OAK RIDGE NATIONAL LABORATORY

operated by

UNION CARBIDE CORPORATION

NUCLEAR DIVISION

for the

U.S. ATOMIC ENERGY COMMISSION

ORNL - TM - 2293

STRUCTURAL ANALYSIS OF THE BROOKHAVEN NATIONAL LABORATORY PORTABLE CESIUM IRRADIATOR

J. H. Evans

NOTICE This document contains information of a preliminary nature and was prepared primarily for internal use at the Oak Ridge National Laboratory. It is subject to revision or correction and therefore does not represent a final report.

RESTRICTION OF THIS DOCUMENT IS OBSOLETE

LEGAL NOTICE

This report was prepared as an account of Government sponsored work. Neither the United States, nor the Commission, nor any person acting on behalf of the Commission:

- A. Makes any warranty or representation, expressed or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method, or process disclosed in this report may not infringe privately owned rights; or
- B. Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.

As used in the above, "person acting on behalf of the Commission" includes any employee or contractor of the Commission, or employee of such contractor, to the extent that such employee or contractor of the Commission, or employee of such contractor prepares, disseminates, or provides access to, any information pursuant to his employment or contract with the Commission, or his employment with such contractor.

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

Contract No. W-7405-eng-26

General Engineering Division

STRUCTURAL ANALYSIS OF THE
BROOKHAVEN NATIONAL LABORATORY PORTABLE CESIUM IRRADIATOR

J. H. Evans

SEPTEMBER 1969

LEGAL NOTICE

This report was prepared as an account of Government sponsored work. Neither the United States, nor the Commission, nor any person acting on behalf of the Commission:

A. Makes any warranty or representation, expressed or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method, or process disclosed in this report may not infringe privately owned rights; or

B. Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.

As used in the above, "person acting on behalf of the Commission" includes any employee or contractor of the Commission, or employee of such contractor, to the extent that such employee or contractor of the Commission, or employee of such contractor prepares, disseminates, or provides access to, any information pursuant to his employment or contract with the Commission, or his employment with such contractor.

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee
operated by
UNION CARBIDE CORPORATION
for the
U.S. ATOMIC ENERGY COMMISSION

This document is
PUBLICLY RELEASABLE

Bary Steels
Authorizing Official

Date: 8-28-67

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

Ry

PREFACE

The analysis contained in this report was prepared for the New York Operation's Office of the United States Atomic Energy Commission as a portion of the documentation required to show that after modification the irradiator analyzed complies with the regulations governing packages or containers in which radioactive material is shipped.

CONTENTS

Abstract	1
1. INTRODUCTION	1
2. SUMMARY	3
3. COMPLIANCE WITH GENERAL STANDARDS FOR ALL PACKAGING	5
Lifting Devices	5
Tie-Down Devices	6
Subbase Tie-Down to Trailer	8
Irradiator Tie-Down to Subbase	14
4. COMPLIANCE WITH STRUCTURAL STANDARDS FOR PACKAGING	18
Load Resistance	18
External Pressure	21
5. COMPLIANCE WITH STANDARDS FOR NORMAL CONDITIONS OF TRANSPORT ..	22
Heat	22
Cold	24
Pressure	24
Free Drop	25
Penetration	27
6. COMPLIANCE WITH STANDARDS FOR HYPOTHETICAL ACCIDENT	
CONDITIONS	29
Free Drop	32
Corner Impact	32
Side Impact on Intermediate Shutter Cave	34
Side Impact on Produce Input Shutter	38
Top Impact	40
Thermal	42
REFERENCES	47
Appendix A. MISCELLANEOUS CALCULATIONS	51
Appendix B. NOMENCLATURE	52

LIST OF FIGURES

<u>Figure Number</u>	<u>Title</u>	<u>Page Number</u>
1	Horizontal Section Through the Brookhaven National Laboratory Portable Cesium Irradiator	2
2	Tie-Down of Irradiator to Subbase and Subbase to Trailer	7
3	Equivalent Beam Cross Section, Stress Distribution, and Deformed Section	8
4	Proposed Bolting Pattern for the Irradiator Tie-Down to the Subbase	15
5	Load Diagram, Simplified Cross Section, and Equivalent Cross Section of Irradiator Considered as a Simple Beam for Load Resistance Analysis	19
6	Unclad Lead Model and Stress-Strain Curve for Lead Assumed to Investigate Effects of Free Drop Condition of Normal Transport	26
7	Plan of Proposed Crash Frame for Irradiator	30
8	Section A - A Through Proposed Crash Frame	31
9	Section B - B Through Proposed Crash Frame	31
10	Collapse Mechanism for Drop on Compound Corner of the Package	33
11	Cross Section of Model Assumed to Investigate the Results of Side Impact on Surface Normal to Intermediate Shutter Cave	35
12	Stress-Strain Curves for Steel and Lead Assumed to Investigate the Results of Side Impact of Irradiator on Surface Normal to the Intermediate Shutter Cave	36
13	Recommended Strengthening of Welds at the Base and Flange of the Intermediate Shutter Cave	37
14	Produce Input Shutter Closure	39
15	Computational Model of Irradiator Used to Determine the Effects of the 30-Minute Thermal Accident Condition	42

1

2

STRUCTURAL ANALYSIS OF THE BROOKHAVEN NATIONAL LABORATORY PORTABLE CESIUM IRRADIATOR

Abstract

An analytical evaluation of the Brookhaven National Laboratory Portable Cesium Irradiator was made to demonstrate its compliance with regulations governing packages in which radioactive material is shipped. Computational procedures and testing and operating experience were used to determine the structural integrity and thermal behavior of the irradiator with respect to the general standards for packaging, the structural standards, standards for normal conditions of transport, and the standards for the hypothetical accident conditions. This analysis demonstrated that with the proposed crash frame, puncture plates, and the other minor modifications, the irradiator complies with all the applicable regulations.

1. INTRODUCTION

When a package containing radioactive or fissile material is shipped from one location to another, the package is subject to regulations that govern its structural integrity and shielding capability. The regulations in Chapter 0529 of the United States Atomic Energy Commission Manual¹ and in Title 10, Part 71, of the Code of Federal Regulations² govern the package when it is shipped within the continental United States. To secure approval for shipment, it must be shown by test, experimental, or computational methods that the package complies with these regulations. The Brookhaven National Laboratory Portable Cesium Irradiator was analyzed to determine whether or not it complies with the regulations. The methods used and the results of the evaluation are reported here.

The Brookhaven National Laboratory Portable Cesium Irradiator shown schematically in Fig. 1 is a self-contained portable facility used for demonstrational irradiation of fresh produce to retard spoilage. The irradiator is clad with 1/2-in.-thick steel, and the biological shielding for the gamma source is provided by the lead body, shutters, and door. The

ORNL DWG 69-9537

Fig. 1. Horizontal Section Through the Brookhaven National Laboratory Portable Cesium Irradiator.

irradiator is divided into two compartments by the intermediate sliding shutter. One compartment houses the ^{137}Cs gamma source, and the other compartment is used for the introduction of fresh fruit and vegetables while the intermediate shutter is closed. A shutter-slide arrangement is used to bring the fresh produce into close proximity to the gamma source while maintaining adequate shielding for personnel.

There is only one unit of this design in existence, and it is the property of the New York Operation's Office of the United States Atomic Energy Commission. This approximately 40,000-lb irradiator is mounted semipermanently in a highway trailer. When the unit is moved from place to place over public highways, it is subject to the regulations governing packages containing radioactive material in transit. Those portions of the regulations applicable to this irradiator and the methods used to show compliance with them are discussed in this report, and design modifications to eliminate deficiencies are recommended where necessary.

2. SUMMARY

The Brookhaven National Laboratory Portable Cesium Irradiator was examined to demonstrate its compliance with the applicable general standards for all packaging, the structural standards for the package, the standards for normal conditions of transport, and the standards for the hypothetical accident conditions. This irradiator has a ^{137}Cs gamma source, and the cladding of the capsule containing the source is designed so that it will not be vented to the atmosphere. There is no circulating coolant provided for in the irradiator, and no fissile material will be shipped in it. Therefore, those portions of the regulations related to these features of packages in which radioactive material is shipped are not applicable to this irradiator.

On the basis of experience and computational procedures, we determined that the existing irradiator is deficient in some areas. There also was insufficient information available in certain areas to demonstrate compliance with the regulations. The minimum condition for compliance in these areas was therefore computed so that inspection of the irradiator will demonstrate compliance. Minor modification of bolting patterns in some areas is proposed along with the use of high-strength bolting material. With respect to the general standards for all packaging, it is noted that the irradiator has no system of lifting devices for the entire unit. Based on our understanding of the routine operating procedures of the irradiator, we feel that no such system of lifting devices is required and have therefore proposed none. With respect to the hypothetical accident conditions, the design for the corner weld joints is such that the irradiator will not withstand the 30-ft-free-drop accident condition without rupture, nor is the thickness of the side-wall plate of the irradiator sufficient to withstand the 40-in. free drop onto a 6-in.-diameter cylinder without puncture.

