Alpha-Omega Servic s, Inc.

CONSOLIDATED APPLICATION,

DESIGN REPORT AND DESCRIPTIVE

DRAWINGS

MODEL 5979

TELETHERAPY COBALT SOURCE

SHIPPING PACKAGE

DOCKET 71-5979

CERTIFICATE OF COMPLIANCE

597

JUNE 1980

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Alpha-Omega Services, Inc.

NUCLEAR • MEDICAL • INDUSTRIAL • HEALTH SERVICES

U.S. NUCLEAR REGULATORY COMMISSION
Division of Fuel Cycle and
Material Safety
Washington, D.C. 20555

Attention: C.E. MacDonald, Chief
Transportation Certification Branch

Subject: Alpha-Omega Services Model 5979 Package
Docket 71-5979
Renewal Application

Gentlemen:

Certificate of Compliance No. 5979, Revision 2 for the Alpha-Omega Services (AOS) Model 5979 Teletherapy Cobalt Source Shipping Package expires on June 30, 1980. It is our desire to renew Certificate No. 5979 and accordingly the following are enclosed:

1. A check in the amount of $150.00 for renewal fee as prescribed in 10CFR170.31.

2. Nine (9) sets of reduced consolidated descriptive drawings of the Model 5979 Package.

   AOS #0090 General Arrangement
   AOS #0091 Container/Transfer Cask
   AOS #0092 Details, inserts (turrets)
   AOS #0093 Details, overpack and pallet

3. Nine (9) copies of the consolidated application and Design Report.

Regarding Item (3) above, it is our understanding that you require a single document which consolidates the original application and all subsequent supplements referenced in the Certificate. The original application for the Model 5979 Package was a letter dated March 20, 1975. The only supplement was submitted May 28, 1975 in compliance with NRC's request and consisted of: 1) the package Design Report earlier submitted to the DOT, and,2) procedural controls for the Model 5979 Package. The

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enclosed Item (3), accordingly, combines all Certificate-referenced submittals into a single document. For your convenience, we have enclosed one (1) set of full-size consolidated drawings, Item (2) above.

We also request some minor adjustments to the renewed Certificate. Since AOS no longer ships unencapsulated ^{60} Cobalt, reference to it in paragraph 5(b)(1) is unnecessary and it may be deleted. Additionally, the reference to gaskets in paragraph (7) may also be deleted as this requirement arose from the shipping of unencapsulated cobalt material.

We trust that the enclosed is sufficient for renewal of Certificate No. 5979. If you or your staff have any questions on this application, please do not hesitate to call.

Respectfully submitted;

Elliot J. Beck, RT
Technical Services Manager

EJB/elb

Enclosures

Alpha-Omega Services, Inc.

NUCLEAR • MEDICAL • INDUSTRIAL • HEALTH SERVICES

March 20, 1975

Transportation Branch, Division of
Materials and Fuel Cycle Facility Licensing
U.S. Nuclear Regulatory Commission
Washington, D.C. 20555

Attention: Special Permits

Reference: D.O.T. Special Permit No. 5979

Dear Sir:

We are in receipt of letter from Department of Transportation which informed us that they no longer handle this permit and advised us to forward our request to you (attached letter).

According to their instructions we are hereby forwarding our application to you for renewal of our permit.

If there are any questions, please contact me.

Very truly yours,

Alpha Omega Services, Inc.

G. E. Harwood

George E. Harwood
President

Enclosure

GEH:dr

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Alpha-Omega Services, Inc.

NUCLEAR • MEDICAL • INDUSTRIAL • HEALTH SERVICES

Procedural Controls

Shipping Container Teletherapy Cobalt

I. Opening & Closeing instructions.

A. Because these containers are also source transfer devices and telecobalt "off license site" handling requires individuals be specificly named in a license to do transfers, unpackaging instructions are not provided.

B. Each container is prominently marked "Caution - Not to be opened until A.O.S. personnel are present." These instructions insure that trained personnel with detection meters and knowledge of container configuration are available to survey and inspect condition before movement into teletherapy room.

C. Repackaging of spent source into container is part of transfer procedure and is done by licensed transfer personnel who also provide gamma leakage survey and affixing of Radioactive III labels.

D. Packaging of outbound shipments from point of manufacture is a "Hot Cell" operation.

E. Unpackaging of inbound spent sources, at point of manufacture is a "Hot Cell" operation.

II. Inspection procedure.

A. 1. At time of loading for outbound shipment the steel & lead container is inspected to insure make up of inserts, plugs, & end caps. Condition of bolts is determined and bolts are replaced as required.

2. Source is loaded into containers in "Hot Cell" and end caps installed.

3. Loaded container is removed from "Hot Cell" & surveys conducted and inner container labels attached.

May 13, 1975

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4. Loaded container is placed in over jacket which has been inspected for condition of attachment bolts and lift points.

5. Complete closure of overjacket is made and gamma leakage surveys conducted so labels can be completed and attached.

B. 1. At site, inspection consists of visual inspection for general condition of container system and specific attention to bolts affecting the closure.

2. Discrepancies if any are noted by field personnel and transmitted to point of manufacture personnel for corrective action.

C. 1. At site, inspection of spent source packaging for return and disposal, consists of insuring all closures are complete and all bolts are in place and tight. Any discrepancies noted on receiving are corrected.

2. Closed container with over jacket is surveyed, labels completed and affixed.

III. Periodic inspections & maintenance.

A. 1. Records of each roundtrip use of each container system are kept.

2. Notes of maintenance performed are part of shipment record.

B. 1. Approximately every two years each container is removed from service. Paint is removed, welds are visualy inspected on inner container and over jacket, moveing parts cleaned, reassembled and system repainted.

IV. Service history & performance.

A. 1. Since 1969 the only maintenance container systems have required is replacement of skids (dwg. D0100-21) which have been replaced with ¼" wall 4" x 4" steel tube welded in place. Other than above, bolt replacement is the only recurring maintenance requirement.

DESIGN OF A FAMILY OF SHIPPING CONTAINERS

FOR THE

TRANSFER OF TELETHERAPY COBALT SOURCES

SUPPLEMENT NUMBER 1

A REPORT SUBMITTED TO:

DEPARTMENT OF TRANSPORTATION

OF THE

UNITED STATES OF AMERICA

BY:

INTERNATIONAL CHEMICALS AND NUCLEAR CORP.

REPORT, STUDIES AND DESIGN PREPARED BY:

R. GRABOVAC & ASSOC.
W. K. BODGER - CONSULTANT
CONSOLIDATED DEVICES, INC.
818 EAST BROADWAY
SAN GABRIEL, CALIFORNIA 91776

ABSTRACT

A family of shipping containers has been designed to provide additional protection for a family of transfer casks. These containers are designed to protect the casks in normal handling and also in the series of four tests provided for in the Department of Transportation regulations.

An exoskeleton type design is used. Impact rigidity is provided by a bolted-up steel frame, in conjunction with 6 flat plywood panels. Fire protection is provided by these plywood panels, and by the steel sheathing on their outer surfaces. Puncture protection is provided by the steel sheathing. Immersion protection is provided by using water resistant materials, and by maintaining the integrity of the cask.

B. A 40 inch drop onto a 6-inch-diameter by 8-inch-long carbon-steel spike.

C. An ASTM standard 1-hour fire.

D. A 24-hour submersion of the container in water to a depth of 3 feet over the uppermost portion of the container without leakage of the contents or loss of any shielding.

2. Each shipping container should be reusable, under normal conditions.

3. The design should be adequate for a family of shipping containers. These containers should be similar in design, differing from one another in minor details in order to accommodate the several members of a family of casks.

4. The design should utilize, to the maximum extent possible, the experience gained from the shipping containers built by International Chemical and Nuclear Corp. and previously approved (#1779) by the Bureau of Explosives of the Association of American Railroads.

Items 1A and 1C above are considered to be the most severe design requirements. Hence the design is based primarily on satisfying these conditions. Provision for satisfying requirement 1B is made also, although this has exerted less influence on the design than the other conditions. It is implicitly assumed that a design which is satisfactory in these respects will also satisfy the requirements of item 1D.

III. CONSIDERATIONS OF FIRE RESISTANCE

Sisler, in Reference 1, reports tests of a collective total of 20 samples in a series of 5 test fires. These tests all simulated the ASTM standard 1 hour fire. Of these 20 samples, 18 were made of wood:

10 Douglas Fir Plywood
2 Solid Douglas Fir
4 Redwood Plywood
2 Solid Redwood

Examination of the results of these tests leads to several conclusions:

1. Douglas fir, either plywood or solid, and redwood plywood all burn away about 2 to 2 1/2, and sometimes 3, inches in the fire test.

2. Solid redwood burns more than the other three.

3. Plywood is preferable, since solid fir and solid redwood are both subject to splitting in the fire test.

4. Maximum interior temperatures are well below the melting temperature of lead.

In addition to the fire tests, Reference 1 also reports a series of tests in which samples of wood were charred or burned under controlled conditions. The samples were exposed to a radiant heat input to simulate the fire temperature, and the progressive charring of the wood was measured with thermocouples. The results of these tests, shown in Figure 1, indicate that burning progresses 2 to 2 1/2 inches in 1 hour, confirming the fire test results.

The results of the controlled tests also suggest that the penetration is greater with plywood than with solid fir or redwood. Due to the occurrence of splitting with solid wood, however, plywood is preferable in spite of a slightly higher penetration.

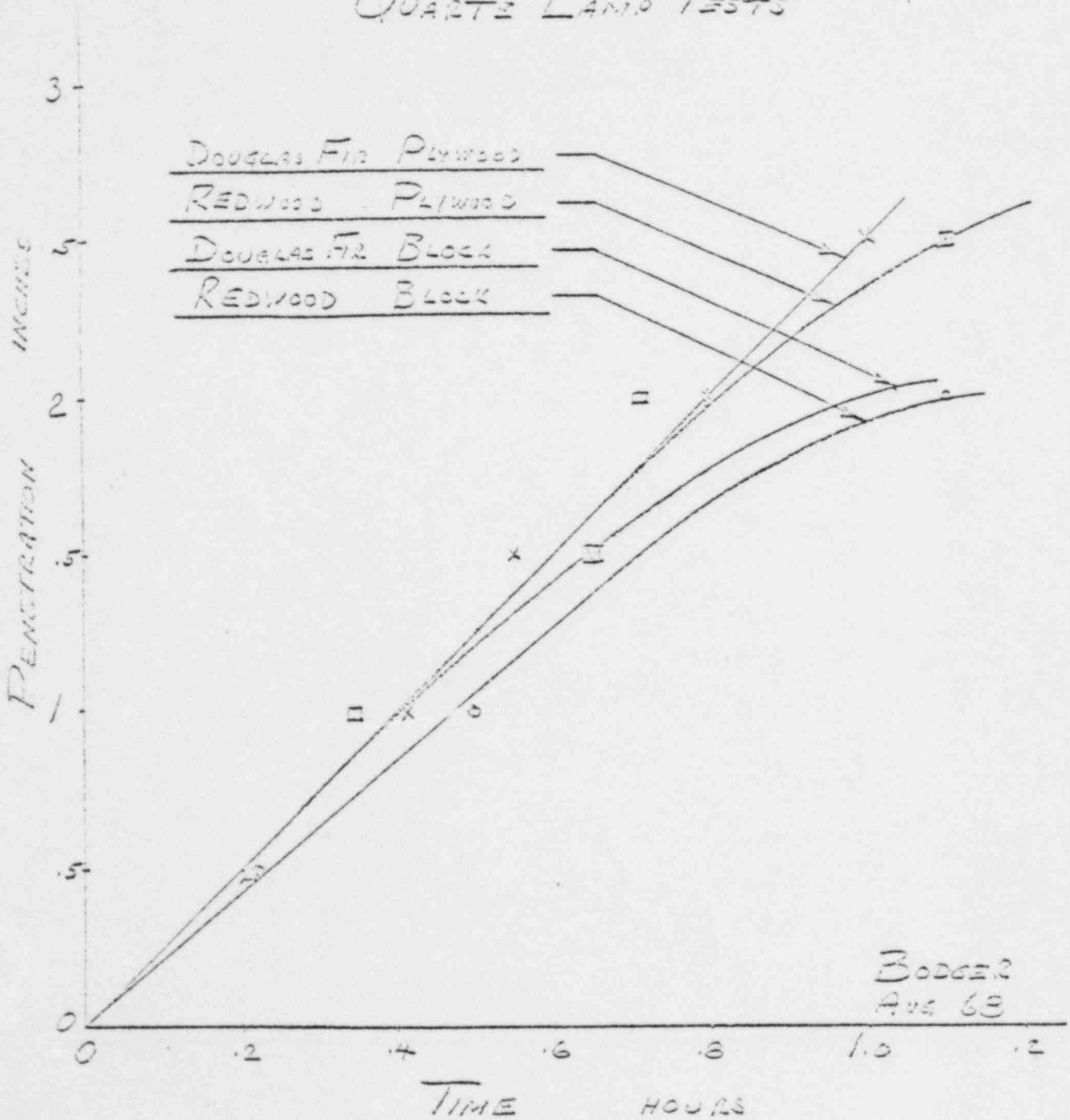
Considering the results of both the fire tests and the controlled tests, several conclusions can be drawn:

1. The container should be made of plywood, either redwood or Douglas fir.
2. The design should allow for burning away 3 inches of thickness.
3. A design thickness of 4 inches is sufficient for fire protection.

