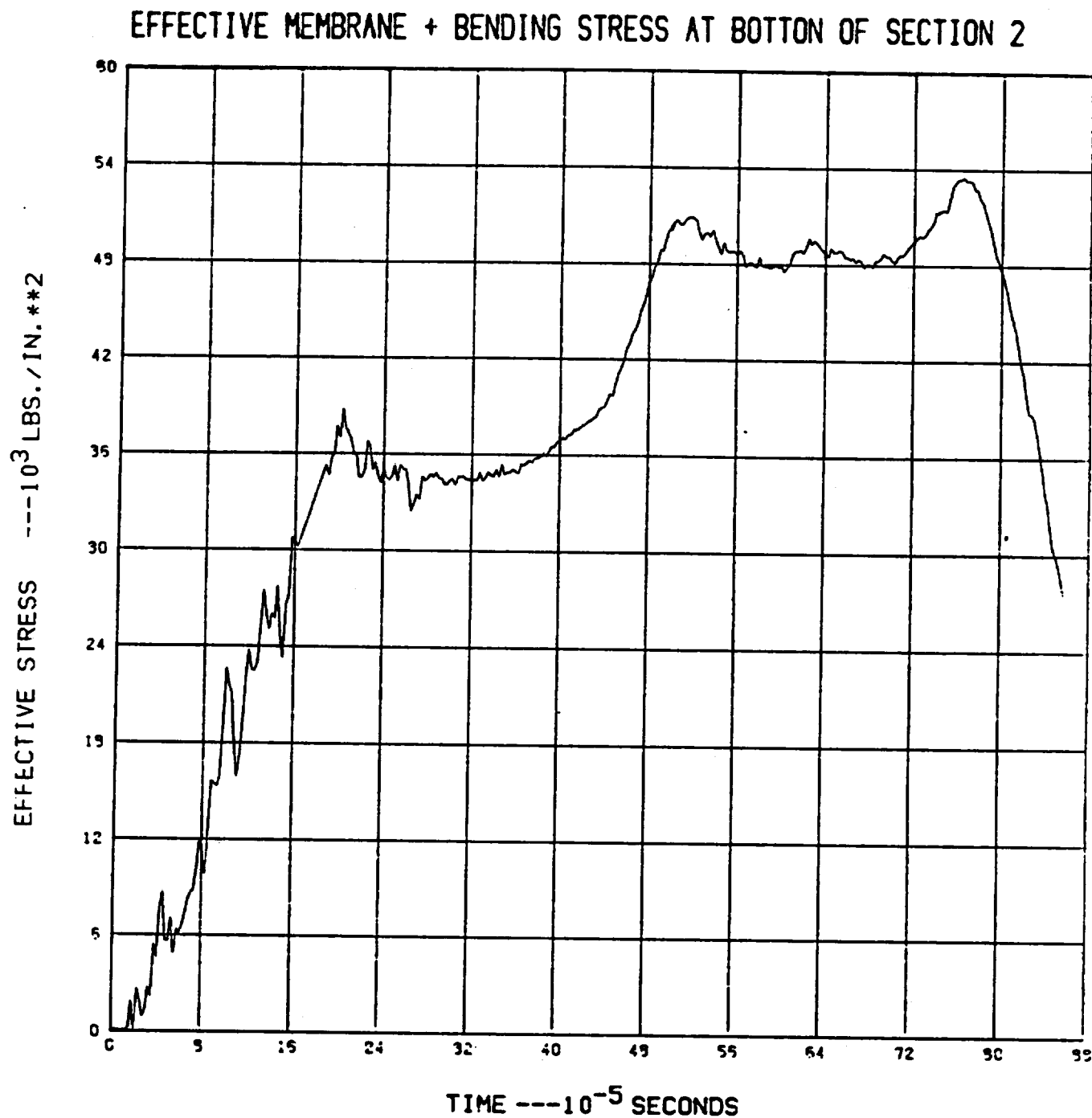
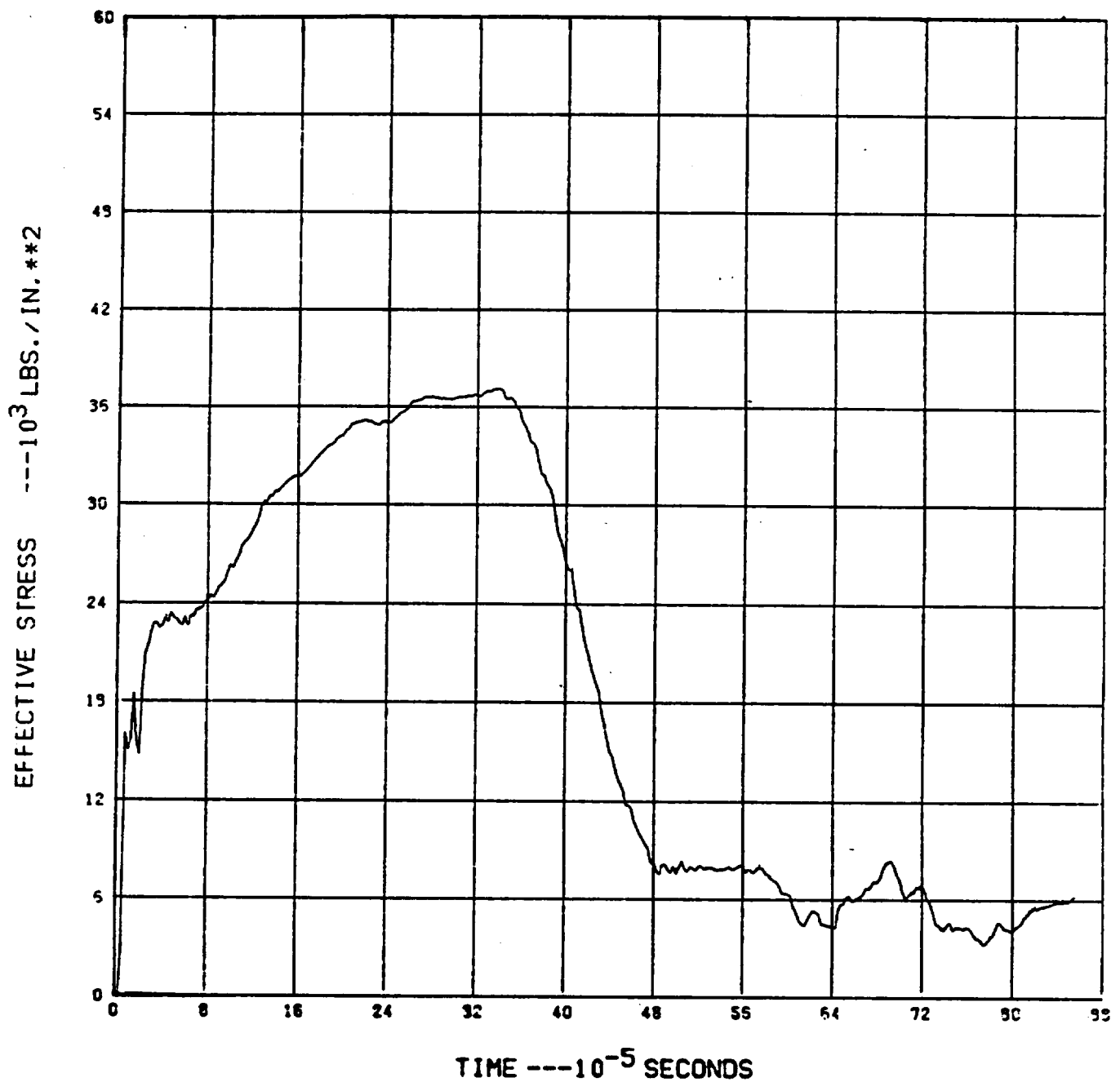