Since it is not practical to repair a deficient weld joint, a design for a crash frame or shock absorbing structure around the irradiator is proposed to "soften" the impact from the 30-ft free drop. It is also proposed that additional plate 1/2 in. thick be placed around the

irradiator parallel to all the existing flat surfaces to increase puncture resistance. This approach was adopted for evaluating the effects of the hypothetical accident conditions, and the package considered was comprised of the irradiator in the proposed crash frame with the proposed puncture plates added. With all of the aforementioned modifications, the irradiator analyzed in this study complies with all of the applicable regulations governing packages in which radioactive material is shipped.

3. COMPLIANCE WITH GENERAL STANDARDS FOR ALL PACKAGING

The general standards for all packaging specify that there shall be no chemical, galvanic, or other reaction among the materials of the package or between the materials and the contents of the package. Further, the standards stipulate that the packaging shall be equipped with a positive closure that will prevent inadvertent opening, and they also cover package lifting and tie-down devices.

The Brookhaven National Laboratory Portable Cesium Irradiator is constructed of carbon steel and lead, and we anticipate no significant chemical, galvanic, or other type of reaction between these two materials of construction or between the materials and the intended contents of the irradiator. The irradiator has positive bolted closures and padlocks to prevent inadvertent opening. The requirements for lifting and tie-down devices and the means of complying with them are discussed in the following sections of this chapter.

Lifting Devices

If there is a system of lifting devices for the cask that is a structural part of the package, the regulations require that this system be capable of supporting three times the weight of the loaded package without generating stress in any material of the package in excess of the yield strength of that material. Further, if there is a system of lifting devices that is a structural part of the lid only, the system shall be capable of supporting three times the weight of the lid and attachments without generating stress in any material of the lid in excess of the yield strength of that material.

The irradiator analyzed in this study has no system of lifting devices for the entire unit, and since we feel that such a system is not required, we have proposed none. Nor does the irradiator have a lid as such, but the source access door should conform to the requirements for lid-lifting devices. The lifting device on the source access door is an

eye bolt. The size of this eye bolt is not known, and it is specified only as a 5-ton eye bolt. It was therefore assumed that the eye bolt is made of SAE 1035 steel with a stress of 55,000 psi at the yield point of the material (page 1763 in Ref. 3). To compute the minimum size of this type of eye bolt that will comply with the regulations, the force or concentrated load on the eye bolt

$$F_B = 3W_{sd} = 7410 \text{ lb} , \quad (3.1)$$

where W_{sd} = the weight of the source access door = 2470 lb (as calculated in Appendix A). The area of the shank of the eye bolt

$$A = \frac{\pi D^2}{4} = \frac{F_B}{S_t} , \quad (3.2)$$

where

D = the diameter of the eye bolt shank, in.,

S_t = tensile stress in the eye bolt shank, psi.

The diameter of the eye bolt shank

$$D = \left(\frac{4F_B}{\pi S_{yp}} \right)^{1/2} = 0.414 \text{ in.} \quad (3.3)$$

A physical inspection would verify that the lifting device on the source access door complies with the regulations.

Tie-Down Devices

If there is a system of tie-down devices that is a structural part of the package, the regulations require that this system be capable of withstanding a static force applied to the center of gravity of the package with a vertical component of two times the weight of the package and its contents, a horizontal component along the direction of travel of ten times the weight of the package and its contents, and a horizontal component in the transverse direction of five times the weight of the package and its contents. This static force must be withstood without generating stress in any material of the package in excess of the yield strength of that material.

The irradiator analyzed in this study is a self-contained unit that is bolted to a subbase which is in turn bolted to the bed of a trailer in a semipermanent mounting, as illustrated in Fig. 2. Therefore, the stresses resulting from the loading requirements for tie-down devices must be evaluated at two locations. These are (1) the point of attachment of the subbase to the trailer and (2) the point of attachment of the irradiator to the subbase.

ORNL DWG 69-9538

KEY PLAN

SECTION A-A

Fig. 2. Tie-Down of Irradiator to Subbase and Subbase to Trailer.

Subbase Tie-Down to Trailer

The stresses developed at the subbase-to-trailer attachment plane (the Z_0 plane in Fig. 2) were computed by considering the irradiator and subbase as a cantilevered beam loaded as prescribed by the regulations governing tie-down. The equivalent beam cross section, stress distribution, and deformed section are illustrated in Fig. 3. The section resisting bending in the x direction at plane Z_0 is a strip of the oak floor of

ORNL DWG 69-9539

Fig. 3. Equivalent Beam Cross Section, Stress Distribution, and Deformed Section.

the trailer loaded in compression, shown as "a" in Fig. 3(a), and the 7/8-in.-diameter tie-down bolts loaded in tension. Similarly, the section resisting bending in the y direction is a strip of the oak floor of the trailer loaded in compression, shown as "b" in Fig. 3(a), and the 7/8-in.-diameter bolts loaded in tension. It was assumed that the subbase is rigid or that plane sections remain and that all stresses are less than the proportional limit. The forces normal to the Z_o plane were summed to obtain the following equation.

$$\Sigma F_B = \int S'_w dA, \quad (3.4)$$

where

F_B = the force in any bolt, lb,

S'_w = the compressive stress in the wood variable from 0 at the neutral axis to a maximum of S_w at $a' = a$, psi,

dA = the differential area = $L da'$

where L = the length of the subbase frame, in.

Letting

F = the force in the most distant bolt from the neutral axis, lb,

d = the distance from the neutral axis to the most distant bolt, in.,

d' = the distance from the neutral axis to any bolt, in.,

A_B = the cross-sectional area of one bolt, in.²,

S_t = the tensile stress in the most distant bolt from the neutral axis, psi, and

n = the number of bolts at any particular d' ;

by similar triangles

$$\frac{F}{d} = \frac{F_B}{d'}$$

and

$$F_B = \frac{Fd'}{d}. \quad (3.5)$$

Then

$$\Sigma F_B = \frac{F}{d}(\Sigma d'n) = \frac{S_t A_B}{d}(\Sigma d'n). \quad (3.6)$$

Also by similar triangles,

$$\frac{S'_w}{a'} = \frac{S_w}{a}$$

and
$$S'_w = \frac{S_w}{a}(a') . \quad (3.7)$$

Then
$$S'_w da = \frac{S_w L}{a}(a' da) = \frac{S_w La}{2} . \quad (3.8)$$

Letting

ϵ'_w = strain in the wood at any point, in./in.,

ϵ_w = maximum strain in the wood, in./in.,

ϵ'_s = strain in the steel at any point, in./in.,

ϵ_s = maximum strain in the steel, in./in.,

E_w = modulus of elasticity of wood, lb/in.², and

E_s = modulus of elasticity of steel, lb/in.² ;

it may be seen from Fig. 3(c) that by similar triangles

$$\frac{\epsilon dx}{y} = \frac{dx}{r}$$

and
$$\epsilon = \frac{y}{r} . \quad (3.9)$$

Also, with reference to Fig. 3(b),

$$\frac{\epsilon'_w}{\epsilon'_s} = \frac{a'}{d'}$$

and
$$\frac{\epsilon_w}{\epsilon_s} = \frac{a}{d} = \frac{\frac{S_w}{E_w}}{\frac{S_t}{E_s}} .$$

Therefore,

$$S_w = \frac{a}{d} \left(\frac{S_t}{E_s} \right) E_w . \quad (3.10)$$

Substituting in Eq. 3.8 for S_w and equating Eq. 3.6 to Eq. 3.8, Eq. 3.11 results since the summation of forces on the plane must be equal to zero.

$$\frac{S_t A_B}{d} (\Sigma d'n) = \frac{a}{d} (S_t) \left(\frac{E_w}{E_s} \right) \left(\frac{La}{2} \right) ,$$

$$A_B \Sigma d'n = \frac{E_w La^2}{E_s} . \quad (3.11)$$

Making the substitution

$$\Sigma d'n = \Sigma d''n - Na, \quad (3.12)$$

where

d'' = the distance from the front of the frame to the bolt in question, in.,

N = the number of d'' 's summed,

and solving for a ,

$$a = \frac{2E_s A_B (\Sigma d''n - Na)}{E_w L} \quad (3.13)$$

In the x direction,

$$E_s = 30 \times 10^6 \text{ lb/in.}^2,$$

$$A_B = 0.462 \text{ in. (page 1110 in Ref. 3),}$$

$$E_w = 1.8 \times 10^6 \text{ lb/in.}^2 \text{ (page 73 in Ref. 4),}$$

$$n = 2,$$

$$N = 20,$$

$$L = 90 \text{ in., and}$$

$$d'' = 1508 \text{ in.}$$

Squaring both sides of Eq. 3.13, rearranging, and substituting in the above values,

$$a^2 + 3.4a - 515 = 0.$$

Solving by the quadratic formula,

$$a = 21.2 \text{ in.}$$

Similarly, in the y direction,

$$b = \frac{2E_s A_B (\Sigma d''n - Nb)}{E_w L} \quad (3.14)$$

and

$$n = 10,$$

$$L = 278 \text{ in.,}$$

$$d'' = 83 \text{ in.,}$$

$$N = 10.$$

Squaring both sides of Eq. 3.14, rearranging, and substituting in the values for the y direction,

$$b^2 + 0.554b - 4b = 0.$$

Solving by the quadratic formula,

$$b = 6.78 \text{ in.}$$

Since the summation of moments is equal to zero,

$$\begin{aligned} M_x &= 10WZ_1 \\ &= \sum nF_B d' + \int S'_w L a' da' , \end{aligned} \quad (3.15)$$

where

W = the weight of the irradiator = 40,000 lb,

Z_1 = the distance from the Z_0 plane to the center of gravity of the irradiator = 34.6 in.