The Sandia tests, both the controlled tests and those in the fire pit, used "edge-on" samples. The results showed evidence of the fire progressing up the glue line more rapidly than thru the wood. In other words, the glue aggravated the fire situation. This condition can be avoided by using flat panels of plywood, instead of edge-on construction. Delamination while burning can be prevented, or at least the wood layers can be held in place, by a layer of steel sheathing attached to the outer surface of the container. This sheathing should be nailed on, but not bonded, to allow a vent path for the gaseous components escaping from the wood.

Fig 1

Penetration of 1300°F Isotherm
in
Quartz Lamp Tests



IV. CONSIDERATIONS OF IMPACT RESISTANCE

The Sandia test (Reference 1) used a container which was essentially a rigid, closed-end, circular cylinder. While this proved to be a satisfactory design, it is not a convenient shape for some applications.

An alternate design was chosen. This is an exoskeleton type -- a rectangular wooden box, with an exterior steel frame.

In a free fall, the container can land in any position -- on a flat surface, on an edge, or on a corner. The exoskeleton construction is durable in any of these types of impact. When landing on a flat surface, the cask simply presses the panel against the concrete (assuming the container lands on concrete). When landing on an edge, two of the flat panels are loaded as plates, while they are held in position by the exoskeleton. When landing on a corner, three panels are loaded as plates, and again they are held in position by the exoskeleton.

The exoskeleton is given rigidity -- i.e., is prevented from moving as a 4 bar linkage -- by having the plywood panels closely fitted to the panels of the skeleton. The six plywood panels are simply fitted to the skeleton, and are not bonded to each other. Thus, there are no bending moments at the edges of these panels.

Crush strips reduce the impact. The Sandia tests used crush strips on the outside of the containers. A similar strip on the inside of the container will serve the same purpose, and is used in the present designs.

V. CONSIDERATIONS OF PUNCTURE RESISTANCE

The experimental and analytical program conducted at the Oak Ridge National Laboratory has established a quantitative relationship to determine the thickness of steel sheathing required to protect a container against rupture in the puncture test. This information has served as a guide in designing steel sheathing.

Using low carbon steel sheathing of 63000 psi ultimate tensile strength and a total weight of 5000 pounds for container and contents, the results of Evans, et al, (Reference 8), indicate a required sheathing thickness of about 0.165 inch. Since the sheathing of the present design is backed up by 4 1/2 inches of

VII. CONTAINER DESIGN

The shipping container design consists mainly of 6 Douglas fir plywood panels fitted to a steel exoskeleton.

The exoskeleton consists of a frame of steel angle sections. End panels are integral welded units. The two end panels are held together by 4 longitudinal angle sections -- one along each of the 4 long edges. All joints are bolted at assembly, to simplify disassembly and reuse.

Douglas fir plywood panels are 4 1/2 inches thick -- i.e., each consists of 6 sheets of 3/4 inch plywood. These panels are bonded together with resorcinol-formaldehyde adhesive; they are not nailed, because nails would provide thermal short-circuits.

Each panel is sheathed with a 1/8 inch steel sheet. This sheathing (a) protects the wood in ordinary handling during normal use, (b) provides protection against penetration in the puncture test, (c) provides some protection against fire, and (d) prevents individual layers of the plywood from peeling off in the event of delamination, during a fire. It is nailed to the plywood all around the edge of the panel, but is not bonded. In addition, there are about 6 nails in the central region of each panel to keep the sheathing from bulging.

In a fire, the volatile components of the wood can escape through the space between the wood and the sheathing, since the sheathing is not bonded. The sheathing would bulge, due to the pressure of the escaping gases; the enclosed gas space would provide a thermal insulation to give some protection to the wood. The bulged shielding would pull out the 6 nails in the central region of the panel, providing a more direct path for the gases to escape.

The steel angle sections of the main frame extend over the loose joints of the plywood panels, preventing a fire from having a direct entry path to the interior. The steel angles also cover the nails around the edge of the panel, protecting them in normal service, keeping them from pulling out, and thermally insulating them (since they are short circuits) during a fire.

At assembly, the plywood panels are closely fitted to the exoskeleton. Thus the plywood provides shear rigidity to the exoskeleton, while the exoskeleton holds the 6 plywood panels together to form a container.

The transfer cask is bolted to the bottom panel of the container. The bolts are adequate for normal service, but will shear off in a severe impact. Crush strips, 2 x 2 redwood (not plywood), are glued and nailed on the inner surface of the plywood panels. They are provided on top, both sides, and both end panels. In a severe impact, the cask will come free of its mounts, and will impact one or several of the crush strips, lessening the shock of the cask.

In the event of impact followed by fire, the plywood panels will deform but still hold the container together. The fire performance of the plywood panels should be unchanged by impact.

In the event of fire followed by impact, the plywood panels may be seriously deteriorated by the fire. In order that the cask should not break through a panel at impact, the panels are reinforced by the heavy sheet steel sheathing. If the cask impacts any of these panels, the panel serves as a cushion to protect the cask, while the steel sheathing contains the cask. Even though the plywood may be seriously charred, the steel sheath maintains the integrity of the plywood panel so that the panel can function as a cushion. The crush strips will still be effective, of course.

Under normal conditions, the shipping container can be reused an indefinite number of times. The exoskeleton is bolted together, hence can be completely disassembled. When the exoskeleton is removed, the plywood container "falls apart," since the 6 panels are not joined together. All parts are painted to facilitate cleaning.

The design encompasses a family of nearly identical shipping containers to match a family of nearly identical transfer casks. There are 3 casks in the family, all identical except for different end caps:

Cap A, for Teletherapy Doubly
encapsulated up to 10 KC Cobalt 60
as in assembly drawing D0100-2

Cap B, for row activity
unencapsulated up to 10 KC
as in assembly drawing D0100-3

Cap A & C, for row activity

unencapsulated up to 15 KC

as in assembly drawing D0100-4

The shipping containers are identical, except for a dummy to fill the excess space with the smaller caps.

VIII. REFERENCES

1. J. A. Sisler, "New Developments in Accident Resistant Shipping Containers for Radioactive Materials," Sandia Corp.
2. W. H. McAdams, "Heat Transmission," McGraw-Hill, 2nd ed., 1942.
3. C. E. Crede, "Vibration and Shock Isolation," John Wiley, 1951.
4. "Wood Handbook," Forest Products Laboratory, U. S. Department of Agriculture, 1955.
5. R. J. Roark, "Formulas for Stress and Strain," McGraw-Hill, 3rd ed., 1954.
6. J. E. Shigley, "Engineering Design," McGraw-Hill, 1963.
7. Code of Federal Regulations, Part 71.
8. "Proceedings of the Second International Symposium on Packaging and Transportation of Radioactive Materials," UCC and USAEC, October 1968.

11-5-68

WKB

PROJECT NO 68.101

Test Requirements

As outlined in Ref 7, the Federal Code of Regulations, the tests are to cover

1. Free drop - 30 feet onto hard surface
2. Puncture - 40 inches onto spike
3. Thermal - 30 minutes at 1475°F
4. Water immersion - 24 hours at 3 feet.

sequentially, in the order indicated.

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PROJECT NO 68.101

Drop Test Considerations

With a large container, the drop test is likely to be ~~___~~ a more severe criterion than the fire ~~test~~; with a small container, the opposite would be true.

Of the 13 tests (or sets of tests) recorded on p 159 & 160 of Ref 1, 8 were concerned with 4000 lb samples. These 8 apply directly to our immediate problem; the others are of largely incidental interest.

Of the 13 tests,

a all were constructed of plywood,
b 11 were of douglas fir, 2 of redwood,
c 12 were (circular) cylinders, 1 was a cube.

Unfortunately, the cube was only 16 inches on a side, the gross weight was only 98 lb; the results thus are of limited usefulness.

Of the 8 tests on 4000 lb samples,

a all were (circular) cylinders,
b 7 were douglas fir, 1 was redwood.
c all 7 of the douglas fir samples were subjected to single drops; the redwood sample was dropped 3 times.
d 7 samples survived the test satisfactorily; 1 douglas fir sample suffered some delamination & Sisler does not state whether this was a pass or a fail.

W. K. BOOGER
SIERRA MADRE, CALIFORNIA

PROJECT NO.

e all were made with a 6 inch wall
f all had 2 inch crush rings; 7 had 5 rings, 1 douglas fir sample had only 3 rings.
g 2 douglas fir samples were subjected to both fire ~~test~~ & drop ~~test~~. One was fire ~~tested~~ first, then drop ~~tested~~; the other the opposite. Each survived the double test. (One smaller sample also was subjected to & survived a double ~~test~~).
h among the douglas fir samples
1 was dropped on end, with minor damage
1 " " " its side, " "
all others were dropped at 45°
i the crush rings appear to have served the purpose of shock reduction quite satisfactorily.

The general conclusion from examining the Sandia ~~test~~ results appears to be
a a (circular) cylinder with a 6 inch wall plus 2 inch crush rings, made of plywood — either douglas fir or redwood — glued & nailed is satisfactory for a 4000 lb gross weight.
b we must investigate further to establish the proper basis for design of an essentially cubical container for 4000 lbs gross weight.

Mechanically, a circular cylinder is a nearly ideal shape. A rectangular box is far less durable, structurally, although integral ends help a lot.

To get some idea of the behavior of a box under impact loading, we should derive some scaling laws. Crede, Ref 3, gives a useful description of the situation.

We shall think of the box as a weightless spring. The shipping cask we shall consider to be a rigid (ie non-yielding) mass. The box & cask together comprise a spring & mass system. When this system is dropped onto a rigid platform (the concrete surface), we assume an ideally plastic impact — ie there is no bounce.

In this situation, Crede gives the compression of the spring as

$$\delta = \frac{V}{\omega_0}$$

where

δ = compression of spring

V = velocity of spring/mass system at impact.

ω_0 = natural frequency of spring/mass system.

To consider a "worst case" type analysis, assume that the mass is attached to the box directly opposite the point of impact.

If the loading is a direct impact on a corner, as in Fig 1, the deformation will be largely a direct compression of the several panels. The deformation under load will be

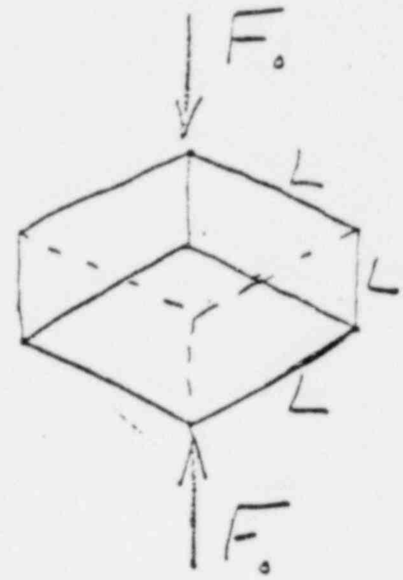

Fig D1

$$\delta \sim \frac{F_0}{bh}\frac{1}{E}L$$

$$\sim \frac{F_0 L}{Ehb} \sim \frac{F_0}{Eh}$$

where

F_0 = total applied force
E = modulus of box material
L = length of a side of the box
b = width of " " " "
h = thickness " " " " "

The spring constant then is

$$k = \frac{F_0}{\delta} \sim Eh$$

At impact, the force generated is

$$F = k\delta \sim k\frac{V}{\omega_0} \sim k\frac{V}{\sqrt{k/m}} \sim V\sqrt{km}$$

$$\sim V\sqrt{\frac{Wk}{g}}$$

where m = mass of cask
W = weight " "

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The direct stress in the panels, due to F, is

$$\sigma \sim \frac{F}{bh} \sim \frac{V}{bh}\sqrt{\frac{Wk}{g}} \sim \frac{V}{bh}\sqrt{\frac{WEh}{g}}$$

$$\sim \sqrt{\frac{V^2 E W}{g b^2 h}}$$

If the spring/mass system has fallen through a distance H, the impact velocity will be

$$V = \sqrt{2gH}$$

so

$$\sigma \sim \sqrt{\frac{2gHEW}{g b^2 h}} \sim \sqrt{\frac{HEW}{h L^2}}$$

This relation does not tell us what the stresses will be in an impact. It does tell us the relative change in stress for any change in H, E, W, b, or h.

This relation also does not tell us what the stresses will be in an impact. But, like the relation on p (5), it tells us the relative change in stress for any change in H, E, W, b, σ or h.

Since the two scaling relations — one for direct stress, the other for bending — came out to the same form, we conclude that this form has general usefulness since in the actual case both bending and direct stress will occur.

Rearranging the σ relation of p (5) & (6) and using the limiting (i.e. failure) values of σ, the minimum design value of h_0, for given values of H, W, & L, is

$$h_{0\,min} \sim \frac{E}{\sigma^2_{failure}}$$

From Table 12 of the Wood Handbook (Ref 4)

	$\frac{E}{10^6}$	Static $\frac{\sigma_{prop}}{10^3}$	Static $\frac{\sigma_{rupt}}{10^3}$	$\frac{E}{\sigma^2_{prop}}$	$\frac{E}{\sigma^2_{rupt}}$
Douglas-fir					
Coast	1.95	7.80	12.2	0.0320	0.0131
Intermediate	1.64	7.40	11.2	0.0300	0.0131
Rocky mountain	1.40	6.30	9.60	0.0352	0.0152
Redwood	1.34	6.90	10.0	0.0282	0.0134

Since $h_{0\,min}$ is proportional to either of the last 2 columns, there is little to choose among these woods. Failure is the real criterion, so the last column is the most significant.

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The stress relation can also be used to compare large and small boxes. Sisler's Sandia tests (Ref 1) include one cube. The characteristic dimension is given as "16 inches"; we assume this to be the exterior edge of the cube, and use it as "L" in the σ relation.