FIGURE 2-13 SECTION 2
PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT BOTTOM OF PLATE

EFFECTIVE PRIMARY MEMBRANE STRESS AT SECTION 3

FIGURE 2-14 SECTION 3
PRIMARY MEMBRANE STRESS

EFFECTIVE MEMBRANE + BENDING STRESS AT TOP OF SECTION 3

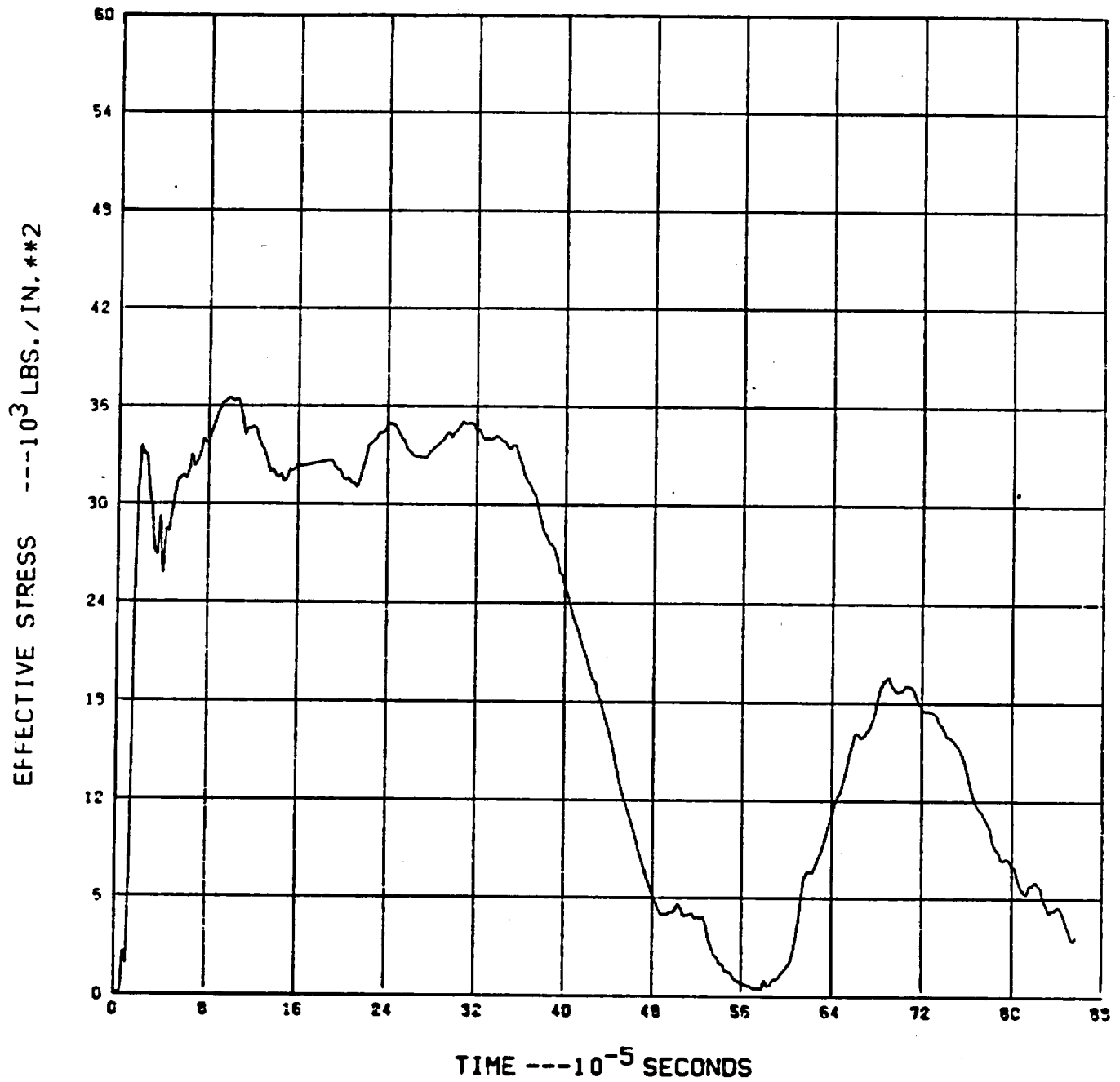