It has been previously shown that

$$F_B = \frac{Fd'}{d} = \frac{S_t A_B d'}{d}$$

and

$$S'_w = \frac{a}{d} \left(\frac{S_t}{E_s} \right) E_w .$$

Therefore,

$$S'_w = \frac{S_t a' E_w}{d E_s} . \quad (3.16)$$

Substituting in Eq. 3.15,

$$10WZ = \frac{S_{tx} A_B n}{d_x} (\sum d'_x)^2 + \frac{S_{tx} L_x E_w}{d_x E_s} \int_0^{21.2} (a')^2 da' , \quad (3.17)$$

where

$$A_B = 0.462 \text{ in.},$$

$$n = 2,$$

$$d_x = 246.8 \text{ in.},$$

$$L_x = 85 \text{ in.}$$

Integrating and solving Eq. 3.17, the tensile stress in the most distant bolt from the neutral axis in the x direction

$$S_{tx} = 15,000 \text{ psi} .$$

Similarly,

$$5WZ = \frac{S_{ty} A_B n}{d_y} (\sum d'_y)^2 + \frac{S_{ty} L_y E_w}{d_y E_s} \int_0^{6.78} (b')^2 db' , \quad (3.18)$$

where

$$A_B = 0.462 \text{ in.},$$

$$n = 10,$$

$$d_y = 76 \text{ in.},$$

$$L_y = 278 \text{ in.}$$

Integrating and solving Eq. 3.18, the tensile stress in the most distant bolt from the neutral axis in the y direction

$$S_{ty} = 18,502 \text{ psi} .$$

The direct tensile stress equally applied to all bolts

$$S'_t = \frac{2W}{NA_B} , \quad (3.19)$$

where N = the total number of bolts securing the irradiator subbase to the trailer bed = 20.

$$S'_t = \frac{2(40,000)}{20(0.462)} = 8658 \text{ psi} .$$

The maximum tensile stress, S_t , occurs in the bolt marked with the symbol $\oplus$ in Fig. 3(a), and the value of the maximum tensile stress

$$\begin{aligned} S_t &= S_{tx} + S_{ty} + S'_t \\ &= 42,160 \text{ psi} . \end{aligned} \quad (3.20)$$

Computing the shearing stress on the bolts, the total shearing force is the vector sum of the 10W and the 5W xy forces.

$$\begin{aligned} F_{\max} &= [(10W)^2 + (5W)^2]^{1/2} \\ &= 11.18W = 447,200 \text{ lb} . \end{aligned} \quad (3.21)$$

The shearing stress

$$S_s = \frac{F_{\max}}{A_s} = \frac{F_{\max}}{NA_B} , \quad (3.22)$$

where

$$A_s = \text{the total shear area, in.}^2,$$

$$N = \text{the total number of bolts} = 20,$$

$$A_B = \text{the cross-sectional area of one bolt} = 0.462 \text{ in.}^2$$

$$S_s = \frac{447,200}{20(0.462)} = 48,370 \text{ psi} .$$

From Mohr's Circle of Stress, the maximum stress

$$S_{\max} = \frac{S_t}{2} + \left[\left(\frac{S_t}{2} \right)^2 + S_s^2 \right]^{1/2} \quad (3.23)$$

$$= 73,800 \text{ psi .}$$

The drawings provided for the analysis of this irradiator do not specify the bolting material, but it is implied that all material must conform to ASTM A-36 or A-373. This would mean that the yield strength of the material is 32,000 to 36,000 psi, and this is inadequate for the bolting material. High-strength bolts should be used to secure the sub-base to the trailer bed, and these bolts should have sufficient strength above the required 73,800 psi to permit pre-load tightening.

Irradiator Tie-Down to Subbase

The stresses developed at the irradiator-to-subbase attachment plane (the Z plane in Fig. 2) must be investigated. Preliminary calculations demonstrated that the existing attachment is inadequate. We therefore propose that the existing 3/4-in.-diameter bolts be replaced with high-strength 1 1/4-in.-diameter NF bolts and that two additional bolts be installed to effect the bolt pattern shown in Fig. 4.

The neutral axes in the x and y directions are the centroids of the compression area and the bolts may be located from the equation

$$\frac{a^2 h}{2} = A_B \Sigma (\bar{b} - a) = A_B \Sigma \bar{b} - A_B \Sigma a , \quad (3.24)$$

where A_B = the area of one bolt, in.², and the other terms are as shown in Fig. 4. Rewriting Eq. 3.24,

$$\frac{a^2 h}{2} + N A_B a - A_B \Sigma \bar{b} = 0 . \quad (3.25)$$

In the x direction,

$$h_x = 76 \text{ in.},$$

$$A_B = 1.073 \text{ in.}^2,$$

$$\bar{b}_x = 120 + 390 = 510 \text{ in.},$$

$$N_x = \text{the number of bolts in tension assumed to be 8.}$$

Fig. 4. Proposed Bolting Pattern for the Irradiator Tie-Down to the Subbase.

Solving Eq. 3.25 by the quadratic formula

$$a_x = 3.7 \text{ in.}$$

Since the compression area contains the five left-hand bolts, the solution with $N = 8$ is the proper one.

Similarly, in the y direction,

$$h_y = 79.75 \text{ in.},$$

$$A_B = 1.073 \text{ in.}^2,$$

$$\Sigma \bar{b}_y = 471.74 \text{ in.},$$

$$N_y = 12.$$

Solving Eq. 3.25 by the quadratic formula,

$$a_y = 3.37 \text{ in.}$$

The moment of inertia in the x direction

$$\begin{aligned} I_x &= \frac{1}{3} h_x a_x^3 + A_B \Sigma (\bar{b}_x - a_x)^2 \\ &= 38,100 \text{ in.}^4 \end{aligned} \quad (3.26)$$

The moment of inertia in the y direction

$$\begin{aligned} I_y &= \frac{1}{3} h_y a_y^3 + A_B \Sigma (\bar{b}_y - a_y)^2 \\ &= 26,600 \text{ in.}^4 \end{aligned} \quad (3.27)$$

The maximum normal stress in the most stressed bolts

$$S_n = \frac{2W}{NA_B} + \frac{M_y b_{y \text{ max}}}{I_y} + \frac{M_x b_{x \text{ max}}}{I_x} . \quad (3.28)$$

The moment in the y direction

$$M_y = 5WZ_2 , \quad (3.29)$$

where

W = the weight of the irradiator = 40,000 lb ,

Z_2 = the distance from the Z plane (Fig. 2) to the center of gravity of the irradiator = 24.5 in.

$$M_y = 4.9 \times 10^6 \text{ in.-lb .}$$

The moment in the x direction

$$M_x = 10WZ_2 = 9.8 \times 10^6 \text{ in.-lb .}$$

Therefore, the maximum normal stress in the most stressed bolts

$$S_n = 37,600 \text{ psi .}$$

The shearing stress

$$\begin{aligned} S_s &= \frac{P}{A_B} = \frac{W[(10)^2 + (5)^2]^{1/2}}{\frac{\pi D^2}{4}} \\ &= 28,000 \text{ psi .} \end{aligned} \quad (3.30)$$

From Mohr's Circle of Stress, the maximum stress

$$S_{\text{max}} = \frac{S_n}{2} + \left[\left(\frac{S_n}{2} \right)^2 + S_s^2 \right]^{1/2} \quad (3.31)$$

$$S_{\max} = 52,000 \text{ psi} .$$

The proposed bolting modification for the irradiator-to-subbase tie-down therefore exceeds the requirements of the regulations.

4. COMPLIANCE WITH STRUCTURAL STANDARDS FOR PACKAGING

The structural standards for large-quantity packaging cover load resistance of the packaging and the external pressure which the package must withstand.

Load Resistance

When regarded as a simple beam supported at its ends along any major axis, the package must be capable of withstanding a static load normal to and uniformly distributed along its length that is equal to five times its fully loaded weight without generating stress in any material of the package in excess of the yield strength of that material.