The Sandia cubical box was dropped 3 times — probably one drop each on side, edge, & corner. The box was destroyed "on the third drop" — probably on the corner. If we were to design a box having the same stress (ie the same likelihood of failure) as the Sandia cube, we should say

$$\sigma_s \sim \sqrt{\frac{H_s E_s W_s}{h_s L_s^2}}$$

$$\sigma_L \sim \sqrt{\frac{H_L E_L W_L}{h_L L_L^2}}$$

where the subscripts denote

s = small sample

L = large sample

Then

$$\frac{\sigma_L}{\sigma_s} = 1 = \sqrt{\frac{H_L E_L W_L h_s L_s^2}{H_s E_s W_s h_L L_L^2}}$$

PROJECT NO.

The thickness of the large box should then be

$$h_L = h_S \frac{H_L E_L W_L L_S^2}{H_S E_S W_S L_L^2}$$

Since the Sandia box was made of douglas fir plywood and we are considering the same material, we have $E_L = E_S$. Thus

$$h_L = h_S \frac{H_L W_L L_S^2}{H_S W_S L_L^2}$$

Using data from the "Drop Test Results" table on p 159 of Ref 1, this gives

$$h_L = 3\,in \frac{30 \times 4000 \times (16)^2}{16 \times 98 \times (45)^2} = 9.68 \times 3\,in$$

$$= 29.0\,in \qquad 230 -$$

where $L_L = 45$ in has been estimated from US Nuclear Corp dwg no. D-0053, and assuming $h_L = 6$ in. In view of this result, it might be more realistic to use L as the centerline length of each member (which it obviously should be) so that $L_S = 16-3 = 13$ in, and to try several values of h_L, using $L_L = 33\,in + h_L$.

Assumed h_L	6	8	10	12	14	16	18	20
" L_L	39	41	43	45	47	49	51	53
" $(L_S/L_L)^2$	.111	.1005	.0914	.0835	.0765	.0704	.0649	.0601
Derived h_L	25.5	23.0	21.0	19.2	17.5	16.1	14.9	13.8

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This result indicates that a cubical box, having a 33 inside dimension and a 16 inch wall, built with the same type of construction as the Sandia box, will exhibit the same performance with a 4000 lb load in a 30 ft drop as the Sandia cubical box exhibited in its ~~test~~. Since the Sandia box failed on the third drop, it was marginally satisfactory at best. Insufficient tests were reported to determine whether failure was due to (a) repeated impacting which ultimately caused failure, or (b) a more severe ~~test~~ which happened to be the third drop.

The conclusion is that a large box, built with the same type of construction as the Sandia box, is not a promising approach.

The detailed construction of the Sandia cubical box is not given in the report. The table on p 159 gives:

Construction Material - "Douglas Fir Plywood Box"
Bonding - "Nailed and glued"

There is no photograph of this specimen, and no specific indication in the text that it did or did not have longitudinal bolts. In the case of the larger specimens, the ~~text~~ mentions a set of longitudinal bolts, and they are shown in several photographs.

The conclusion on p 10 seems unreasonably pessimistic. Perhaps this is because the cubical box was not properly bolted. While the circular cylinder is a better structural shape than the cubical box, the box nevertheless is a pretty good shape - especially when all six panels are present & properly joined. Therefore we shall try the box from a different approach.

Think of the box as a square cylinder with open ends & compare it with a round cylinder with open ends. The basic information on these cylinders can be obtained from Roark, Ref 5.

For the circular cylinder, Fig 3 (a), Roark gives on p 156

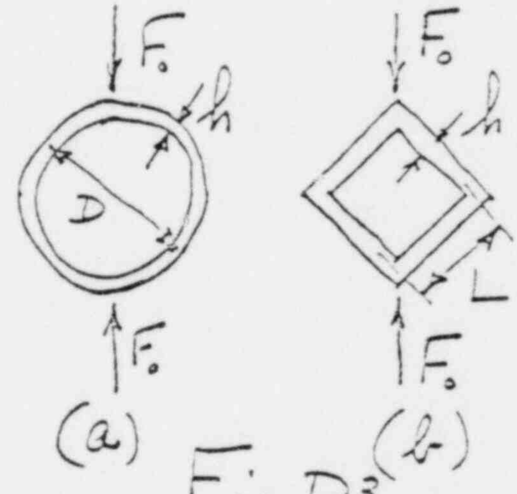


Fig D3

$$\delta = 0.149 \frac{F_0 \left(\frac{D}{2}\right)^3}{EI}$$

$$M_{max} = 0.318 \, F_0 \left(\frac{D}{2}\right)$$

The spring constant then will be

$$k = \frac{F_0}{\delta} = \frac{8}{0.149} \frac{EI}{D^3} = 53.6 \frac{EI}{D^3}$$

To fit the square cask into the round cylinder

$$D = \sqrt{2}\, L$$

So

$$k = 53.6 \frac{EI}{(\sqrt{2})^3 L^3} = 19.0 \frac{EI}{L^3}$$

PROJECT NO.

As on p (3) & (4), we can say that at impact

$$\delta = \frac{V}{\omega_0} = V\sqrt{\frac{m}{k}} = V\sqrt{\frac{W}{gk}}$$

$$F = k\delta = V\sqrt{\frac{kW}{g}} = \sqrt{\frac{V^2 kW}{g}} = \sqrt{\frac{2gHkW}{g}}$$

$$= \sqrt{2HkW}$$

the bending moment will then be

$$M = 0.318\,\frac{L}{\sqrt{2}}\sqrt{2HkW} = 0.318\sqrt{L^2HkW}$$

$$= 0.318\sqrt{L^2HW\,19.0\,\frac{EI}{L^3}} = 1.38\sqrt{\frac{HWEI}{L}}$$

and the bending stress

$$\sigma = \frac{Mc}{I} = \frac{Mh/2}{I} = \frac{h}{2I}\,1.38\sqrt{\frac{HWEI}{L}}$$

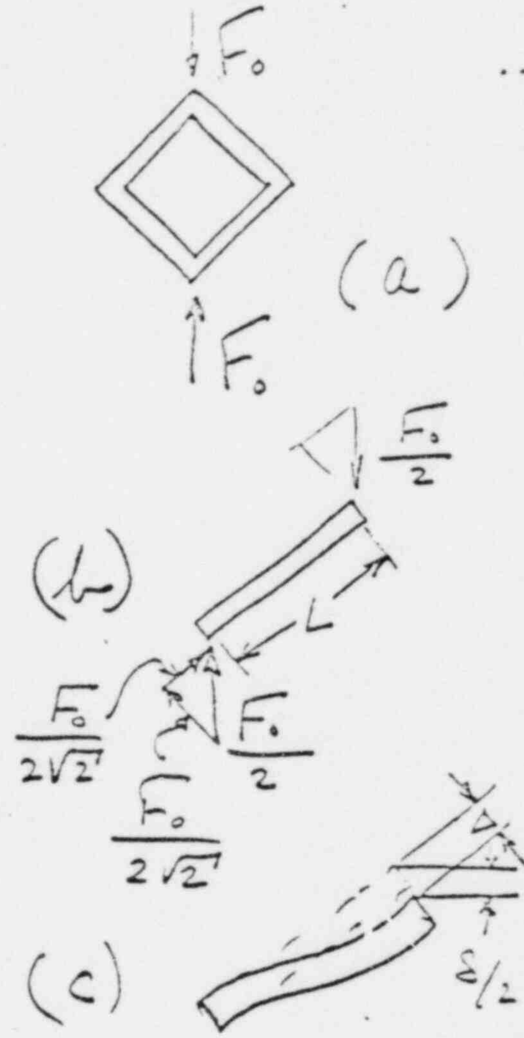
$$= 0.690\sqrt{\frac{HWEh^2}{LI}} = 0.690\sqrt{\frac{HWEh^2}{Lbh^3/12}}$$

$$= 2.40\sqrt{\frac{HWE}{Lbh}}$$

this is a definite equation (not just a proportionality) for the bending stress at impact for a circular cylinder with open ends.

PROJECT NO.

For the square cylinder with open ends, we have to synthesize a solution. Taking the sketch of Fig 3(b), we take out a member as in Fig 4. From symmetry, the loads are as in Fig 4(b), the ends do not rotate, and the deformations are as in Fig 4(c). Roark does not tabulate the case of Fig 4 (b) & (c) but it consists of two cantilevers with end loads, so (Roark p100)



$$\Delta = 2\,\frac{F_o}{2\sqrt{2}}\,\frac{(L/2)^3}{3EI}$$

$$= \frac{F_o L^3}{24\sqrt{2}\,EI}$$

$$M_{max} = \frac{F_o}{2\sqrt{2}}\,\frac{L}{2}$$

$$= \frac{F_o L}{4\sqrt{2}}$$

The deflection, Δ, is normal to the $\mathbb{C}$ of this member. So

$$\frac{\delta}{2} = \frac{1}{\sqrt{2}}\,\Delta$$

and

$$\delta = \sqrt{2}\,\Delta = \frac{F_o L^3}{24EI}$$

PROJECT NO.

The spring constant is

$$k = \frac{F_0}{\delta} = \frac{24EI}{L^3}$$

Again, as on p (12), we can say the force at impact is

$$F = \sqrt{2HkW}$$

and the moment is

$$M = \frac{L}{4\sqrt{2}}F = \frac{L}{4\sqrt{2}}\sqrt{2HkW} = \frac{1}{4}\sqrt{L^2HkW}$$

$$= \frac{1}{4}\sqrt{L^2HW\frac{24EI}{L^3}} = \sqrt{\frac{3}{2}}\sqrt{\frac{HWEI}{L}}$$

and the bending stress

$$\sigma = \frac{Mc}{I} = \frac{Mh/2}{I} = \frac{h}{2I}\sqrt{\frac{3}{2}}\sqrt{\frac{HWEI}{L}}$$

$$= \sqrt{\frac{3}{8}}\sqrt{\frac{HWEh^2}{LI}} = \sqrt{\frac{3}{8}}\sqrt{\frac{HWEh^2}{L\,bh^3/12}}$$

$$= \sqrt{\frac{9}{2}}\sqrt{\frac{HWE}{Lbh}} = 2.12\sqrt{\frac{HWE}{Lbh}}$$

This is a definite equation (not just a proportionality) for the bending stress at impact - neglecting stress concentration - for a square cylinder with open ends.

PROJECT NO.

Comparing the σ equations for the two cases – p (12) & (13) – it appears that σ is about 11% smaller for the square cylinder than for the round cylinder. This makes sense, and we must conclude that the cubical box in the Sandia tests was not held together as well as the round cylinders were.

The results of p (12) & (14) were for cylinders with open ends, whereas we must have closed ends. It is reasonable to assume that the comparison with closed ends would be about the same as with open ends; we shall make this assumption.

The conclusion of this analysis is that we can use, for a square container, the same wall thickness as Sandia used in their round containers. We must do a very careful job of holding the several parts together, however. We should also provide, in the design, for keeping the stress concentration at the corners reasonably small.

The cask must be attached to the base with a sufficiently strong attachment that it will withstand the acceleration load. Crede gives this acceleration as (Ref 3, p 92)

$$a = V\omega_0 = V\sqrt{\frac{k}{m}} = \sqrt{\frac{V^2 k}{m}} = \sqrt{\frac{2gHkg}{W}}$$

$$= g\sqrt{\frac{2Hk}{W}} = g\sqrt{2\frac{H}{\delta_{stat}}}$$

Using k from p (14) as a fair approximation (the ends will increase k beyond the p (14) value and crushing of the crush rings will decrease it)

$$\frac{a}{g} = \sqrt{\frac{2Hk}{W}} = \sqrt{\frac{2H}{W}\frac{24EI}{L^3}}$$

$$= \sqrt{\frac{48\,HEI}{WL^3}}$$

The force (i.e. the equivalent static force) to be resisted) is

$$F_{dyn} = W\frac{a}{g} = W\sqrt{\frac{48HEI}{WL^3}} = \sqrt{\frac{48\,HWEI}{L^3}}$$

$$= \sqrt{\frac{48\,HWEbh^3}{12L^3}} = 2\sqrt{\frac{HWEbh^3}{L^3}}$$

$$= 2\sqrt{\frac{30\,\text{ft}\; 4000\,\text{lb}\; 1.95\cdot 10^6\,\frac{\text{lb}}{\text{in}^2}\, 33\,\text{in}\,(6\,\text{in})^3}{(32\,\text{in})^3}\frac{12\,\text{in}}{\text{ft}}}$$

$$= 1.56\cdot 10^6\,\text{lb}$$

PROJECT NO.

If this load is to be carried by 12 bolts, the load per bolt will be

$$F_{bolt} = \frac{1.56 \cdot 10^6\ lb}{12\ bolt} = 130\,000\ \frac{lb}{bolt}$$

On the basis of an ultimate shear load of

$$shear = 0.6\ tensile = 0.6 \times 64000\ psi$$

$$= 38400\ psi$$

the bolt size would be

$$A = \frac{F_{sh}}{shear_{ult}} = \frac{130\,000\ lb/bolt}{38400\ lb/in^2}$$

$$= 3.38\ in^2/bolt$$

or

$$d = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \times 3.38\ in^2}{\pi}} = 2.07\ in$$

It would be virtually impossible to get a good connection of the mount to the wood base with this size bolt. The wood would crush to some extent anyway, making a/g somewhat smaller but still requiring about $d = 1.5$ in to hold the cask firm.

The solution is to use bolts of some reasonable size, and let the cask shear them off. The cask will still be contained and the performance of the container will not be influenced adversely.