FIGURE 2-15 SECTION 3
PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT TOP OF PLATE

EFFECTIVE MEMBRANE + BENDING STRESS AT BOTTOM OF SECTION 3

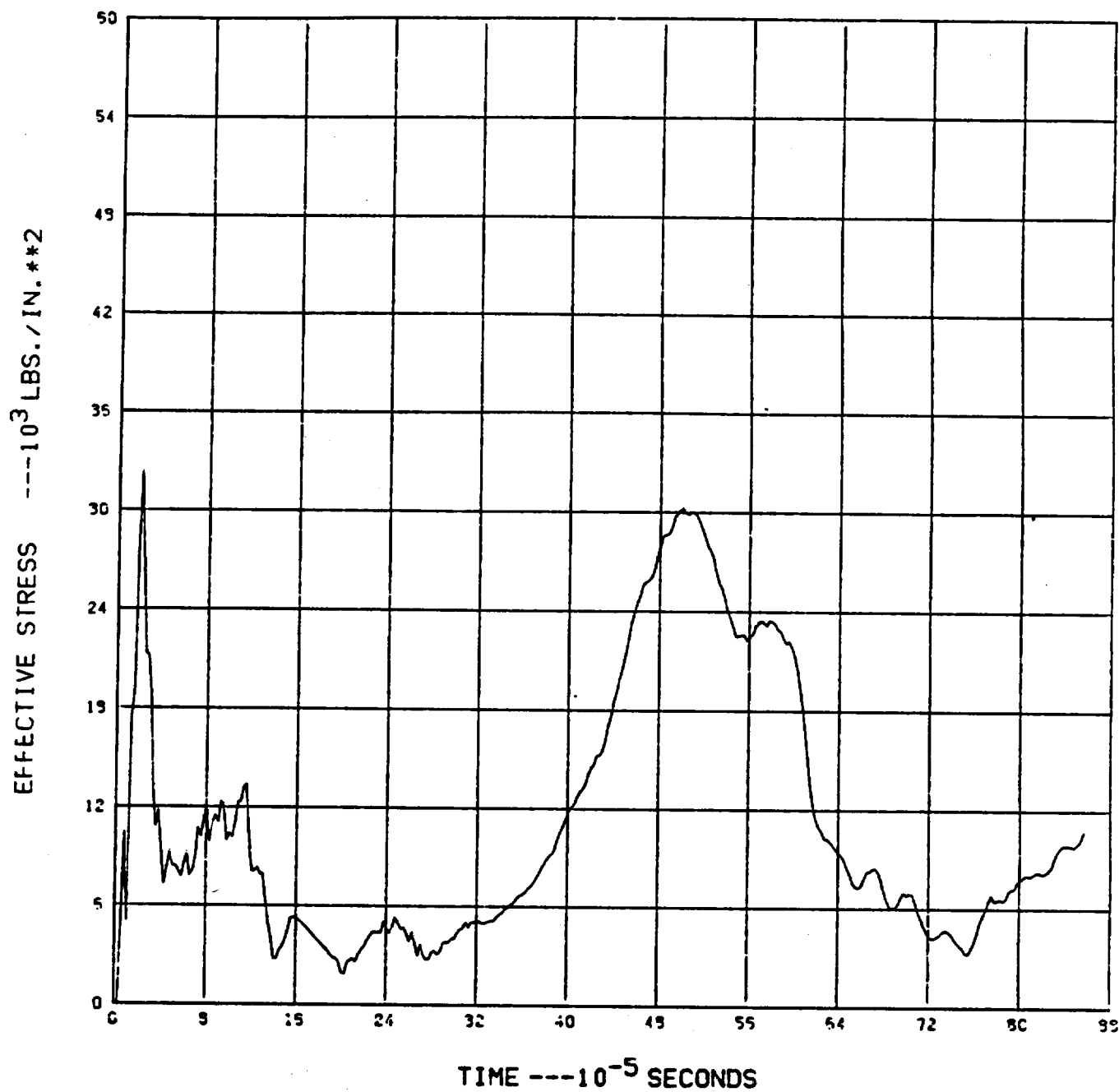


FIGURE 2-16 SECTION 3
PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT BOTTOM OF PLATE