The load diagram for the irradiator considered as a simple beam is shown in Fig. 5(a), and a simplified cross section of the irradiator is shown in Fig. 5(b). By reinforced beam theory,

$$R = \frac{\text{modulus of elasticity of steel}}{\text{modulus of elasticity of lead}} \quad (4.1)$$

$$= \frac{E_s}{E_\ell} = 15,$$

where

$$E_s = 30 \times 10^6 \text{ psi},$$

$$E_\ell = 2 \times 10^6 \text{ psi}.$$

Also,

$$\frac{\text{area of lead}}{\text{equivalent area of steel}} = \frac{A_\ell}{A_s} = R = 15. \quad (4.2)$$

Therefore, by assuming the simplified irradiator geometry shown in Fig. 5(b), the lead-steel cross section of this irradiator can be converted to the equivalent steel cross section shown in Fig. 5(c) by using the following relationships.

$$y_{\text{lead}} = y_{\text{steel equivalent}} \quad (4.3)$$

and

Fig. 5. Load Diagram, Simplified Cross Section, and Equivalent Cross Section of Irradiator Considered as a Simple Beam for Load Resistance Analysis.

$$\frac{x_{\text{lead}}}{R} = x_{\text{steel equivalent}} \quad (4.4)$$

with direction

$$x_{1e} = \frac{x_1}{R} = 1.67 \text{ in.}$$

and

$$x_{2e} = \frac{x_2}{R} = 0.517 \text{ in.}$$

The moment of inertia of the equivalent cross section about X_0 shown in Fig. 5(c)

$$\begin{aligned} I_{x0} = & \frac{1}{12}(2)b_1(h_1)^3 + 2b_2h_2\left(\frac{h_1 - b_2}{2}\right)^2 + \frac{1}{12}(2)b_3(h_3)^3 \\ & + 2b_3h_3\left(\frac{h_4 - b_3}{2}\right)^2 + \frac{1}{12}(2)b_4(h_4)^3 \\ & + \frac{1}{12}(2)b_5(h_5)^3 + 2b_5h_5\left(\frac{h_4}{2} + \frac{h_5}{2}\right)^2 \end{aligned} \quad (4.5)$$

Where

$$b_1 = 1.017 \text{ in.},$$

$$h_1 = 37 \text{ in.},$$

$$b_2 = 40.5 \text{ in.},$$

$$h_2 = 0.5 \text{ in.},$$

$$b_3 = 24 \text{ in.},$$

$$h_3 = 0.5 \text{ in.},$$

$$b_4 = 0.5 \text{ in.},$$

$$h_4 = 23 \text{ in.},$$

$$b_5 = 1.67 \text{ in.},$$

$$h_5 = 7.5 \text{ in.},$$

$$I_{x0} = 32,067 \text{ in.}^4$$

The maximum moment of force

$$\begin{aligned} M_{\max} &= \frac{10WL}{4} - \frac{WL^2}{8} \\ &= \frac{5WL}{2} - \frac{10W}{L} \left(\frac{L}{2} \right) \left(\frac{L}{4} \right) = \frac{10WL}{8} \\ &= 4,250,000 \text{ in.-lb.} \end{aligned} \quad (4.6)$$

The maximum stress

$$S_{\max} = \frac{Mc}{I_{x0}}, \quad (4.7)$$

where c = the distance from the neutral axis to the extreme or outermost fiber = $h_1/2$, in.

$$S_{\max} = 2450 \text{ psi} .$$

The irradiator as a simple beam will withstand the required loading.

External Pressure

The regulations state that the package shall be adequate to assure that the containment vessel will suffer no loss of contents (that is, it will not leak) if subjected to an external pressure of 25 psig. In the case of the irradiator analyzed in this study, the cladding around the capsule (ORNL drawing C-RD-2128-A) within the irradiator is considered as being the containment vessel. The maximum stress that might be incurred by the capsule cladding would be a shearing stress in the weld around the end plug of the capsule. The shearing stress on the weld

$$S_s = \frac{F}{A} = \frac{F}{Ct} = \frac{pA}{Ct} = \frac{p(bh)}{2(b+h)t} , \quad (4.8)$$

where

F = total force on the end plug, lb,

A = area of weld securing the plug to the cladding, in.²,

C = perimeter of the end plug, in.,

t = thickness of the weld, in.,

p = pressure = 25 psig.

The maximum allowable shear stress

$$S_{s \max} = 0.55S_{yp} = 16,500 \text{ psi}, \quad (4.9)$$

where S_{yp} = the stress at the yield point of the material. Solving Eq. 4.8 for t,

$$t = \left(\frac{p}{S_s} \right) \frac{bh}{2(b+h)} = 0.000164 \text{ in. (minimum)} .$$

We do not know what the actual thickness of the end-plug weld in the existing irradiator is and no thickness is specified on the drawings. However, we cannot conceive of a method whereby a weld as small as the calculated thickness could be produced. We therefore conclude that the actual thickness of the weld exceeds the calculated value and that the containment vessel or capsule cladding will suffer no loss of contents if subjected to an external pressure of 25 psig.

5. COMPLIANCE WITH STANDARDS FOR NORMAL CONDITIONS OF TRANSPORT

The regulations for normal conditions of transport for a single package require that the effectiveness of the package will not be reduced substantially by normal conditions of transport and that there will be no release of radioactive material from the containment vessel. The contents of the irradiator analyzed in this study are so limited that there will be no gases or vapors in the package that could reduce the effectiveness of the packaging. There is no circulating coolant other than atmospheric air provided for in this irradiator, and there is no mechanical cooling device required or provided. No fissile material will be shipped in the irradiator, and the containment vessel or the cladding of the capsule is so designed that it will not be vented to the atmosphere under normal conditions of transport. These normal conditions include the effects of heat, cold, pressure, free drop, and penetration.

Heat

The irradiator must be able to withstand direct sunlight at an ambient temperature of 130°F in still air without reducing the effectiveness of the packaging. To analyze the effects of this condition, it was assumed that the irradiator is sitting in an upright position and that the temperature condition has reached a steady state; that is, the rate of solar heat absorption of the irradiator, Q_s , is equal to the rate of solar heat transfer from the irradiator to the atmosphere by convection, Q_c . It was also assumed that the emissivity, e , of the irradiator = 0.3, the convection coefficient, U , of still air = 1 Btu/hr·ft²·°F, and that the average solar heat flux⁵ $q_s = 217$ Btu/hr·ft².

The area of the irradiator receiving solar radiation, A_t , was assumed to be the top of the irradiator (that portion seen in plan view) whose area equals 24.26 ft². The area of the irradiator transferring heat to the atmosphere by convection, A_c , is the total area of the irradiator. This total area is 110.32 ft². The solar heat absorbed by the

irradiator

$$Q_s = q_s e A_t = 1579 \text{ Btu/hr} . \quad (5.1)$$

The solar heat transferred from the cask to the atmosphere by convection

$$Q_c = U A_c \Delta T \quad (5.2)$$

or

$$\Delta T = \frac{Q_c}{U A_c} ,$$

but

$$Q_c = Q_s . \quad (5.3)$$

Therefore,

$$\Delta T = \frac{Q_s}{U A_c} = 14.30 \text{ } ^\circ\text{F} .$$

The average temperature of the shell of the irradiator

$$T_s = 130 + \Delta T = 144.3^\circ\text{F} . \quad (5.4)$$

If it is assumed that the air in the shielding cavity reaches this temperature and that the cavity was sealed at a temperature of 70°F and a pressure of 14.7 psia, by the perfect gas law where a constant volume is assumed and

p = pressure, psia,

T = temperature, $^\circ\text{R}$;

the final pressure

$$p_2 = \frac{p_1 T_2}{T_1} = 16.76 \text{ psia} \quad (5.5)$$

where $T_2 = T_s = 144.3^\circ\text{F}$.

No exposed surface of the irradiator will be at a temperature in excess of 144.3°F when fully loaded and subjected to an ambient temperature of 130°F in still air, and these temperatures do not exceed the operating temperature limits of any of the materials of the irradiator. Neither the resulting increase in pressure of 2.0 psia nor the temperatures will reduce the effectiveness of the irradiator under the stipulated normal conditions of heat.

Cold

The irradiator must be able to withstand an ambient temperature of -40°F in still air and shade. Using the same methods of analysis used for the condition of heat where $-40^{\circ}\text{F} = 420^{\circ}\text{R}$,

$$p_2 = \frac{p_1 T_2}{T_1} = 11.65 \text{ psia} . \quad (5.6)$$

An outside ambient temperature of -40°F would reduce the temperature of the hot condition by 170°F , and the resulting pressure differential of 3.0 psia and temperature differential are within the operating pressure and temperature range of the materials of the irradiator. The range of the temperature is not great enough to significantly change the thermal conductivities of these materials. When loaded, brittle fracture is not credible because the temperatures of all the components of the irradiator are well above the ductile-brittle transition temperatures of the materials from which the components are constructed. Therefore, the condition of cold will not reduce the effectiveness of the packaging, and the irradiator conforms with this condition of normal transport.