All of the foregoing has been based on an all-wood container — i.e. the wood must provide the structural strength, both before and after a fire. Alternatively, the structural load can be carried by a separate (steel) frame so that the principal function of the wood is as a fire protection.

One possible construction, not sketched here, is a rectangular steel frame — an exoskeleton — with 6 rectangular plywood panels. The panels are fitted to the frame, and thus provide shear rigidity — i.e. they keep the frame panels from acting like 4-bar linkages.

The wood panels are 4 inches thick, or more, and are sheathed on the outside with a layer of 1/16 inch steel sheet. This steel sheet weighs about 2.5 lb/ft^2, and thus is equivalent (in weight) to about 1.0 inch of plywood. Douglas fir weighs from 2.5 to 2.9 lb/board foot.

Crush strips will be 2×2 redwood — not plywood — mounted on the inside of the container.

11-5-62
S WKB

PROJECT NO 62.101

Spike Test Considerations

The spike ~~test~~ consists of a free fall of 40 inches, landing on a spike 6 inch diameter × 8 inch long. This may precede or follow a fire. The end of the spike is square - i.e. it is blunt, with $\frac{1}{4}$"R edge.

~~This test~~ is of virtually no importance with a very small container. This is true because the flat surface of the spike is just another flat, unyielding surface. The only risk comes when the container lands on the edge of the spike. Since the container must also withstand a 30 ft fall onto a hard flat surface, the spike ~~test~~ does not control.

With a large container, the situation is different. The spike may penetrate a relatively weak portion of the container, since the 6 inch diameter is small compared with the container dimensions.

The proposed exo-skeleton design is planned primarily for protection in the 30 ft fall, and fire. It is vulnerable to the spike ~~test~~ and must be modified.

The container will be dropped so as to impact a flat surface on the spike - ie it will not be skewed.

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We may take a basic premise that a clear area of plywood panelling large enough for the 6 inch spike to impact without obstruction will be a vulnerable location. We can also say that such an area will not be penetrated if it is adequately supported from the rear. We can also say that shielding of the panel by a portion of the exo-skeleton will prevent penetration. Since testing will be done "flat-to-flat", the possibility of landing on the edge of the spike is not a major consideration.

First look at the question of impacting on a panel. With a free fall of 40 inches, the impact velocity is about

$$V \cong 200 \text{ in/sec}$$

(See Crede, Ref 3, p 93). The entire assembly will be subjected to an acceleration of

$$a = V\omega_0$$

(Crede p 92). The natural frequency ω_0 comes from static deflection. If the entire assembly might deflect a panel about 0.1 in, then the natural frequency would be (Crede, p 39)

$$\omega_0 = 10 \frac{\sim}{\text{sec}} \cong 60 \frac{\text{rad}}{\text{sec}}$$

and the acceleration

$$a = V\omega_0 = 200 \frac{\text{in}}{\text{sec}} 60 \frac{\text{rad}}{\text{sec}} = 12000 \frac{\text{in}}{\text{sec}^2} \times \frac{g \text{ sec}^2}{386 \text{ in}} \cong 30 \text{ g}$$

If the assembly weighs 5000 lb, the force required to stop it would be

$$F = W \times \frac{a}{g} = 5000 \text{ lb} \times 30 = 150\,000 \text{ lb}$$

A 6 inch spike will have a circumference of about 20 inches. So the line force will be

$$\frac{\text{force}}{\text{inch}} = \frac{F}{C} = \frac{150\,000 \text{ lb}}{20 \text{ in}} = 7500 \text{ lb/in}$$

The average shear stress in a 4½ inch thick panel will be

$$\tau = \frac{F}{Ch} = \frac{150\,000 \text{ lb}}{20 \text{ in}\ 4.5 \text{ in}} = 1515 \text{ psi}$$

The exact data are not readily available. Wood Handbook (Ref 4, p 75) however gives data on shear parallel to grain. A reasonable estimate is that we can use this value for "punch-out" shear strength.

$$S_{su} \cong 1000 \text{ psi}$$

This indicates that we are deficient in strength by a factor of about 1½. Considering the approximations made in arriving at τ above, we have to conclude that the spike would penetrate the container.

A portion of the load will be carried by the $\frac{1}{16}$ inch steel sheet facing. If it has an ultimate shear strength of about

$$S_{su} \cong 40\,000 \text{ psi}$$

it will carry about

$$F_{steel} \cong 20 \text{ in} \times \tfrac{1}{16} \text{ in} \times 40\,000 \text{ psi}$$

$$= 50\,000 \text{ psi}$$

this is about $\frac{1}{3}$ of the total load. This is a big help, but the whole thing is still marginal & we still conclude that the spike will penetrate the container.

If we consider the panel to be backed up, the load will tend to crush the panel. Distributing the load over the entire spike area, the crushing stress would be

$$\sigma_{crush} = \frac{F}{A} \cong \frac{150\,000 \text{ lb}}{28 \text{ in}^2} \cong 5400 \text{ psi}$$

Actually it is not reasonable to assume the same value of F for both of these cases. Since we have not worked out W_0 for either case, however, this guess is about as good as any. The crushing strength is given in Ref 4 as about

$$S_{uc} \cong 7000 \text{ psi}$$

Direct loading – ie crushing – seems OK.

PROJECT NO.

Probably the best solution to the "punch-out" problem is to cover vulnerable areas with parts of the exoskeleton.

End Panel

Nominal dimensions are as sketched in Fig 1(a)

A possible bracing arrangement is as in Fig 1(b). The inner diagonal is

$$\text{Diag} = \sqrt{(27\tfrac{1}{8})^2 + (21\tfrac{1}{8})^2}$$

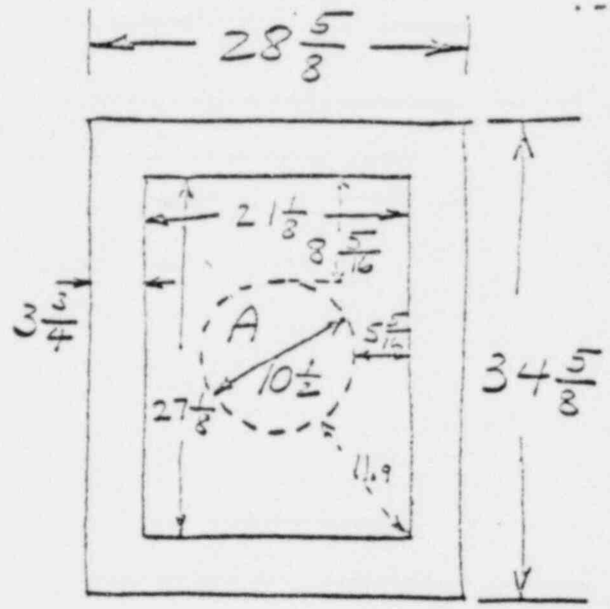
$$= 27\tfrac{1}{8}\sqrt{1+\left(\frac{21\frac{1}{8}}{27\frac{1}{8}}\right)^2}$$

$$= 34.3 \text{ in}$$

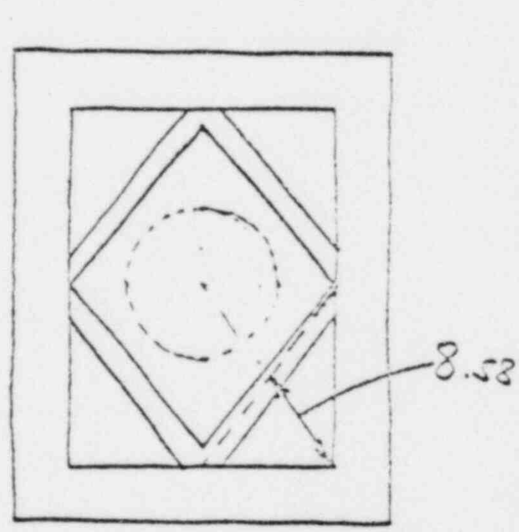
The $\frac{1}{4}$ diagonal is

$$\frac{\text{Diag}}{4} = 8.58 \text{ in}$$

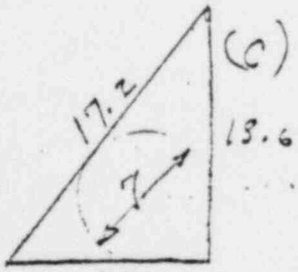
The largest circle inscribed in this triangle is about 7 inch diameter (scaled). For a 5 inch circle the intercepts are at $9\frac{1}{2}$ & $7\frac{1}{2}$ in.



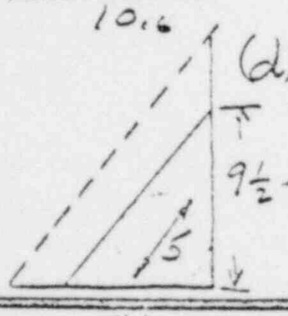
(a)



(b)



(c)



(d)

Fig S1

PROJECT NO.

A 6 inch circle won't quite go into the center space, but it comes too close. To make a 5 inch circle, the diagonal can be as sketched.

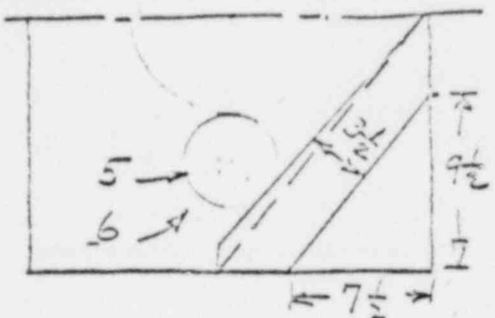


Fig S 2

The diagonal brace should be $3\frac{1}{2}$ or 4 inch X $\frac{1}{8}$ inch strap.

Four diagonals as sketched, welded all around, will protect the end panels adequately.

PROJECT NO.

Bottom Panel

Bottom panel has two 4×6 skids. Each end & each side covered by exoskeleton with $3\frac{5}{8}$ overlap. Center portion ($6\frac{3}{4}$ in at midwidth) backed up by cask.

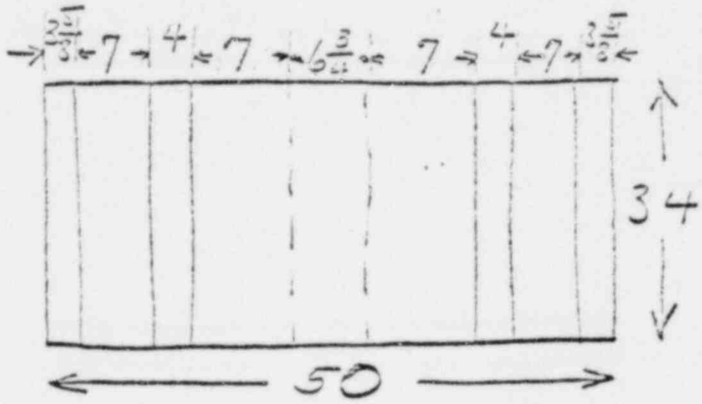


Fig S 3

With this set of protectors, best arrangement is to divide remaining length into 4 equal parts, of 7 inches each. Even this optimum is about 2 inches too much per panel. There is no really practical way to add braces, etc, for protection.

Increase the $\frac{1}{16}$ inch steel sheet covering to $\frac{1}{8}$ inch steel sheet covering. This will carry about $2/3$ of the spike load and solve the penetration problem. In addition, it will perform its other functions as well as, or better than, the thinner sheet.

Top Panel & Side Panels

These panels are similar to the bottom panel, except that they do not have skids – i e they are more hopeless. Increase the sheet to $\frac{1}{8}$ inch on them also.

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Evans, Nelms, & Stoddart (Ref 8, p 253) provide a set of curves based on their empirical equation

$$h = \left(\frac{W}{S_u}\right)^{0.71}$$

For a weight of $W = 5000$ lb and a steel strength of $S_u = 63000$ psi this gives $h = 0.165$ in. This is the thickness for marginal resistance to penetration. It agrees completely with the result (p 4) that a $\frac{1}{16}$ in thick steel sheath will carry about $\frac{1}{3}$ of the load.

This result is interpreted as confirming that a $4\frac{1}{2}$ inch plywood panel sheathed with $\frac{1}{16}$ inch steel sheet would be marginal, but that the same plywood panel sheathed with a $\frac{1}{8}$ inch steel sheet will resist penetration.

Fire Test Considerations

The Sandia fire tests, Ref 1, were realistic. We assume that any new design of shipping container will be subjected to substantially the same conditions.

Sisler states (Ref 1, p 143) that the fire test is equivalent to an 1850°F black body, and recommends this for design purposes. We shall use this figure as our basis.

In the detailed studies of charring and temperature penetration, Sisler further states that a quartz lamp was programmed to provide an 1850°F black-body source. He also states (p 151) that they actually measured $q = 11$ Btu/ft²sec in this setup. This can be checked:

For 1850°F black body (See McAdams, ref 2, p 49)

$$q = \sigma T^4$$

$$= 0.173 \cdot 10^{-8} \frac{Btu}{ft^2\, hr\, (°R)^4} (1850+450)^4 (°R)^4 \frac{hr}{3600\ sec}$$

$$= 13.1 \frac{Btu}{ft^2\, sec}$$

For $11 \frac{Btu}{ft^2\, sec}$

$$T = \sqrt[4]{\frac{q}{\sigma}} = \sqrt[4]{\frac{11 \frac{Btu}{ft^2\, sec}}{0.173 \cdot 10^{-8} \frac{Btu}{ft^2\, hr\, (°R)^4}} \left(\frac{3600\ sec}{hr}\right)} = 2190\,°R = 1730\,°F$$

F ②

PROJECT NO.