EFFECTIVE PRIMARY MEMBRANE STRESS AT SECTION 4

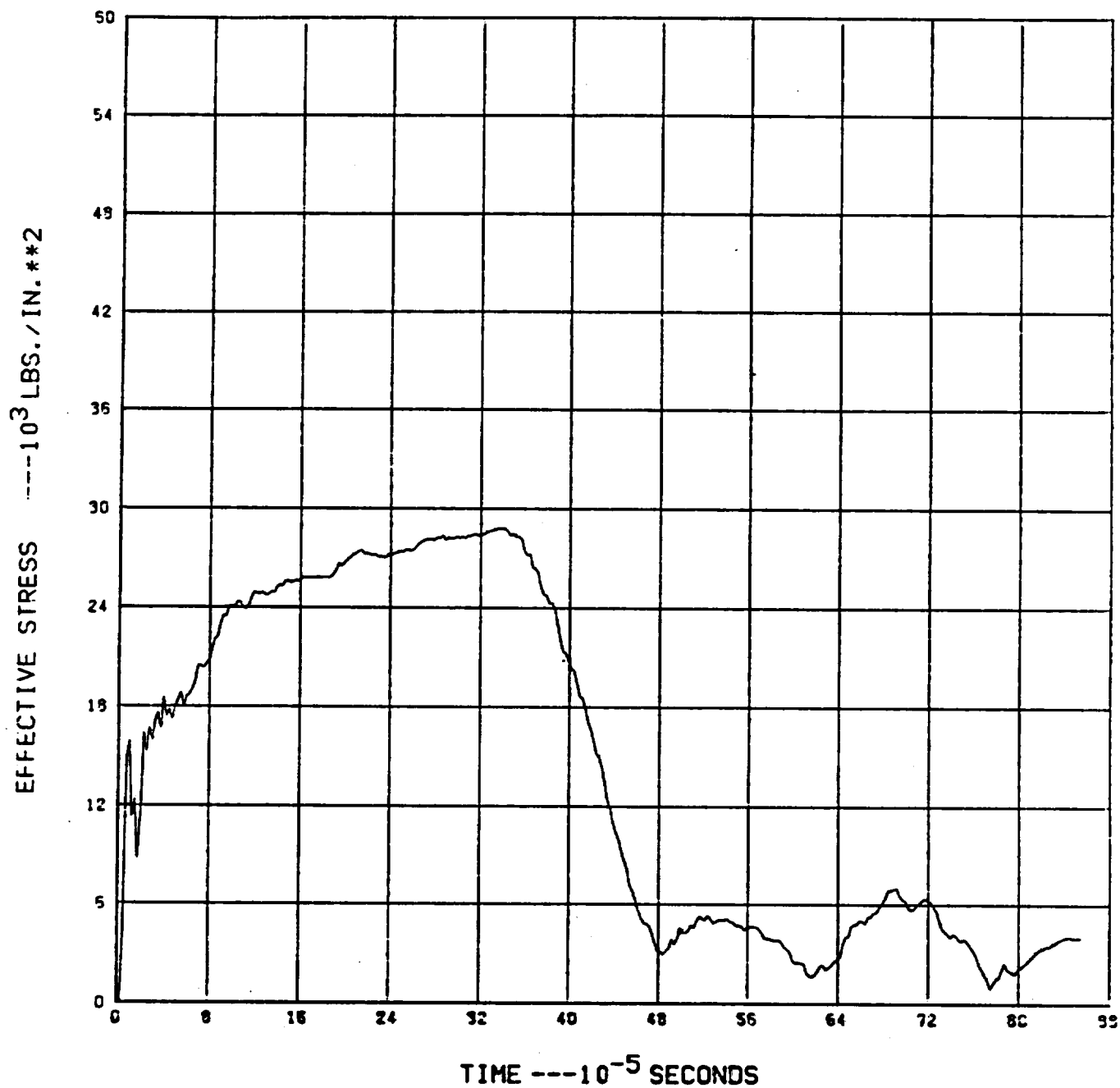


FIGURE 2-17 SECTION 4
PRIMARY MEMBRANE STRESS

EFFECTIVE MEMBRANE + BENDING