Pressure

The regulations for normal conditions of transport specify that the package be able to withstand an atmospheric pressure of 0.5 times the standard atmospheric pressure, the resulting pressure being 7.35 psia. This is a more severe requirement than either of those governing temperature conditions. The weak point in the irradiator is probably the long horizontal side. Since stress on this geometric configuration cannot be feasibly computed, the more conservative case of a plate 41 in. wide, 73 in. long, and 1/2 in. thick with its edges supported was assumed. From Case 36 on pages 217 and 218 in Ref. 6, the maximum stress at the center of the plate

$$S_{\max} = \frac{Bwb^2}{t^2} , \quad (5.7)$$

where

B = factor from table on page 218 of Ref. 6 = 0.61
 where $a/b = 2.07$,

a = length of plate = 73 in.,

b = width of plate = 41 in.,

$w = p$ = pressure, psi,

t = thickness of plate = 1/2 in.

$$S_{\max} = 20,000 \text{ psi} .$$

Since the maximum stress at the center of the plate is less than the yield stress of the material, no significant damage will result from the pressure condition of normal transport. However, it should be pointed out that a more severe case than that which actually exists was assumed for computational purposes.

Free Drop

Any shipping package weighing over 30,000 lb must be able to withstand a free drop through a distance of 1 ft onto a flat essentially unyielding horizontal surface, striking the surface in a position in which maximum damage is expected to result. The energy the cask must absorb upon impact, U_i , is the product of the weight of the cask, W , and the drop height, h .

$$U_i = Wh = 40,000(12) = 480,000 \text{ in.-lb} . \quad (5.8)$$

To study the effects of this condition, the unclad lead model and the stress-strain curve for lead shown in Fig. 6 were assumed. Neglecting elastic action, the impact energy absorbed by the unclad lead model impacting on its end when dropped 12 in.

$$U_i = P\Delta = S_c A \epsilon_{av} L = VS_c \epsilon_{av} = \frac{W}{2\rho} S_c \epsilon_{\max} = Wh , \quad (5.9)$$

where

P = force, lb,

Δ = deformation, in.,

S_c = compressive stress of lead = 2000 psi max,

Fig. 6. Unclad Lead Model and Stress-Strain Curve for Lead Assumed to Investigate Effects of Free Drop Condition of Normal Transport.

A = cross-sectional area of lead model, in.²,

ϵ = strain, in./in.,

L = length of model = 85 in.,

V = volume, in.³,

ρ = density of lead, lb/ft³.

The maximum strain

$$\epsilon_{\max} = \frac{2\rho h}{S_c} = 0.005 \text{ in./in.} \quad (5.10)$$

The average strain

$$\epsilon_{\text{av}} = \frac{\epsilon_{\max}}{2} = 0.0025 \text{ in./in.} \quad (5.11)$$

The resulting deformation

$$\Delta = \epsilon_{\text{av}} L = 0.2125 \text{ in.} \quad (5.12)$$

Therefore, no significant structural damage will result from the normal free-drop condition of transport nor will the shielding geometry of the irradiator be disturbed by a significant amount.

Penetration

The regulations for normal conditions of transport specify that the package must be able to withstand the impact of the flat end of a 1.25-in.-diameter steel cylinder weighing 13 lb that has been dropped from a height of 4 ft normally onto the exposed surface of the package that is expected to be the most vulnerable to puncture. The portion of the irradiator analyzed in this study most vulnerable to impact is the long side. For computational purposes, a simply supported plate 48 in. wide, 85 in. long, and 1/2 in. thick was assumed. To determine the elastic deflection in the plate, the impact energy absorbed

$$U_i = Wh = 13(48) = 624 \text{ in.-lb} = \frac{1}{2} P\Delta . \quad (5.13)$$

Using equations from Case 37 on page 217 and constants on page 218 of Ref. 6, the deflection

$$\Delta = y = \frac{2Pb^2}{Et^2} , \quad (5.14)$$

where

Δ = deflection, in.,

P = applied dynamic force, lb,

b = short dimension or width of plate = 41 in.,

E = modulus of elasticity of steel = 30×10^6 psi,

t = thickness of plate = 1/2 in.

Then,

$$\frac{P\Delta}{2} = \frac{\alpha P^2 b^2}{2Et^3} = 624 , \quad (5.15)$$

where α = a constant from page 218 of Ref. 6 = 0.11 when $a/b = 2.07$

where a = the long dimension of the plate = 85 in. Solving for the applied force

$$P = 5040 \text{ lb} .$$

The maximum stress at the center of the plate

$$S_{\max} = \frac{3P}{2\pi t^2} (1.3 \log \frac{2b}{\pi r_0} + 1 - \beta) , \quad (5.16)$$

where

r_o = the radius of the steel cylinder, in.,

β = a constant from page 21 of Ref. 6 = 0.61.

$$S_{\max} = 30,000 \text{ psi} .$$

Although the yield stress of the steel plate or cladding of the irradiator may be exceeded locally, the damage resulting from impact of the steel cylinder is not significant in that the effectiveness of the shielding of the irradiator will not be reduced by the penetration condition of normal transport.

6. COMPLIANCE WITH STANDARDS FOR HYPOTHETICAL ACCIDENT CONDITIONS

The regulations stipulate that a single package used for shipment of a large quantity of radioactive material shall be so designed and constructed and its contents so limited that when it is subjected to the hypothetical accident conditions, the reduction in shielding will not be sufficient to increase the external radiation dose rate to more than 1000 mr/hr at a distance of 1 meter from the outside surface of the package and no radioactive material will be released from the package except for gases containing total radioactivity not to exceed 0.1% of the total radioactivity of the contents of the package. The hypothetical accident is comprised of the specified free-drop, puncture, thermal, and water-immersion conditions, in that order.

The first accident condition is a free drop of the package through a distance of 30 ft onto a flat unyielding horizontal surface. Then the package shall be subjected to a free drop through a distance of 40 in. to strike in a position in which maximum damage is expected the top end of a vertical cylindrical mild-steel bar mounted on an essentially unyielding horizontal surface. The mild-steel bar shall have a diameter of 6 in. with the top horizontal and its edges rounded to a radius of not more than 1/4 in., and it shall be of such length that it will cause maximum damage to the package but not less than 8 in. long. The long axis of this bar shall be normal to the surface of the package upon impact. The third condition consists of exposure of the package for 30 minutes within a source of radiant heat having a temperature of 1475°F and an emissivity of 0.9 or equivalent. For calculational purposes, it shall be assumed that the package has an absorption coefficient of 0.8. Following thermal exposure, the package shall be immersed in water to a depth of at least 3 ft for a period of 24 hours.

Because of the many uncertainties associated with computing the amount of lead shielding in the irradiator that might melt during thermal exposure, we have taken the position that no rupture or puncture of the steel shell of the irradiator may be permitted to result from the free drop or puncture conditions of the hypothetical accident. With

this position, the water-immersion condition will have no detrimental effect on the shielding capability of the irradiator, and this condition was not investigated further.

From a study of the fabrication drawings of the irradiator, it was apparent that the design for the corner welds was not adequate to permit the irradiator to withstand the 30-ft free fall without rupture. This is especially true of compound corners. Experience has shown that unless back welded, corner welds will rupture if the root is placed in tension by bending. We therefore propose that the irradiator be surrounded by a protective structure or crash frame, as shown in Figs. 7, 8, and 9, for corner protection.

Fig. 7. Plan of Proposed Crash Frame for Irradiator.

Fig. 8. Section A - A Through Proposed Crash Frame.

Fig. 9. Section B - B Through Proposed Crash Frame.

In addition, data from Ref. 7 indicate that the 1/2-in.-thick steel shell of the irradiator is not adequate to withstand the impact on the vertical cylindrical bar resulting from the 40-in. free drop without puncture. To protect the irradiator from puncture, it will be necessary to weld the equivalent of an additional 1/2-in.-thick steel plate to all the flat surfaces. The method of attachment is not significant as long as the added plate is parallel to the existing wall it protects. This additional plate may be spaced any convenient distance from the existing surface without affecting the validity of the protection. Assuming that this protective plate has been added, the hypothetical accident condition of puncture was not investigated further.

Free Drop

The first in the sequence of hypothetical accident conditions to which the package must be subjected is a free drop through a distance of 30 ft onto a flat essentially unyielding horizontal surface, striking the surface in a position in which maximum damage is expected to occur. For analysis of the effects of this condition, the package considered is comprised of the irradiator within the proposed crash frame, as illustrated in Fig. 7. The forces and deflections resulting from the 30-ft free drop of the package must be determined for the possible orientations of the package upon impact.

Corner Impact

Impact on a typical compound corner of the package, as shown by the circled corner in Fig. 7, was considered first. The effect of the corner impact on the crash frame surrounding the irradiator might best be analyzed by using plastic hinge theory, which simply stated says that the force to cause a member to fail in bending times the distance through which this force acts is equal to the summation of the moments at the plastic hinges times the angle through which they act. On impact, eight

plastic hinges, as shown in Fig. 10, must develop if the crash frame is to collapse. The work done by the forces in the Y' and Z' directions

Fig. 10. Collapse Mechanism for Drop on Compound Corner of the Package.

was neglected since the motion is primarily in the X' direction. Because the lengths of all the collapsing members are approximately of the same magnitude, the impact energy absorbed by the crash frame

$$U_i = Wh = 8M\theta_x . \quad (6.1)$$

$$Wh = 20.88 \times 10^6 \text{ in.-lb} , \quad (6.2)$$

where

W = weight of the irradiator and crash frame estimated to be 58,000 lb,
h = drop height = 30 ft = 360 in.