The two figures do not agree. Results of the detailed studies should be modified to allow for

$$\frac{13.7 \frac{Btu}{ft^2 sec}}{11 \frac{Btu}{ft^2 sec}} = 1.244$$

a 25% increase in heat input.

Lacking any more specific information, we shall assume that the 25% greater heat input rate will result in a 25% greater penetration in the stated one hour test period.

After any given portion of the wood sample has fully charred, it reaches a more or less steady temperature of about 1800°F. We can thus take 1800°F as an indication of complete destruction. The data from the quartz lamp test, which Sisler presents as his curves C-1 through C-4 (pages 171-182 of Ref 1) have been replotted have been replotted here as Fig #1 to show the rate of advance of total destruction - ie the progression of the 1800°F isotherm.

Sisler's data appear to indicate

a Redwood is slightly better than fir, in either block or plywood form.

b Block is slightly better than plywood; specifically, plywood requires about 1/2 inch more thickness than block.

These conclusions are based only on the quartz lamp test. These conclusions are not in agreement with those of Sisler (Ref 1, p 151).

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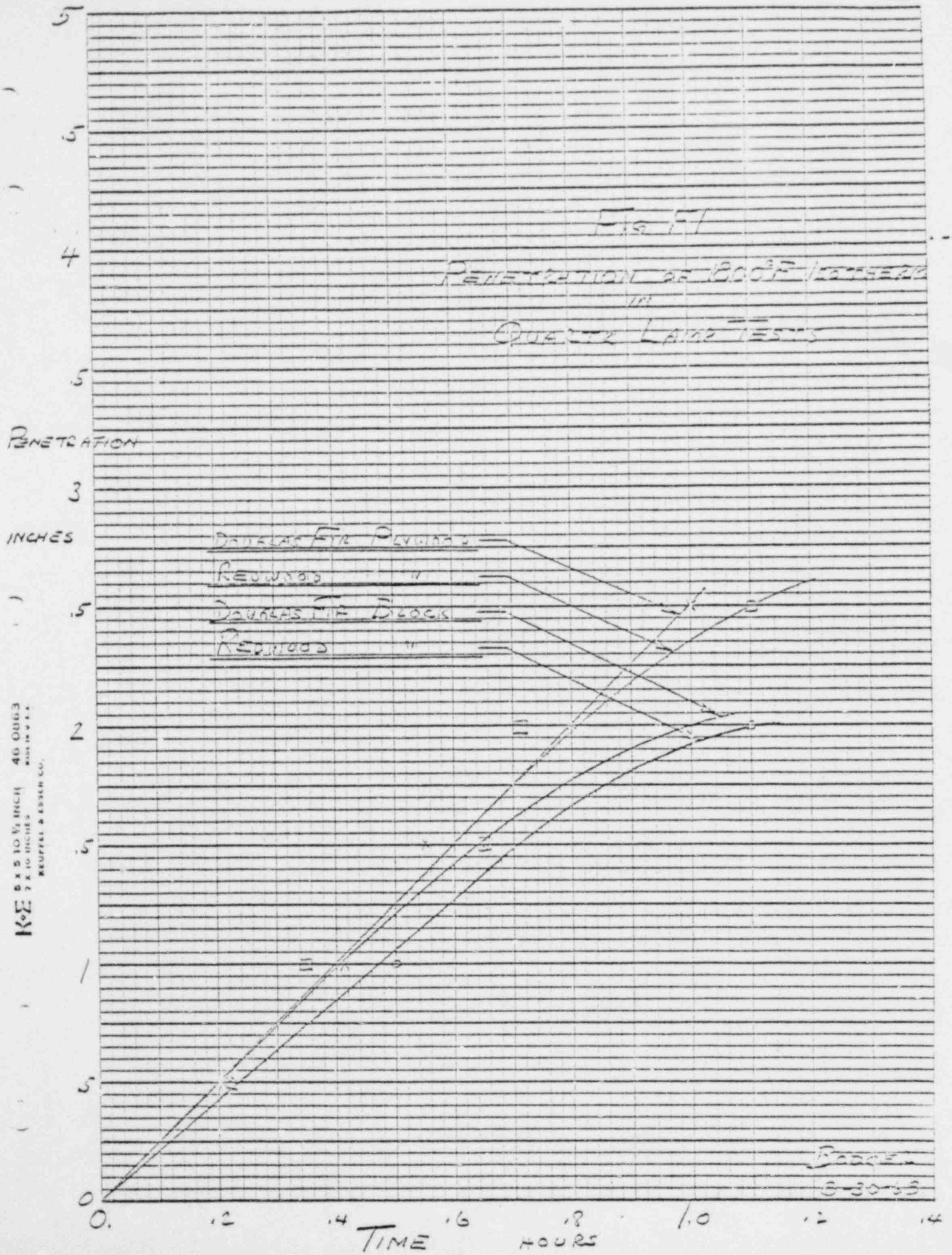


Fig F1
Penetration of Isotherm
in
Quartz Lamp Tests
Penetration
inches
Douglas Fir Plywood
Redwood "
Douglas Fir Block
Redwood "
Time
hours

PROJECT NO.

The quartz lamp data indicate that we can expect to lose, in a 1 hour fire,

	11 $\frac{Btu}{ft^2\text{-}sec}$	13.7 $\frac{Btu}{ft^2\text{-}sec}$
Douglas fir plywood	$2\frac{1}{2}$ in	$3\frac{1}{8}$ in
Redwood "	$2\frac{3}{8}$	3
Douglas fir block	2	$2\frac{1}{2}$
Redwood "	2	$2\frac{1}{2}$

Examination of the results of the five fire tests (Ref 1, pp 163 – 168) indicates

a Solid fir, fir plywood, redwood plywood all burn away about 2 to $2\frac{1}{2}$, and sometimes 3 inches.

b Solid redwood burns more than the other three.

c Plywood is preferable since solid fir and solid redwood are both subject to splitting.

d Maximum interior temperatures are well below melting temp of lead (ie below 621°F). See Fig F 2.

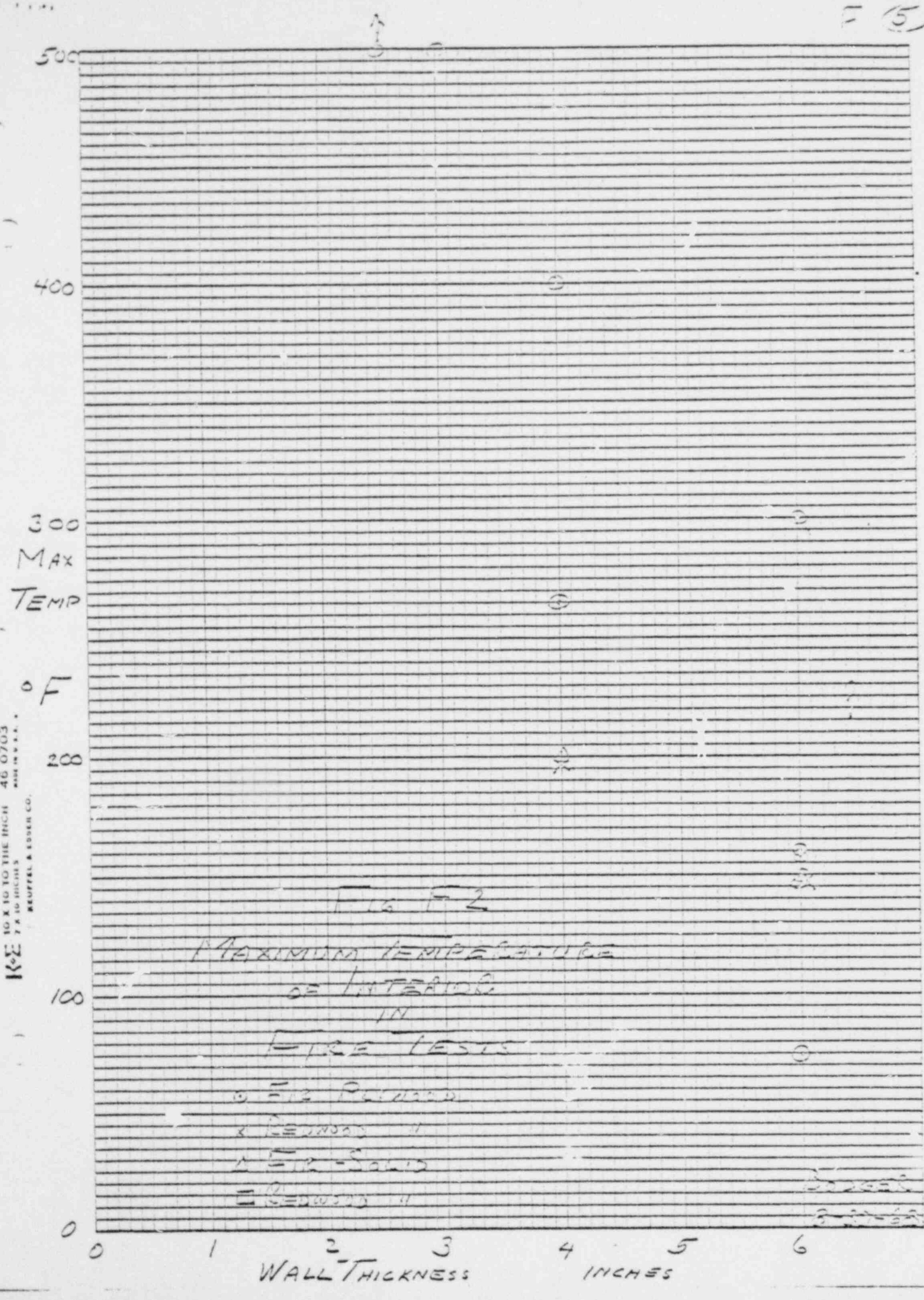
Conclusions:

Based on all of the foregoing, the container

a Should be made of plywood – either fir or redwood.

b The design should allow for burning away 3 inches of thickness.

c A design thickness of 4 inches is satisfactory for fire protection, since interior temperature does not exceed 400°F.


500
400
300
Max
Temp
°F
200
100
0
0
1
2
3
4
5
6
Wall Thickness Inches
Fig F2
Maximum Temperature
of Interior
in
Fire Tests
○ Fir Plywood
x Redwood "
△ Fir-Solid
□ Redwood "

10-10

PROJECT NO.

One can also ask, quite logically, — How much will the fire resistance be influenced by the addition of a thin steel outside the wooden container?

Sisler gives a little information, but does not state it explicitly. Items 1, 2, & 3 of Fire Test I tabulation on p 163 all had steel outer shells. Items 1 & 2 did not use wood for insulation, hence are not relevant here. Item 3 had a 3 inch layer of Douglas Fir plywood insulation.

During the fire test, 2 inches of this wood burned away. The container was considered to have passed the test. This 2 inches of burning is the same as the least amount of burning of the unshielded containers. It thus looks as if the steel sheathing had a negligible effect on fire resistance.

The conclusions on p ④ still apply, with no change.

Lifting Eyes

Lifting eyes should be provided. There will be 4 eyes, located at the 4 corners of the exoskeleton. Each eye will be a piece of steel strap, welded to a vertical member of the end frame, as in Fig L 1

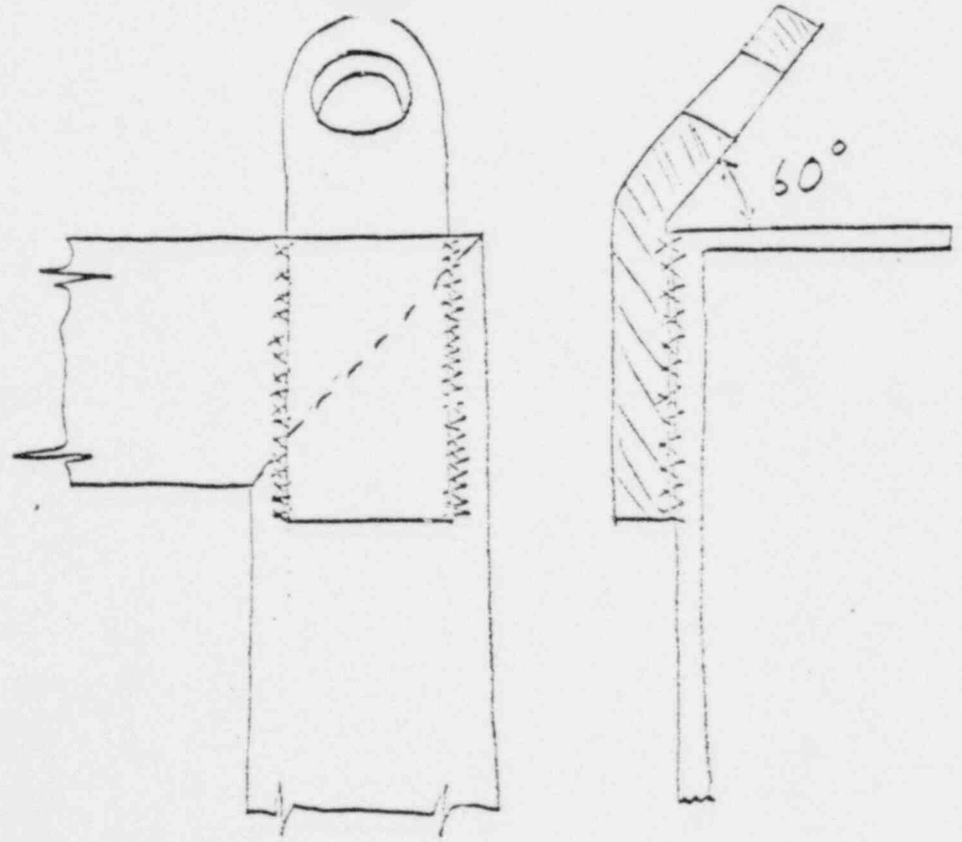


Fig L 1

Material will be low carbon steel strip. Approximate properties

Tensile strength = S_u = 63 000 psi
Yield " = S_y = 46 000
Elongation = El = 38%
Reduction of area = RA = 62%

The DOT regulation requires that the total weight shall be lifted without stressing the lifting eyes to more than $\frac{1}{3}$ of S_y. It is good practice to dimension the eyes so that this requirement will be met while using only two diagonally opposite eyes.