STRESS AT TOP OF SECTION 4

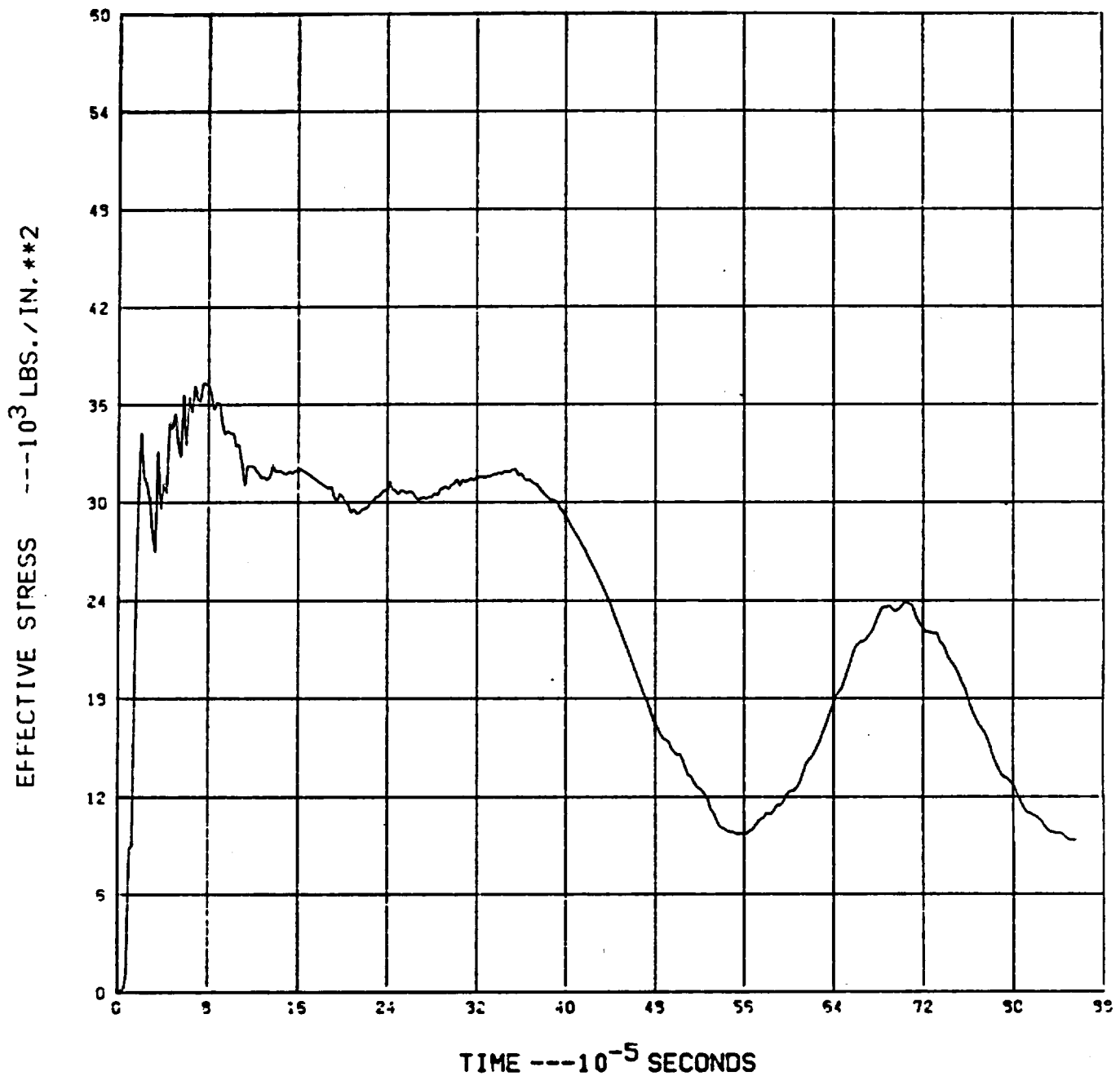


FIGURE 2-18 SECTION 4
PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT TOP OF PLATE