Assuming an idealized stress-strain diagram (a horizontal line), the bending moment to cause complete plastic action in the 10 WF 136 members of the crash frame

$$M = KS_{yp} z , \quad (6.3)$$

where

K = ratio of plastic moment to elastic moment = 1.2 ,

S_{yp} = yield or plastic stress, psi,

z = section modulus of 10 WF 136 in the Y - Y direction.

$$M = 7.4 \times 10^6 \text{ in.-lb.}$$

The angle of rotation in the x direction

$$\theta_x = \frac{Wh}{8M} = 0.353 \text{ radians} = 20.22^\circ. \quad (6.4)$$

An x-direction angle of 20.22° is equivalent to an x deflection

$$\Delta_x = \ell \sin \theta = 13.8 \text{ in.}, \quad (6.5)$$

where

ℓ = length of member = 40 in.,

θ = angle of rotation, $^\circ$.

This deflection of 13.8 in. would not allow the corner of the irradiator to contact the impact surface of the crash frame. It may therefore be concluded that since the corner of the irradiator does not strike the impact surface, the root of the corner weld of the irradiator will not be placed in tension by bending and it will not rupture as a result of the 30-ft free drop and corner impact.

Side Impact on Intermediate Shutter Cave

The 30-ft free fall of the package with the impact surface normal to the intermediate shutter cave (shown in Fig. 1) must be considered. Since the distance from the edge of the crash frame to the end cap is small and the mathematics associated with the true case are extremely involved and difficult, the model consisting of the cave and four 10 WF 136 beams loaded in compression illustrated in Fig. 11 was assumed. It was also assumed that all the strain energy is stored in these members.

The cross-sectional area of the lead

$$A_\ell = (h_1 - 1)(b_1 - 1) - (h_2 + 1)(h_2 + 1), \quad (6.6)$$

where h_1 , h_2 , b_1 , and b_2 are as shown in Fig. 11.

$$A_\ell = 473 - 218 = 255 \text{ in.}^2$$

Fig. 11. Cross Section of Model Assumed to Investigate the Results of Side Impact on Surface Normal to Intermediate Shutter Cave.

The cross-sectional area of the steel cladding

$$\begin{aligned} A_{sc} &= (h_1 + b_1 - t)2t + (h_2 + b_1 + t)2t \\ &= 50.9 + 33.4 = 84.3 \text{ in.}^2 \end{aligned} \quad (6.7)$$

The cross-sectional area of the structural steel beams

$$A_{sb} = 40.33(4) = 160.12 \text{ in.}^2 \quad (6.8)$$

Assuming the stress-strain curves illustrated in Fig. 12, and letting

$$N = \frac{S_{\text{steel}} \epsilon_{\text{steel}}}{S_{\text{lead}} \epsilon_{\text{lead}}} = 5 \quad (6.9)$$

and the steel equivalent area of the lead

$$A'_{\ell} = \frac{A_{\ell}}{5} = 51 \text{ in.}^2 ; \quad (6.10)$$

the total cross-sectional area of the shutter cave

$$A = A'_{\ell} + A_{sc} + A_{sb} = 295.4 \text{ in.}^2 \quad (6.11)$$

Fig. 12. Stress-Strain Curves for Steel and Lead Assumed to Investigate the Results of Side Impact of Irradiator on Surface Normal to the Intermediate Shutter Cave.

The strain energy stored in the crash frame members

$$u = SA\Delta = Wh, \quad (6.12)$$

and the deformation in the members

$$\Delta = \frac{Wh}{SA} = \frac{20.88 \times 10^6}{40,000(295.4)} = 1.766 \text{ in.} \quad (6.13)$$

Taking the length as 34 in., the strain

$$\epsilon = \frac{\Delta}{L} = 0.052 \text{ in./in.} \quad (6.14)$$

Strains of this magnitude do not represent a shell rupture hazard. However, it is recommended that the welds at the base and flange of the intermediate shutter cave be strengthened as shown in Fig. 13. We cannot definitely predict whether the end cap will rupture under these conditions, but rupture of the end cap would be of little consequence as long as the intermediate shutter is in place. The intermediate shutter is

Fig. 13. Recommended Strengthening of Welds at the Base and Flange of the Intermediate Shutter Cave.

held in place by a stainless steel 1/4-in. tapered pin (the taper is not specified) loaded in a single shear. The applied force on the shutter cave caused by impact of the package

$$P = SA = 40,000(295.4) = 11.82 \times 10^6 \text{ lb} . \quad (6.15)$$

The acceleration of the package

$$a = \frac{Pg}{W} , \quad (6.16)$$

where

g = gravitational constant = 32.2 ft/sec^2 ,

W = weight of irradiator in its crash frame = $58,000 \text{ lb} .$

$$a = \frac{(11.82 \times 10^6)32.2}{58,000} = 6562 \text{ ft/sec}^2 .$$

The force applied to the pin securing the intermediate shutter

$$P = \frac{W_{is} a}{g}, \quad (6.17)$$

where

$$\begin{aligned} W_{is} &= \text{weight of intermediate shutter} \\ &= 2190 \text{ lb (from Appendix A),} \\ a &= \text{acceleration of package} = 6562 \text{ ft/sec}^2, \\ g &= \text{gravitational constant} = 32.2 \text{ ft/sec}^2. \end{aligned}$$

$$P = \frac{2190(6562)}{32.2} = 446306 \text{ lb.}$$

The total area of the pin required to secure the intermediate shutter

$$A = \frac{P}{0.55S_u}, \quad (6.18)$$

where S_u = the ultimate strength of 300-series stainless steel
= 75,000 psi.

$$A = \frac{446306}{0.55(75,000)} = 10.8 \text{ in.}^2$$

The existing pin is therefore inadequate to secure the door under the 30-ft free-drop side-impact conditions. The required area could be halved by loading in double shear, or it could be reduced by using high-strength material.

Side Impact on Produce Input Shutter

The produce input shutter (illustrated in Fig. 1) is secured by bolts, as shown in Fig. 14. The fabrication drawings for the irradiator do not specify the number, size, or material of these bolts. We are therefore unable to say whether or not the irradiator complies with the regulations governing the free-drop resulting in a side impact on the produce input shutter. Using high-strength (160,000 psi ultimate strength) bolting material, the bolt area required to show compliance with the regulations was computed. The force applied to the bolts securing the produce input shutter resulting from the 30-ft free drop and side impact on the input shutter

ORNL DWG 69-9550

Fig. 14. Produce Input Shutter Closure.

$$P = \frac{W_{ps} a}{g}, \quad (6.19)$$

where

W_{ps} = weight of produce input shutter
 = 2710 lb (from Appendix A),
 a = acceleration of package = 6562 ft/sec²,
 g = gravitational constant = 32.2 ft/sec².

$$P = \frac{2710(6562)}{32.2} = 552,279 \text{ lb}.$$

The required minimum area of the bolt group

$$A = \frac{P}{2S_u}, \quad (6.20)$$

where S_u = the ultimate strength of the bolt material = 160,000 psi.

$$A = \frac{552,279}{2(160,000)} = 1.73 \text{ in.}^2$$

On site inspection of the bolting for the produce input shutter will verify compliance with the regulations.

Top Impact

A top impact orientation of the package must be considered to determine whether the source access door will remain closed after impact. In a top impact orientation, the package will strike on the four vertical extensions of the crash frame surrounding the irradiator. These extensions will transmit part of the load directly to the four vertical columns at the corners of the crash frame. These columns have a ratio

$$\frac{L}{r} = 18.2 ,$$

where

L = the unsupported length of the column = 49.6 in.,

r = the least radius of gyration of the column = 2.72 in.

Therefore, these columns will not buckle. The first failure would then be the collapse of the four long crash frame columns. The collapse equation may be written

$$U_i = P\Delta = 4(4M\theta) = \frac{\theta}{\frac{\ell}{2}} = 16M\theta , \quad (6.21)$$

where

U_i = impact energy absorbed by the four long crash frame columns, in.-lb ,

P = applied force, lb,

Δ = deflection or deformation, in.,

M = applied moment of force, in.-lb,

θ = angle through which the member rotates, °,

ℓ = effective length of the column = 95 in.

The total force applied to the long columns

$$P_1 = \frac{32M}{\ell} = \frac{32KS_{yp}z}{\ell} = 905,000 \text{ lb} , \quad (6.22)$$

where

K = ratio of the maximum elastic moment to the final plastic moment = 1.2 ,

S_{yp} = the yield or plastic stress, psi,

z = the section modulus of the column.