The assumed ~~total~~ load of cask and shipping container is 5000 lb, so we design on the basis of 2500 lb per eye.

From the ~~statics~~ of the situation we get the load acting on the eye to be

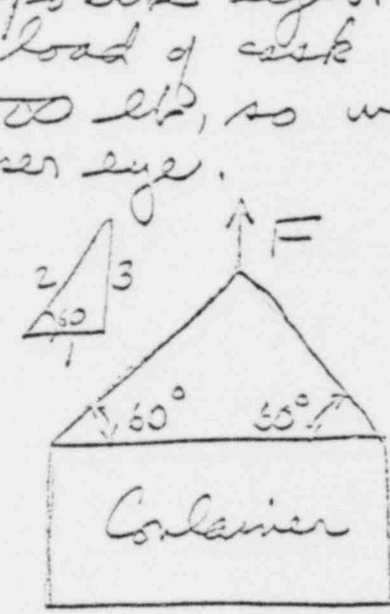


Fig L2

$$P = \frac{F}{2} \times \frac{2}{\sqrt{3}}$$

$$= \frac{F}{\sqrt{3}} = \frac{5000 \text{ lb}}{\sqrt{3}}$$

$$= 2900 \text{ lb}$$

We assume this to be a straight pull on the eye.

PROJECT NO.

Neglecting any stress concentration, the required tensile area in the eye is

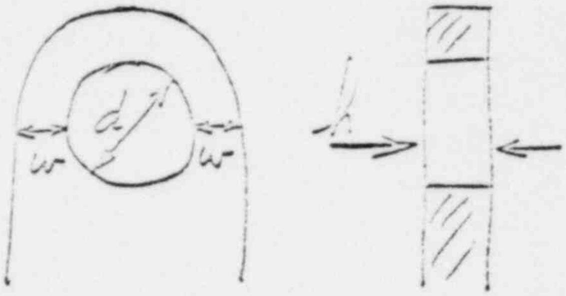
$$P = A\sigma = A\frac{S_y}{3}$$

or

$$A = 3\frac{P}{S_y} = 3 \times \frac{2900\ lb}{46000\ psi} = 0.189\ in^2$$

or

$$2hw = 0.189\ in^2$$



$h = \frac{1}{8} \quad \frac{3}{16} \quad \frac{1}{4} \quad \frac{5}{16} \quad \frac{3}{8}$ in

$w = 0.76 \quad 0.507 \quad 0.38 \quad 0.304 \quad 0.253$ in

These are the minimum allowable dimensions.

Fig L 3

A reasonable design would be a

$2\frac{1}{2} \times \frac{1}{4}$ strap

with $1\frac{1}{2}$ diameter hole

Design of Container

Fire test considerations indicate that a 6 inch wall thickness is ample. We shall use 6 inches as a minimum figure.

In the Sandia tests, with a 4000 lb load dropped from 30 ft at 45°, a 6 inch wall thickness was ample. However their outside diameter was 30 inches in all cases; thus their inside diameter was only 30 − 12 = 18 inches, and their mean diameter was 24 inches. Our dimensions are larger.

Selecting a tentative inside dimension of the cylindrical portion of our box as 26 inches, the equivalent inside diameter of a circular cylinder would be (see Drop Test Considerations p 10)

$$D = \sqrt{2}\,L = \sqrt{2} \times 26 \text{ in} = 36.8 \text{ in}$$

Perhaps it would be more convenient to convert the other way, and say that the equivalent side of the square box would be

$$L = \frac{1}{\sqrt{2}}\,D = \frac{1}{\sqrt{2}}\,18 \text{ in} = 12.7 \text{ inches}$$

This is the inside dimension of the square box that is equivalent to the Sandia cylinders.

PROJECT NO.

Using the σ equation on p (14) of the Drop Test notes,

$$\sigma = 2.12\sqrt{\frac{HWE}{Lbh}}$$

as a scaling law, we see that we must hold

$$\frac{HWE}{Lbh} = \text{constant}$$

Since H & W are both given and E is fixed by the choice of material, we have

$$HWE = \text{constant}$$

therefore

$$Lbh = \text{constant}$$

Using this to deduce the design of the USNuclear box from the dimensions of the equivalent square Sandia box, we get

$$(Lbh)_{USN} = (Lbh)_{SS}$$

or

$$h_{USN} = h_{SS}\frac{L_{SS}\,b_{SS}}{L_{USN}\,b_{USN}}$$

where

h = wall thickness of box

L = inside dimension of side of box

b = " " " length " "

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PROJECT NO.

Referring to the Sandia report (Ref 1) and to USN drawing D-0053, we get

	SS	USN
L	12.7	26
l	42.5	33
h	6	?

From these we get

$$h_{USN} = 6 \text{ in } \frac{12.7 \times 42.5}{26 \times 33} = 0.630 \times 6 \text{ in}$$

$$= 3.78 \text{ in}$$

This result indicates that a 4 inch wall would be satisfactory for protection in the drop test.

Since the fire ~~test~~ requirement is a 6 inch wall, we shall use the 6 inch wall.

Due to the possibility of crushing the edges and/or corners, we shall gusset the corners. Disregarding the gussets, the box dimensions now are

	Length	Width	Height
Inside	33	26	27
Outside	45	38	39
Outside Rings	49	42	43

General Idea - Open - No Rings

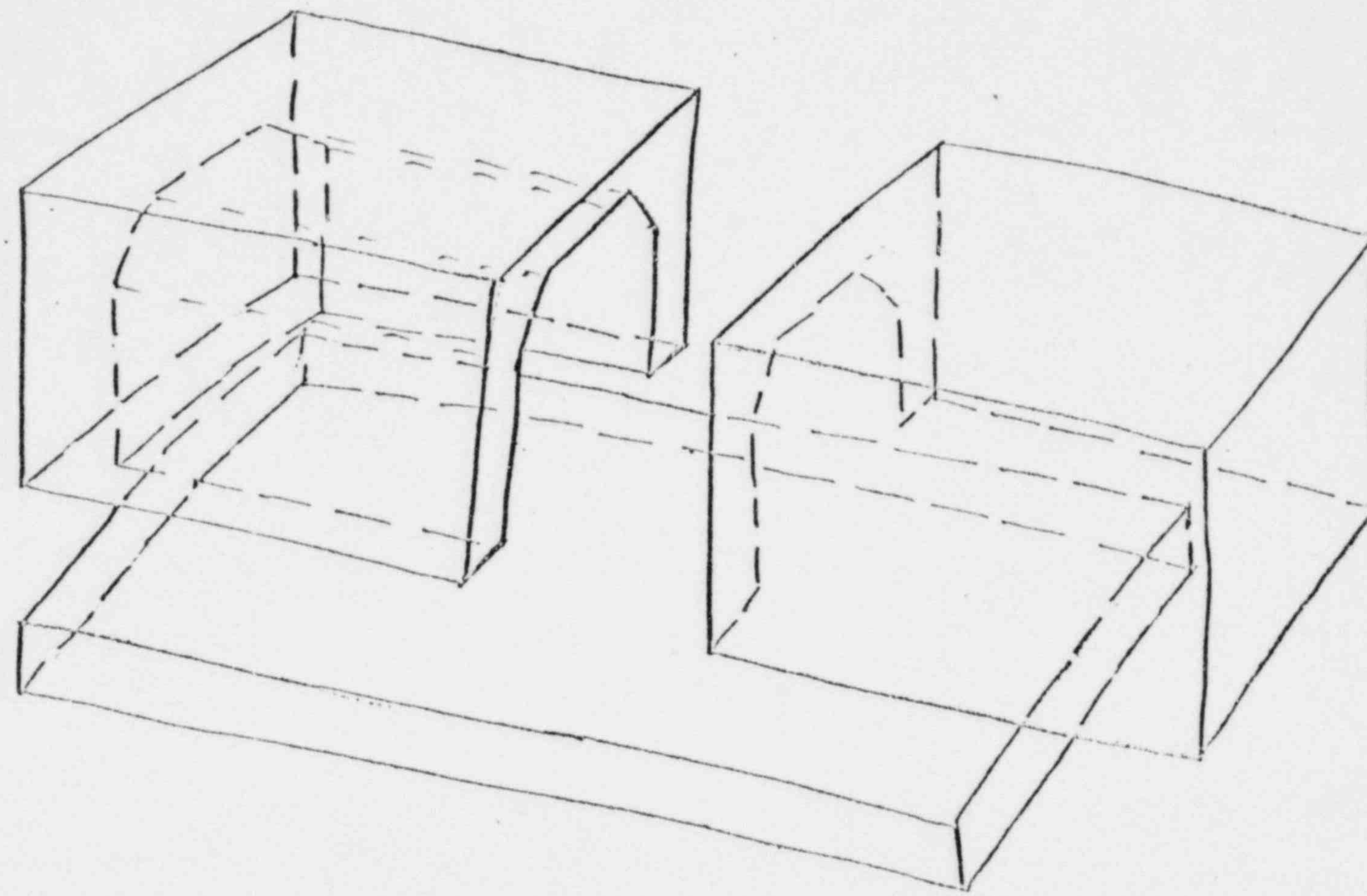

Fig C1

WKB
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PROJECT NO.

C (4)

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General Idea — Closed — No Rings

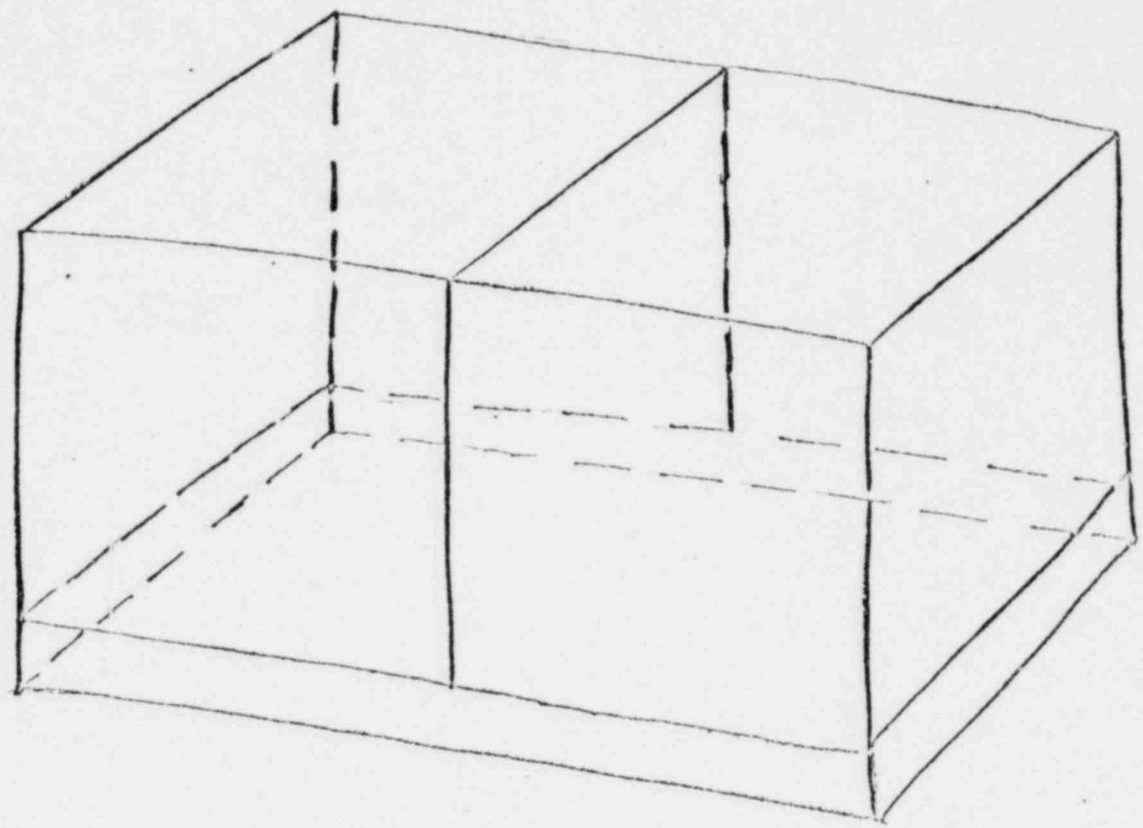

Fig C2

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C (6)