EFFECTIVE MEMBRANE + BENDING STRESS AT BOTTOM OF SECTION 4

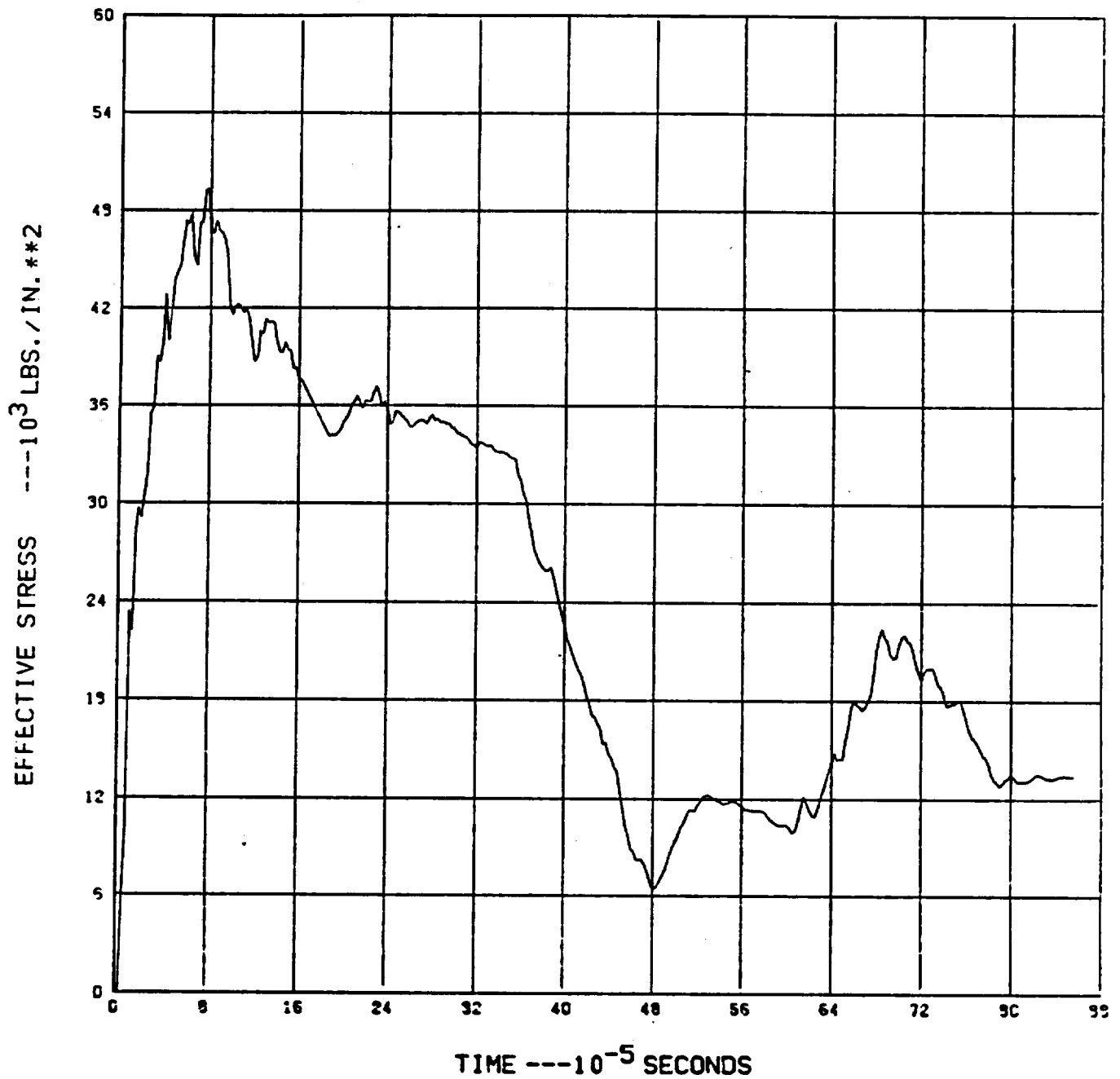
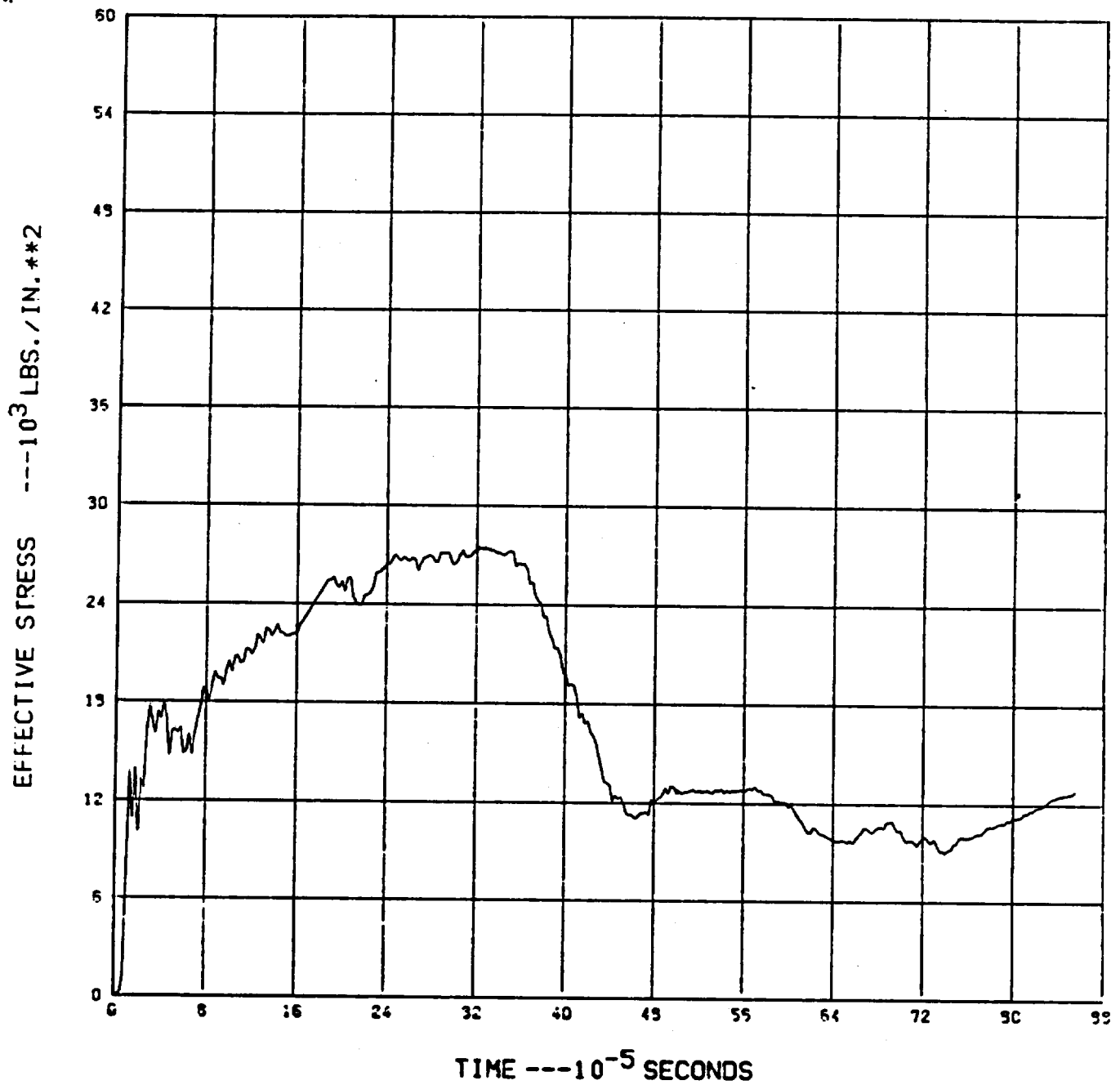