Similarly, the four short beams would collapse and the total force applied to the short beams

$$P_2 = \frac{32KS_{yp}z}{\ell} = 1.21 \times 10^6 \text{ lb} , \quad (6.23)$$

where ℓ = the effective length of the beam = 71 in. The total applied force

$$P = P_1 + P_2 = 2.12 \times 10^6 \text{ lb} . \quad (6.24)$$

Observation of the geometry of the crash frame indicates that a deflection, Δ , of 10 in. is possible before the center of the columns contact the impact surface. Since the impact energy that can be absorbed by the crash frame columns

$$U_i = P\Delta = 21.2 \times 10^6 \text{ in.-lb} ,$$

it may be concluded that the columns have the capacity to absorb all of the impact energy previously computed.

$$U_i = Wh = 58,000(360) = 20.88 \times 10^6 \text{ in.-lb} . \quad (6.2)$$

The equivalent static force applied to the source access door

$$P' = \frac{W_{sd}a}{g} = \frac{W_{sd}}{g} \left(\frac{Pg}{W} \right) = \frac{W_{sd}P}{W} , \quad (6.25)$$

where

W_{sd} = weight of source access door

= 2470 lb (from Appendix A),

P = total applied force = 2.12×10^6 lb,

W = weight of the irradiator in the crash frame = 58,000 lb .

$$P' = \frac{2470(2.12 \times 10^6)}{58,000} = 90,300 \text{ lb} .$$

The source access door is secured by four 3/4-in.-diameter bolts with an area, A , of 1.336 in.² The stress

$$S = \frac{P'}{A} = \frac{90,300}{1.336} = 67,589 \text{ psi} . \quad (6.26)$$

We have no knowledge of the bolt material used so the existing bolts should be replaced with high-strength bolts (material with a yield stress of 160,000 psi). It would also be advisable to add additional bolts at the top on the far side of the source access door to eliminate the

possibility of a moment being applied to these bolts. An equivalent closure is required for the produce input shutter.

Thermal

The package must be exposed for 30 minutes within a source of radiant heat having a temperature of 1475°F and an emissivity coefficient of 0.9 or equivalent. For calculational purposes, it shall be assumed that the package has an absorption coefficient of 0.8. The package shall not be cooled artificially until after the 30-minute exposure period has expired and the temperature at the center of the package has begun to fall.

A SIFT-TOSS program for an IBM-7090 computer was used with the engineering model illustrated in Fig. 15 and an assumed initial cask temperature of 150°F to determine the effects of the 30-minute thermal test on the irradiator. The computer solution indicated an average outside shell

Fig. 15. Computational Model of Irradiator Used to Determine the Effects of the 30-Minute Thermal Accident Condition.

temperature of 1112°F and an average lead temperature of 865°F. However, the SIFT-TOSS program is inaccurate for this analysis since there is no provision for accounting for the latent heat of fusion of the lead that may be melted during the 30-minute thermal test. The computer solution must therefore be corrected.

The lead superheat, H_s , is defined as the "apparent" quantity of input heat required to raise the temperature of the lead from its melting point to the computed average temperature. Taking the temperature of the melting point of lead as 621°F, we may say that the superheat temperature rise

$$\Delta T = T_m - 621^\circ\text{F} = 244^\circ\text{F} , \quad (6.27)$$

where T_m = the average computed temperature of the lead = 865°F.

$$H_s = mC_p\Delta T , \quad (6.28)$$

where

m = mass of the lead,

C_p = specific heat of molten lead = 0.038 Btu/lb·°F .

Taking m as 1, the specific superheat

$$H'_s = 9.27 \text{ Btu/lb} .$$

Comparing H'_s with the latent heat of fusion of lead, which is 10.7 Btu/lb, it can be recognized that the major portion of the lead will melt during the 30-minute thermal test. It is therefore conservative to assume that all of the lead will melt and have an average temperature of 621°F. The total initial volume of lead, V_i , is the difference between the volume enclosed by the outer shell and the volume of the cavity of the irradiator.

$$V_i = V_{os} - V_c \quad (6.29)$$

$$= 122,000 - 40,700 = 81,000 \text{ in.}^3$$

The volume increase factor

$$f = \frac{V_f}{V_i} = \frac{\rho \text{ of solid lead at } 70^\circ\text{F}}{\rho \text{ of molten lead at } 621^\circ\text{F}} , \quad (6.30)$$

where

V_f = final volume, in.³,

ρ = density = 1/V .

The volumetric increase in the lead

$$\begin{aligned}\Delta V_{\ell} &= V_f - V_i = V_i f - V_i (f - 1) \\ &= 6400 \text{ in.}^3\end{aligned}\quad (6.31)$$

For rectangular geometry, an expression for the increase in the volume of the outer shell, ΔV_{os} , of the irradiator caused by thermal expansion may be written as follows.

$$\begin{aligned}\Delta V_{os} &= V_{os \text{ final}} - V_{os \text{ initial}} \\ &= \Sigma [(\ell + \alpha_s \ell \Delta T)(w + \alpha_s w \Delta T)(b + \alpha_s b \Delta T) - \ell w b] \\ &= \Sigma [\ell(1 + \alpha_s \Delta T)w(1 + \alpha_s \Delta T)b(1 + \alpha_s \Delta T) - \ell w b] \\ &= \Sigma \ell w b [(1 + \alpha_s \Delta T)^3 - 1],\end{aligned}\quad (6.32)$$

where

ℓ , w , and b = linear dimensions of the rectangular components, in.,

α_s = the coefficient of thermal expansion for steel

= 8.4×10^6 in./in. $^{\circ}$ F,

ΔT = temperature increase, $^{\circ}$ F.

In general, it can be seen that

$$\Delta V_{os} = V_i (1 + \alpha_s \Delta T)^3 - 1. \quad (6.33)$$

In this specific case,

$$\Delta V_{os} = 3074 \text{ in.}^3,$$

where $\Delta T = (1112 - 70) = 1042^{\circ}$ F. Similarly, the volumetric increase of the inner cavity of the irradiator

$$\Delta V_{ic} = V_{ic \text{ initial}}(1 + \alpha_s \Delta T)^3 - 1 = 545 \text{ in.}^3, \quad (6.34)$$

where $\Delta T = (621 - 70) = 451^{\circ}$ F.

If any gap that exists between the lead and the steel shell is neglected, the initial volume of air entrapped in the shielding cavity, V_{ai} , is the volume of the seven expansion pans shown on ANC drawing CM-7017. This volume is numerically equal to 4258 in.^3 . It was assumed that the shielding cavity was sealed at a temperature of 70° F and a pressure of 14.7 psia. It was further assumed that after the 30-minute thermal exposure, the outer shell of the irradiator expands to such an

extent that the pressure within the cavity is still 14.7 psia. Then, by the perfect gas laws,

$$\frac{V_{ai}}{T_i} = \frac{V_{af}}{T_f}, \quad (6.35)$$

$$V_{af} = \frac{V_{ai} T_f}{T_i} = 9235 \text{ in.}^3,$$

where

V_{af} = final volume of air entrapped in the shielding cavity, in.³,

T_f = final temperature = 1081°R (melting temperature of lead),

T_i = initial temperature = 530°R.

The required increase in the volume of the outer shell

$$\begin{aligned} \Delta V &= \Delta V_\ell - \Delta V_{os} + \Delta V_{ic} - V_{ai} + V_{af} \\ &= 8578 \text{ in.}^3 \end{aligned} \quad (6.36)$$

This volume increase will be effected primarily by deflection of the flat plates making up the outer shell of the irradiator. Since deflection is proportional to b^2 , where b is the smaller of the plate dimensions and a is the larger, certain smaller plates may be neglected and only those tabulated below need be considered.

<u>Location</u>	<u>Size (in.)</u>	<u>Area (ab) (in.³)</u>
Bottom	41 by 85	3485
Top	41 by 56	2295
Front	37 by 83.5	3085
Back right	36 by 37	1332
		<u>$\Sigma ab = 10201$</u>

We have little knowledge of the laws of gross deformation of flat plates at elevated temperatures. Therefore, the following assumptions, which appear reasonable to us, were made without any attempt to prove them.

1. The deflected surface is similar to a segment of an ellipsoid.
2. The semi-axes of the approximate ellipse projected on the original flat surface are a and b , the length and width of the flat plate.
3. The volume of a segment of the ellipse, V_e , is approximately equal to

$$\frac{1}{3} \pi ab(\delta) , \quad (6.37)$$

where δ = the maximum deflection.

4. The maximum deflection, δ , does not vary significantly from plate to plate. This would be true for plates at normal temperatures within the elastic limit since the variation in b is small. The elastic normal-temperature equation for maximum deflection

$$\delta = \frac{\alpha_s P b^2}{E t^3} , \quad (6.38)$$

where $0.016 > \alpha > 0.027$. Under these conditions,

$$\begin{aligned} \Delta V &= \sum V_e \\ &= \sum \frac{1}{3} \pi ab(\delta) . \end{aligned} \quad (6.39)$$

The maximum deflection

$$\delta = \frac{3\Delta V}{\pi \sum ab} = 0.8 \text{ in.}$$

At the temperatures resulting from the stipulated thermal exposure conditions, a deflection of several times the computed value of 0.8 in. would not present a shell rupture hazard. We therefore conclude that the irradiator analyzed in this study will withstand the thermal condition of the hypothetical accident. It should be noted that no allowance was made for the resistance of heat flow offered by the proposed puncture protection plates and the resulting air gap, the air gap which probably exists between the outer shell and the lead, and by the gaps resulting from the expansion pans. The end doors on the irradiator need not be analyzed since they present a less severe case than the outer shell because of the absence of corners exposed to the heat.