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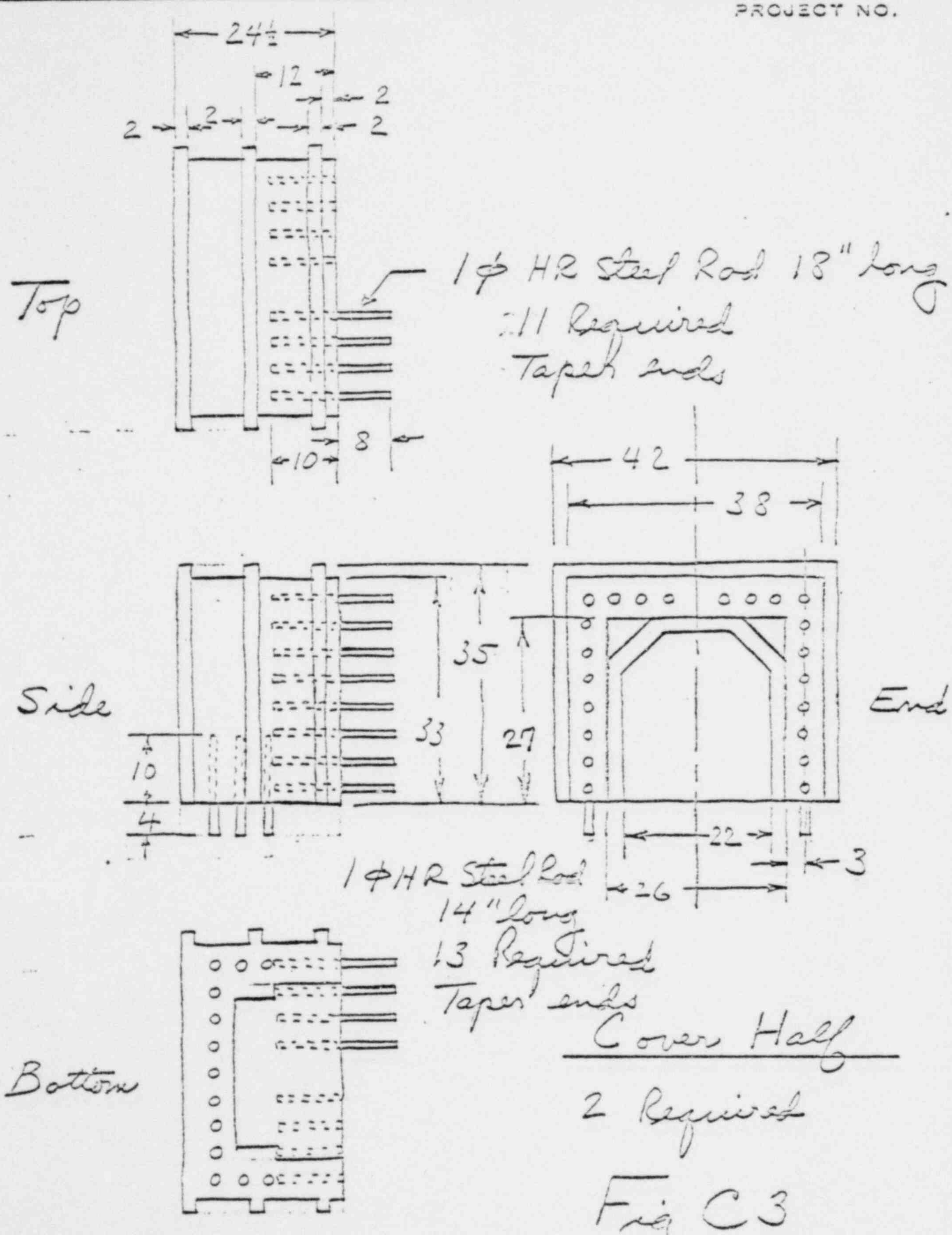

Cover Half

2 Required

Fig C3

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Base

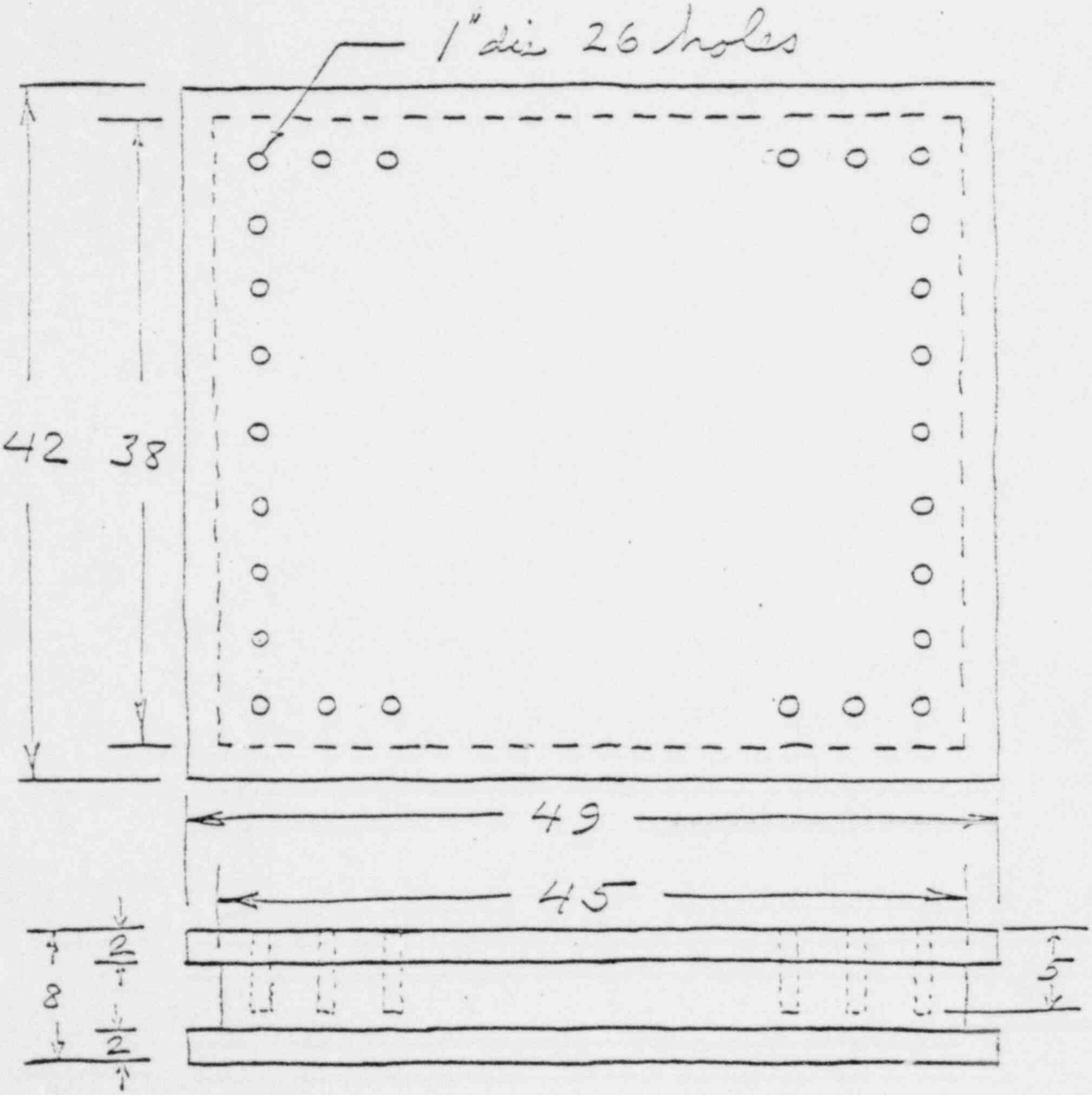

Fig C4

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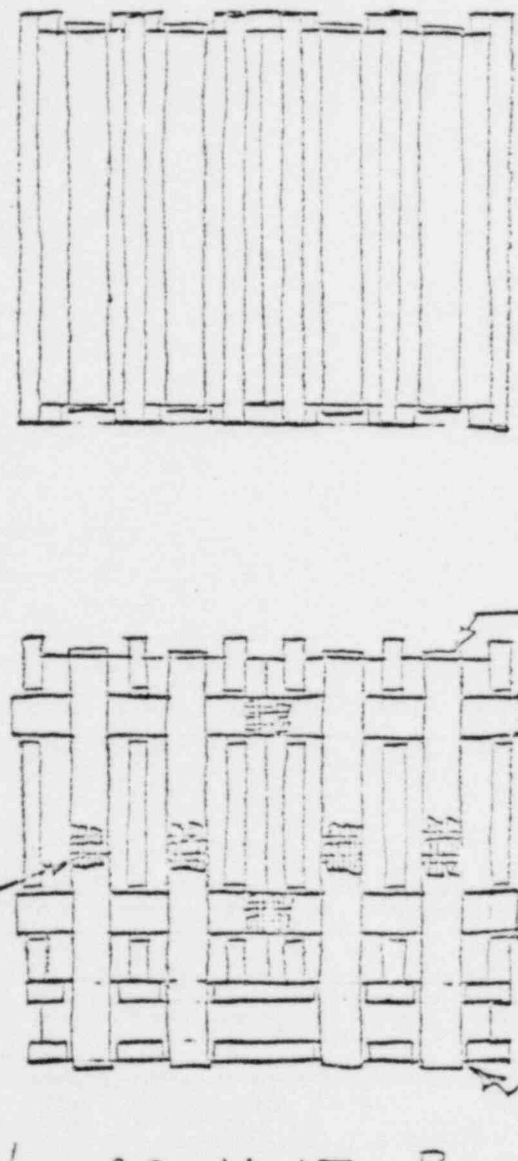
Strapping Arrangement

Fig C5

Top

Side



$\frac{1}{8} \times 4$ hot rolled steel straps
4 around
2 along

Notch crush ring as required
$4\frac{1}{2} \times 2$

$\frac{1}{2}$-20 UNF Bolts
SAE Grade 2 steel
2 per joint

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The strength requirement of the bands is somewhat problematical. The size $\frac{1}{8} \times 4$, is chosen arbitrarily; the thickness ($\frac{1}{8}$ in) is small enough that it is easily formed to fit the top without undue deformation of the steel, and with fairly light fabrication equipment. The width (4 in) is large enough that it can easily accommodate two $\frac{1}{2}$ inch bolts. The tensile strength of such a strap (before forming) will be approximately (See Ref 6)

$$\text{strength} = \tfrac{1}{8}\,\text{in} \times 4\,\text{in} \times 60\,000\ \text{psi}$$

$$= 30\,000\ \text{lb}$$

The strength of one $\frac{1}{2}$-20 UNF SAE Grade 2 steel bolt will be about (Ref 6)

$$\text{strength} = 0.1599\ \text{in}^2 \times 64\,000\ \text{psi}$$

$$= 10\,200\ \text{lb}$$

Thus two bolts will have about 2/3 of the strength of the unformed strap, or about equal the strength of the formed strap.

Maximum force holding the top halves together is about $4 \times 20000 = 80000$ lb and maximum force holding the cover to the base is about $8 \times 20000 = 160000$ lb.

Crushing load on the wood is about 5000 lb/in width. This compares with about 800 psi crush strength (Ref 4, Table 12).

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To summarize, the container should have dimensions as outlined on p(3) - (9) of these notes. Material should be 3/4 inch douglas fir plywood, exterior grade. Successive layers of plywood to be bonded with resorcinol-formaldehyde adhesive, nailed with cement coated nails - either 8 penny driven about 20 per square foot or 10 penny driven about 12 per square foot - and cured under a pressure of 180 to 200 psi.

The cask should be bolted to the base; the details of this attachment are not outlined in these calculations. The two halves of the cover are then fitted together, and this assembly is attached to the base. All straps are then bolted tight.

10-10 A completely different design can also be used. This is the exo-skeleton design, in which a steel framework carries the main structural loads at impact, and the wood panels serve most as fire protection.

Each wood panel consists of several layers of Douglas fir plywood, bonded together with resorcinol-formaldehyde adhesive. The plywood sheets should not be nailed together. The outer surface of each panel should be sheathed with 1/16 inch steel sheet. This steel sheet should not be bonded to the wood, as there must be a path for the escape of the volatile components as the wood distills and chars. The steel should be firmly nailed all around the periphery, and there should

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be 6 nails spaced roughly uniformly in the central area of the panel. The 6 nails in the center will keep the steel sheet from bulging in normal operation & handling. In a fire, they may pull out or their heads may burn off, allowing the volatiles to escape through the nail holes. The nails around the periphery will be protected from coming out, by being covered by the endo-skeleton. All nails should be cement coated.

The cork will be mounted on small bolts inside the container. These will be sufficient for normal handling; they will shear under severe impact loading, such as in the drop test. The impact will be induced by 2×2 redwood crush strips mounted on the inside of the container. The crush strips should be glued & nailed to the container.

Structurally, the wood panels fit snugly into the steel exo-skeleton, giving shear rigidity to the skeleton - ie preventing the frames from acting as 4-bar linkages. The wood panels are fitted loosely to each other at the edges & corners; there are no bending moments at these joints.

The wood panels should be about as good for fire resistance with flat surfaces exposed as with edges exposed, except that plywood delaminates when burning. The steel sheathing is expected to prevent delamination.

ABSTRACT

Additional design studies have been carried out on the family of shipping containers proposed in a report dated November 13, 1968. The details are outlined in this Supplement Number 1.

Two significant changes in manufacturing procedure have been made: A "welded-up box" type of construction has been adopted. And a high-strength alloy steel has been substituted for the low-carbon steel.

I INTRODUCTION

The basic report on the design of the International Chemicals and Nuclear Corporation shipping container was submitted to the Department of Transportation on November 13, 1968. After submittal of this report, additional design studies were carried out. As a result of these design studies, changes in manufacturing procedures are being made. The basic design concept remains the same.

In this report, the changes and the logic underlying them are outlined.

II CONSIDERATIONS OF IMPACT RESISTANCE

In a free fall, the container can land (on an unyielding surface) in any of the following ways:

- a Side contact
- b Edge contact
- c Corner contact
- d Combination of side and edge contact
- e Combination of edge and corner contact
- f Combination of side and corner contact
- g Combination of side, edge, and corner contact.