FIGURE 2-19 SECTION 4
PRIMARY MEMBRANE PLUS PRIMARY BENDING STRESS
AT BOTTOM OF PLATE

EFFECTIVE PRIMARY MEMBRANE STRESS AT SECTION 5

FIGURE 2-20 SECTION 5
PRIMARY MEMBRANE STRESS

EFFECTIVE MEMBRANE + BENDING STRESS AT TOP OF SECTION 5

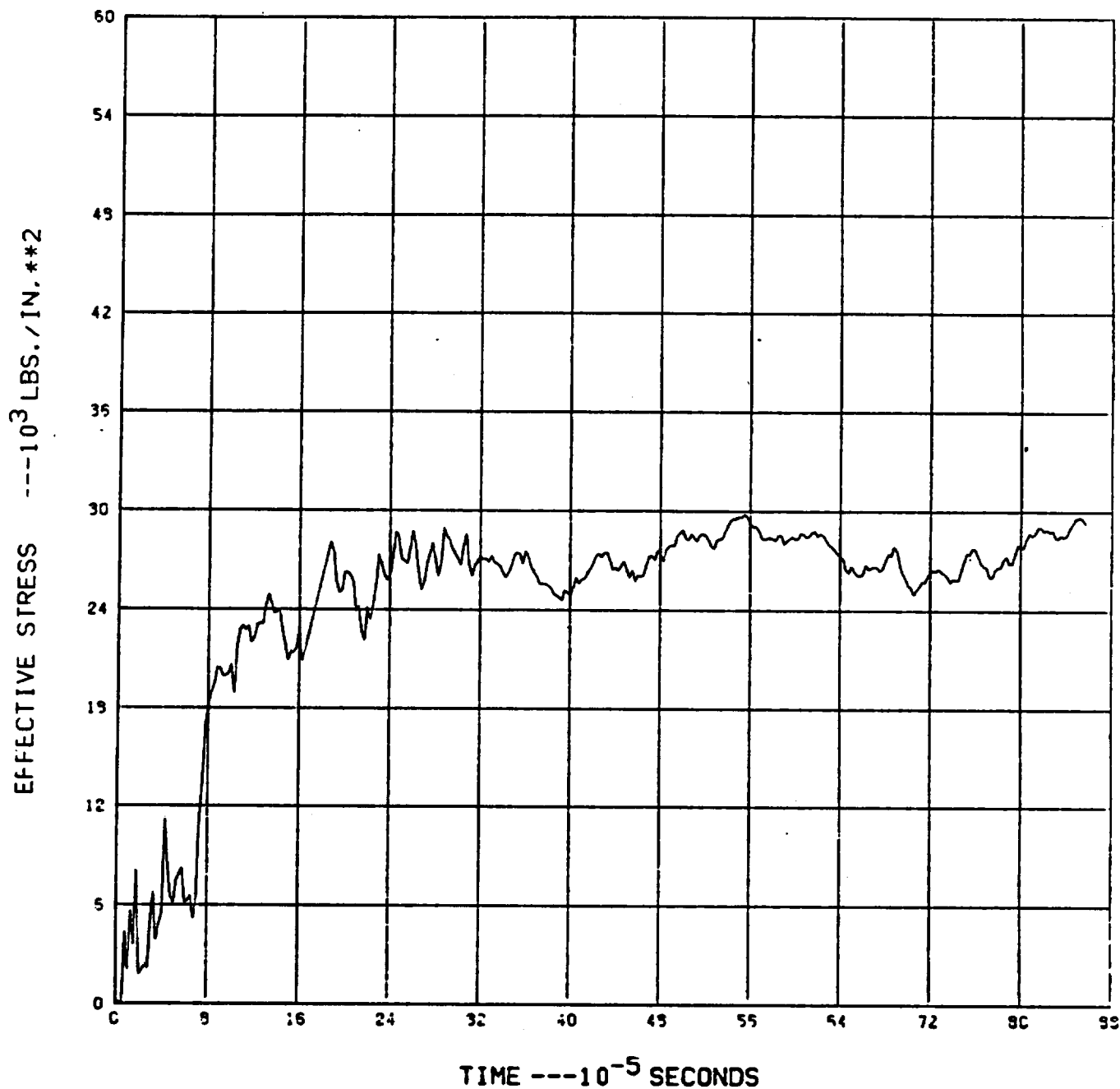


FIGURE 2-21 SECTION 5
PRIMARY MEMBRANE PLUS PRIMARY BENDING
AT INSIDE EDGE OF WELD

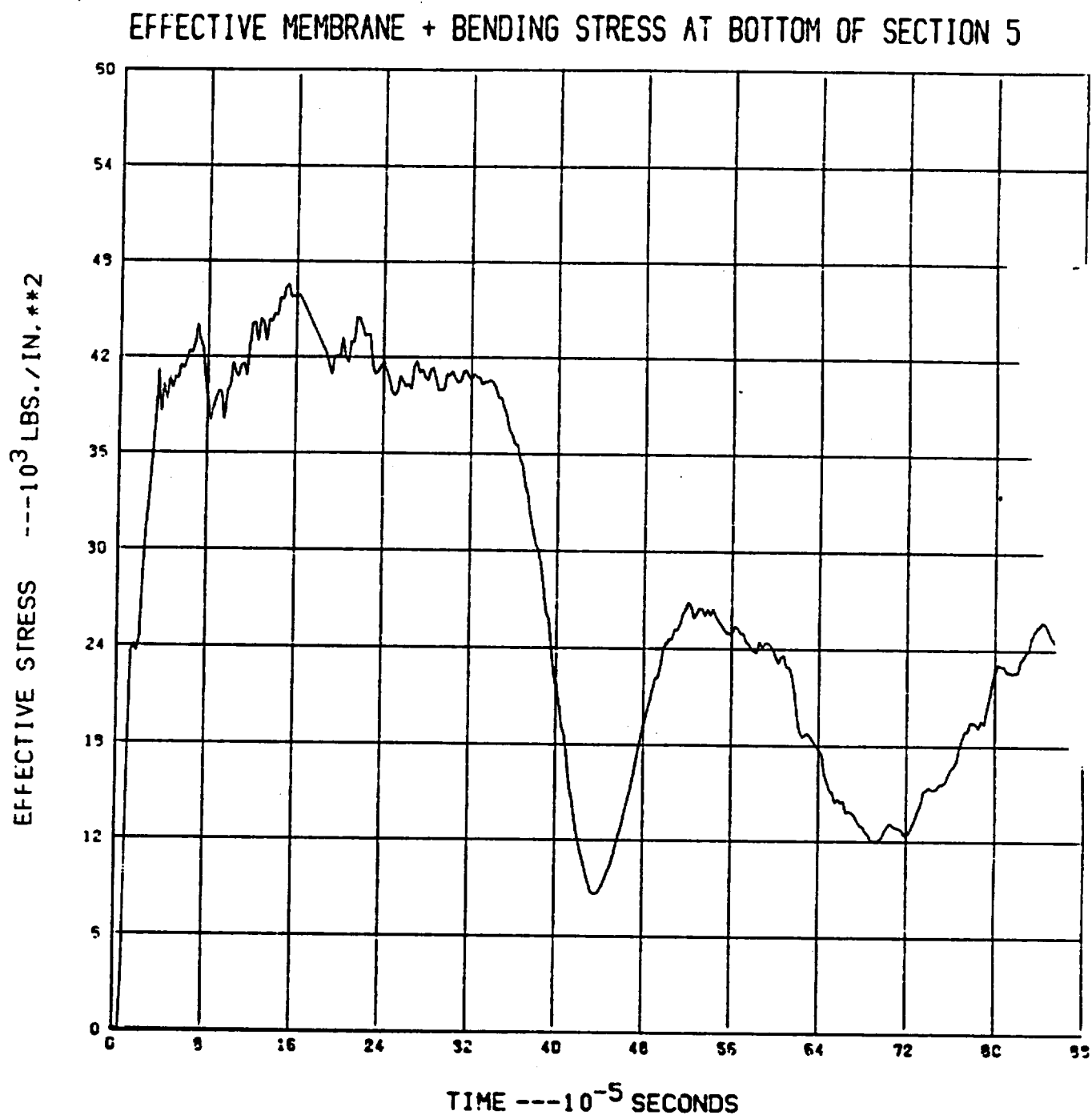


FIGURE 2-22 SECTION 5
PRIMARY MEMBRANE PLUS PRIMARY BENDING
AT OUTSIDE EDGE OF WELD

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

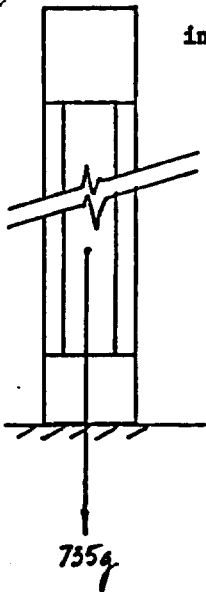
Shipping Cask Containment Vessels," will be used. Applying these criteria, the allowable stress intensities become:

Primary membrane	$2.4 S_m = 40 \text{ ksi.}$
Membrane plus bending	$3.6 S_m = 60 \text{ ksi.}$

for $S_m = 16.7 \text{ ksi.}$ 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

$G = 735 \text{ g's}$

A = cross sectional area of shielding sidewalls

For the 1-in.-thick shielding:

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

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

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

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

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:

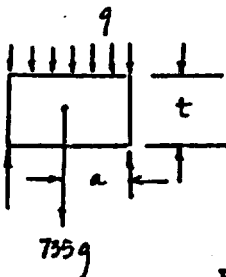
$$A = \frac{\pi}{4} [(6 \text{ in.})^2 - (2 \text{ in.})^2] = 25.1 \text{ in.}^2$$

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

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:

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

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

$$t_1 = 5 \text{ in.}, t_2 = 4 \text{ in.}$$

$$\nu = 0.3 = \text{Poisson's ratio}$$

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

$$a = 3 \text{ in.}$$

For the 5-in.-thick lid:

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

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

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

For the 4-in.-thick bottom plate:

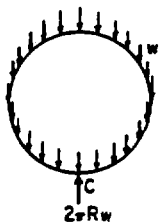
$$q = \frac{(735 \text{ g}) (32 \text{ lb})}{28.3 \text{ in.}^2} = 831 \frac{\text{lb}}{\text{in.}^2}$$

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

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