REFERENCES

1. United States Atomic Energy Commission AEC Manual, Volume 0000, General Administration, Part 0500, Health and Safety, Chapter 0529, "Safety Standards for the Packaging of Radioactive or Fissile Materials," August 22, 1966.
2. Code of Federal Regulations, Title 10, Part 71, Transport of Licensed Materials; also see Federal Register, Vol. 31, pp. 9941-9949, July 22, 1966.
3. E. Oberg and F. D. Jones, Machinery's Handbook, The Industrial Press, New York, 17th ed., 1964.
4. "Wood Handbook", U.S. Department of Agriculture Handbook No. 72, U.S. Government Printing Office, Washington, D. C., 1955.
5. J. H. Evans et al., "Analysis of HFIR Irradiated Fuel Element Shipping Cask," USAEC Report ORNL TM-959, Oak Ridge National Laboratory, January 1965.
6. R. J. Roark, Formulas for Stress and Strain, Mc-Graw-Hill Company, Inc., New York, 1954.
7. A. E. Spaller, "Structural Analysis of Shipping Casks, Vol. 2, Resistance to Puncture," USAEC Report ORNL TM-1312, Vol. 2, Oak Ridge National Laboratory, September 1966.

APPENDICES

Appendix A

MISCELLANEOUS CALCULATIONS

The weight of the irradiator body

$$W = V\rho , \quad (A.1)$$

where

V = the volume, in.³,

ρ = the density of lead = 0.41 lb/in.³

$$\begin{aligned} W &= \rho(V_{\text{overall body}} - V_{\text{cavity}} + V_{\text{intermediate shutter}} \\ &\quad + V_{\text{intermediate shutter recess shielding}} - V_{\text{recess cavity}}) \\ &= 38,000 \text{ lb} . \end{aligned}$$

The weight of the intermediate shutter (drawing DM-9023)

$$W_{is} = V\rho = 2190 \text{ lb} . \quad (A.2)$$

The weight of the produce input shutter (drawing DM-9022)

$$W_{ps} = V\rho = 2710 \text{ lb} . \quad (A.3)$$

The weight of the source access door (drawing DM-9024)

$$W_{sd} = V\rho = 2470 \text{ lb} . \quad (A.4)$$

The centroid of the irradiator with respect to the xy plane

$$\bar{V}_y = \Sigma V_y . \quad (A.5)$$

$$\begin{aligned} \bar{V}_y &= \left[(V_{\text{intermediate shutter recess shield}})(\text{xy distance to centroid}) \right. \\ &\quad \left. + (V_{\text{intermediate shutter recess shield}}) \left(\frac{\text{xy distance to centroid}}{V_{\text{total}}} \right) \right] \\ &= 5.7 \text{ in.} \end{aligned}$$

Appendix B

NOMENCLATURE

- A = area, in.²
 A_B = cross-sectional area of one bolt, in.²
 A_c = area of irradiator transferring heat to the atmosphere by convection, in.²
 A_ℓ = area of lead, in.²
 A_s = equivalent area of steel, in.²
 A_t = area of irradiator receiving solar radiation, in.²
 a = acceleration, ft/sec²
 C = perimeter of end plug, in.
 C_p = specific heat, Btu/lb.°F
 c = the distance from the neutral axis to the outermost fiber, in.
 D = diameter, in.
 d = distance from the neutral axis to the most distant bolt, in.
 d' = distance from the neutral axis to any bolt, in.
 E = modulus of elasticity, lb/in.²
 E_ℓ = modulus of elasticity of lead, lb/in.²
 E_s = modulus of elasticity of steel, lb/in.²
 E_w = modulus of elasticity of wood, lb/in.²
 e = emissivity
 F = force in the most distant bolt from the neutral axis, lb
 F = total force on the end plug, lb
 F_B = force on any bolt, lb
 f = volume increase factor
 g = gravitational constant = 32.2 ft/sec²
 H_s = lead superheat or "apparent" quantity of heat required to raise the temperature of lead from the melting point to the computed average temperature, Btu
 h = drop height, in.
 I = moment of inertia, in.⁴
 K = ratio of plastic moment to elastic moment
 L = length, in.
 M = moment of force, in.-lb

- m = mass
 n = number of bolts at any particular d'
 P = applied force or concentrated load, lb
 p = pressure, psig or psia
 Q_c = rate of solar heat transfer from irradiator to atmosphere by convection, Btu/hr
 Q_s = rate of solar heat absorption by irradiator, Btu/hr
 q_s = heat flux, Btu/hr·ft²
 R = modulus of elasticity of steel divided by modulus of elasticity of lead = E_s/E_ℓ
 r = radius of gyration, in.
 S = stress, psi
 S_c = compressive stress, psi
 S_n = normal stress, psi
 S_s = shearing stress, psi
 S_t = tensile stress, psi
 S'_w = compressive stress in wood variable from 0 at the neutral axis to a maximum at $a' = a$, psi
 S_w = maximum compressive stress in wood at $a' = a$, psi
 S_{yp} = stress at yield point of material, psi
 T = temperature, °F or °R
 T_f = final temperature, °F or °R
 T_i = initial temperature, °F or °R
 T_m = average computed temperature, °F
 T_s = average temperature of the shell of the irradiator, °F
 t = thickness, in.
 U = convection heat transfer coefficient, Btu/hr·ft²·°F
 U_i = energy absorbed by irradiator or crash frame upon impact, in.-lb
 V = volume, in.³
 V_{ai} = initial volume of air entrapped in the shielding cavity, in.³
 V_{af} = final volume of air entrapped in the shielding cavity, in.³
 V_e = volume of segment of ellipse, in.³
 V_f = final volume, in.³
 V_i = initial volume, in.³
 V_{ic} = volume of inner cavity, in.³

V_{ℓ} = volume of lead, in.³
 V_{os} = volume of outer shell, in.³
 W = weight, lb
 W_{is} = weight of intermediate shutter, lb
 W_{ps} = weight of produce input shutter, lb
 W_{sd} = weight of source access door, lb
 w = load for unit length, lb/in.
 z = section modulus of member

 α = constant from Ref. 6
 α_s = coefficient of thermal expansion for steel, in./in..^{°F}
 β = constant from Ref. 6
 Δ = deflection or deformation, in.
 δ = maximum deflection, in.
 ϵ = strain, in./in.
 ϵ'_s = strain in steel at any point, in./in.
 ϵ_s = maximum strain in steel, in./in.
 ϵ'_w = strain in wood at any point, in./in.
 ϵ_w = maximum strain in wood, in./in.
 θ = angle of rotation, °
 ρ = density, lb/ft³

Internal Distribution

1. M. Bender	21. R. A. Robinson
2. C. E. Bettis	22. J. D. Rollins
3. D. D. Cannon	23. R. D. Seagren
4. G. B. Child	24. L. B. Shappert
5. F. C. Davis	25. W. C. Stoddart
6. H. G. Duggan	26. D. A. Sundberg
7-11. J. H. Evans	27. W. E. Unger
12. W. F. Ferguson	28. H. K. Walker
13. F. C. Fitzpatrick	29. D. C. Watkin
14. W. R. Gall	30. W. R. Winsbro
15. J. H. Gillette	31-32. Central Research Library
16. K. W. Haff	33. Document Reference Section
17. A. R. Irvine	34. GE Division Library
18. E. M. King	35-44. Laboratory Records Department
19. B. B. Klima	45. Laboratory Records, ORNL R.C.
20. W. A. Pate	46. ORNL Patent Office

External Distribution

- 47-51. J. Allentuck, AEC, New York, New York
52. R. F. Barker, AEC, Washington, D. C.
53. W. G. Belter, AEC, Washington, D. C.
54. P. Bezler, Brookhaven National Laboratory, Upton, L. I., New York
55. D. C. Costello, AEC, Washington, D. C.
56. J. Cusack, Brookhaven National Laboratory, Upton, L. I., New York
57. G. Deitz, AEC Division of Isotope Development, Washington, D. C.
58. E. E. Fowler, AEC, Washington, D. C.
59. J. C. Glynn, AEC, Washington, D. C.
60. W. A. Johnson, AEC, Oak Ridge, Tennessee
61. J. W. Jones, Bettis Atomic Power Laboratory, Post Office Box 79, West Mifflin, Pa. 15122

62. J. A. Lenhard, AEC, Oak Ridge, Tennessee
63. J. A. Lieberman, AEC, Washington, D. C.
64. E. C. Lusk, Battelle Memorial Institute, 505 King Avenue,
Columbus, Ohio 43201
- 65-69. J. S. Milau, Brookhaven National Laboratory, Upton, L. I.,
New York
70. W. A. Nelson, AEC, Washington, D. C.
71. D. E. Patterson, AEC, Washington, D. C.
72. W. A. Pryor, AEC, Oak Ridge, Tennessee
73. J. A. Sisler, AEC, Washington, D. C.
74. M. Steinberg, U. S. Department of the Interior, Washington, D. C.
- 75-89. Division of Technical Information Extension
90. Laboratory and University Division, AEC, Oak Ridge, Tennessee