2.7.4.4 Side Drop. 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:



$$M_{\max} = \frac{3}{2} WR^2 \text{ at pt. C} \quad (1)$$

$$T = T_A U + V_A Z + L T_T \quad (2)$$

$$V = T_A Z + V_A U + L T_V \quad (3)$$

where W = weight per unit inch circumference

R = radius = 3 in.

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

$$V_A = 0$$

$$LT_A = -WRXZ$$

$$LT_V = -WRXU$$

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

$$U = \cos X = -1$$

$$Z = \sin x = 0$$

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

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

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

$$LT_T = 0$$

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

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:

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

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

$$LT_T = 0$$

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

Substituting into equations (1) through (3):

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

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

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

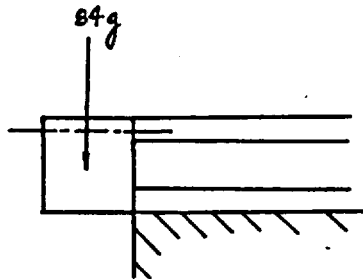
Then:

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

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

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.

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

where $G = 84 \text{ g's}$

$A_{\sigma} = \text{stress area of a } 3/8\text{-in. pin} = 0.1104 \text{ in.}^2$

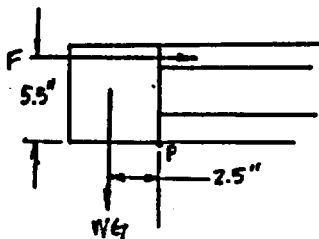
3 = number of pins

$W_{\text{lid}} = 40 \text{ lb}$

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

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

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



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

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

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

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:

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

where $G = 84 \text{ g's}$

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

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

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.