Of these 7 landing positions

- a is least severe. There is only a tendency to crush the wood panel due to direct compressive load.

b is most severe. If the total impact force is F, there is a load $F_p = \frac{1}{\sqrt{2}} F$ acting at the center of each of the two adjacent panels.

c is somewhat less severe. The load on each of the adjacent panels is ${}_cF_p' = \frac{1}{\sqrt{3}} F = 0.82\ F_p$.

d, e, f, and g are less severe than b.

d the impact force can be resolved into 2 components, one as in case a, one as in case b.

e the impact force can be resolved into 2 components, one as in case b, the other as in case c.

f the impact force can be resolved into 2 components, one as in case a, the other as in case c.

g the impact force can be resolved into 3 components, one as in case a, one as in case b, the other as in case c. This is the most general case of landing.

In each case, when the force is resolved into components, only a component (at the most) is acting as in case b. The other components are acting in the less severe fashion of case a and/or case c. For this reason, all other cases are less severe than case b.

Landing as in case b has been studied in detail; the calculations are appended in the work-sheets entitled "Exoskeleton Considerations". These calculations form the basis of the changes in manufacturing procedures, which are recorded here as design changes.

There are 2 significant changes:

1. The upper members of the exoskeleton are welded together to make a box, and the steel sheathing is welded to the exoskeleton instead of being attached to the wood panels. This forms a box type cover, of solid steel. The 5 wood panels are fitted into this steel box; they still are not bonded at the edges, and hence have no bending moments at the edges. The frame and steel sheathing of the base also are welded together as a unit, and the wood panel of the base is fitted to this unit. The cover is bolted to the base at assembly.
2. The skeleton members, sheathing, and bolts all are made of 4130 AMS ~~6350~~ alloy steel, heat treated to Rc 42-44 after welding. This treatment gives

Tensile strength	201,000 psi
Yield strength	175,000 psi
Elongation	10%
Reduction of area	42%

In the severe case of landing squarely and symmetrically on an edge, the steel case will be deformed. Plastic deformation of the steel will absorb the energy. The side panels will bulge about 5 inches, the wood panels inside will break, but the steel outer shell is expected not to fail, as the maximum strain will be about 5%. The bolts also will deform plastically, but will not fail.

III CONSIDERATIONS OF PUNCTURE RESISTANCE

The original design was shown to be puncture resistant in the "open" areas of the panels. Since the panels are now welded to the exoskeleton, the skeleton members can be considered to be at least as penetration-resistant as the panels. In addition, the use of a high-strength steel has increased the tensile strength by more than a factor of 3, decreasing the steel thickness requirement by a factor of $2\frac{1}{4}$.

The present design, therefore, can be considered to be completely resistant to puncture.

IV CONSIDERATIONS OF FIRE RESISTANCE

The original design was considered to be completely satisfactory in regard to fire resistance. None of the changes has been of such a nature as to diminish this resistance.

Welding of the steel panels to the exoskeleton frame may improve the fire resistance. This would be the case if there had originally been any tendency for the skeleton members to warp or deflect away from the panels, exposing the wood to the flames.

V CONTAINER DESIGN

The main items of change in the container design are the welded-up construction, and change of type of steel, as described in section II.

The row of bolts around the apron at the underside of the base, which holds the cover to the base, is designed to carry the load imposed by the impact force on the panels. This bolt load is the portion of the panel load which the bolted edge is required to carry. The remainder of the panel load is carried in tension by the steel sheathing at the other edges of the panel. The bolt loading is partly shear and partly tensile; the bolts are designed to withstand this combined load. The apron also is designed to withstand the loading imposed on it by the bolts.

Ex 2-13-69
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Exoskeleton Considerations

There are several considerations that relate directly to the steel framework - ie the exoskeleton - of the shipping container. They were not studied explicitly in the original design studies, but will be covered here.

This work is a supplement to the report "Design of a Family of Shipping Containers for the Transfer of Teletherapy Cobalt Sources", submitted November 13, 1968. References to the report or to the basic calculations or to literature references, all refer to those in the November 13 report, unless otherwise specified.

This study is subdivided into the following parts:

1. Puncture Resistance
2. Impact Resistance
   a. Side Contact
   b. Edge Contact
   c. Corner Contact
3. Fire Resistance

All relate specifically to the durability of the exoskeleton. All follow the same pattern as the corresponding work in the Nov 13 report.

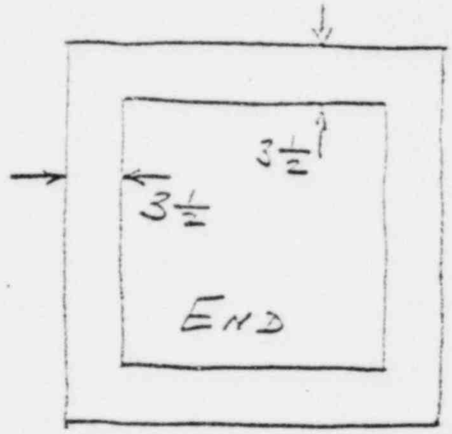
W. K. BOGGER

1. Puncture Resistance

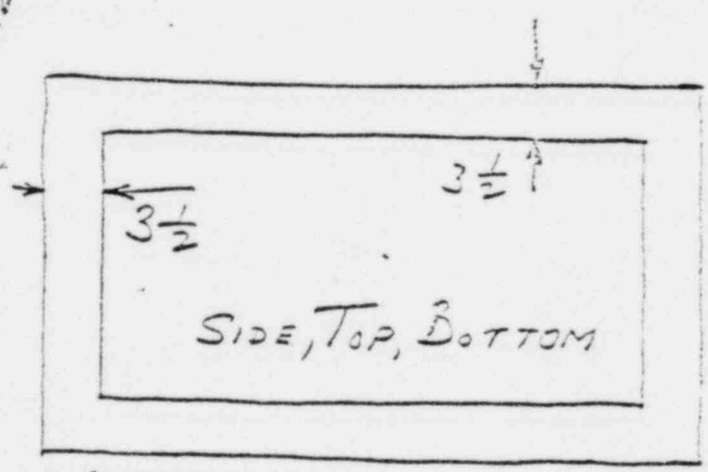
The question of penetration of the shipping container in the "spike test" has been investigated in the section of original calculations titled "Spike Test Considerations." The specific question of penetration or severance of one of the members of the exoskeleton was not checked.

The construction is as indicated in Fig Ex 1. The framework is of angle sections, $3\frac{1}{2}$ wide on each leg, by $\frac{3}{16}$ thick. About half of this $3\frac{1}{2}$ inch width is backed up by the $\frac{1}{8}$ inch sheet covering the wood panels; all of it is backed up by the $4\frac{1}{2}$ inch thick wood panels.

The wood panel with its sheet covering has already been shown to be puncture resistant. Is the corner member also resistant?



(a)



(b)

Fig Ex 1

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Evans, Nalins, & Stoddart (Ref 8, p 255) found that a thickness of

$$h = \left(\frac{W}{S_u}\right)^{0.71}$$

gave marginal puncture resistance. For weight W = 5000 lb and steel strength S_u = 63000 psi, this gives a required thickness of h = 0.165 in.

A continuous sheet of 3/16 in = 0.1875 in steel would be more than adequate. A $3\frac{1}{2}$ in wide strip of $\frac{3}{16}$ in steel covers only about $\frac{1}{2}$ of the spike; this, however, is adequately compensated by the $\frac{1}{2}$ inch backup sheet which backs up half the $3\frac{1}{2}$ inch width, and all the surface not covered by the exoskeleton. Evans, et. al. found that there were significant deformations over a region whose diameter was 3 times the punch diameter. This indicates that the ½ inch cover sheet will be involved in the deformation, and much of the reasoning leading to the conclusion that the $\frac{1}{2}$ in cover was satisfactory also applies here.

In addition, the entire steel structure is backed up the wood panelling. In the "open" area, the panelling is $4\frac{1}{2}$ inches thick and the reasoning is as described in "Spike Test Considerations". Behind the $3\frac{1}{2}$ inch exoskeleton member, the wood is in effect from 3 to 4 feet thick, since it is supported by the neighboring panel "edge on".

W. R. Dodson

While this treatment is less direct than that in the "open" area, the conclusion seems to be that the spike will not penetrate the exoskeleton member. It also follows that the member will not be severed. If the member had been "severed", it would have been only one leg of a symmetric angle, and the (member) would still have held the container together.

Ex

PROJECT NO.

2. Impact Resistance

The side panels have been shown, in notes on "Drop Test Considerations" to be adequate for the 30 foot free fall. The question remains - Will the skeleton remain completely intact, without "opening-up" at the edges and allowing the panels to slip out?

The container can land in any position. There are 3 "ideal" positions which typify & characterize the possibilities and which will serve to check the results. These are, landing on a. a side

b. an edge

c. a corner.

Take them in turns.

a. Side Contact

If the container lands on a side, the cask will fail the hold-down bolts and tend to crush the crush strips. After crushing the crush strips, the cask will tend to crush the $4\frac{1}{2}$ inch thick panel. While it may make an impression, it can not punch out a portion of the panel, because the panel will be backed up by the hard landing surface. Any impression will be wide and shallow because of the small curvature (i.e. large radius of curvature) of the cask.

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b. Edge Contact.

The "ideal" case, or the simple one to analyze, is the symmetric case of Fig Ex 2. Any case between this case and side contact (part a.) will be intermediate in intensity, as the impacting force will be divided into two components: one acting as in Fig Ex 2, the other as in part a, above.

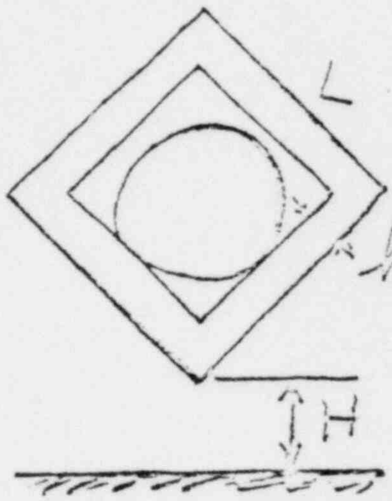

Fig Ex 2

The actual magnitude of the force at the edges of the pangle, tending to open up the angles of the exoskeleton, can be estimated.

At impact, the force exerted by the cask is

$$F = ma = \frac{Wa}{g}$$

where

m = mass of cask
W = weight " "
a = acceleration "
$= V\omega_0$
ω_0 = natural frequency

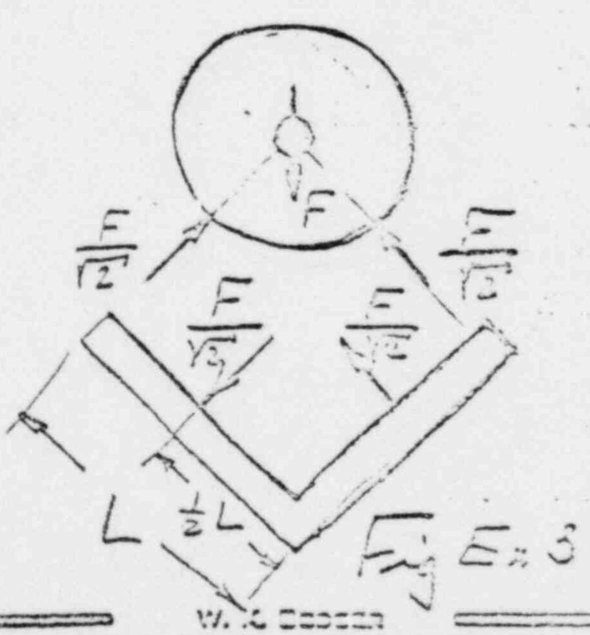

Fig Ex 3

The natural frequency of the system is

$$\omega_0 = \sqrt{\frac{k}{m}} = \sqrt{\frac{kg}{W}}$$

$$= \sqrt{\frac{2g\,E h^3\left[1+0.462\left(\frac{b}{a}\right)^4\right]}{0.203\,(1-\nu^2)\,W\,b^2}}$$

and the acceleration at impact (see p ⑥) is

$$a = V\omega_0 = V\sqrt{\frac{2g\,E h^3\left[1+0.432\left(\frac{b}{a}\right)^4\right]}{0.203\,(1-\nu^2)\,W\,b^2}}$$

The impact velocity, V, comes from a free fall from height H, so

$$V = \sqrt{2gH}$$

and

$$a = \sqrt{2gH\frac{2g\,E h^3\left[1+0.462\left(\frac{b}{a}\right)^4\right]}{0.203(1-\nu^2)\,W\,b^2}}$$

or

$$\frac{a}{g} = \sqrt{\frac{4HE\,h^3\left[1+0.462\left(\frac{b}{a}\right)^4\right]}{0.203\,(1-\nu^2)\,W\,b^2}}$$

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The force acting on one panel then becomes

$$F_p = \frac{F}{\sqrt{2}} = \frac{1}{\sqrt{2}} W \frac{a}{g}$$

$$= \frac{W}{\sqrt{2}} \sqrt{\frac{4HEh^3[1+0.462\left(\frac{b}{a}\right)^4]}{0.203(1-\nu^2)Wb^2}}$$

$$= \sqrt{\frac{2WHEh^3[1+0.462\left(\frac{b}{a}\right)^4]}{0.203(1-\nu^2)b^2}}$$

The "opening up" force along the exoskeleton, ie the force per unit length, will be

$$f = \frac{F_p}{2a+2b} = \sqrt{\frac{WHEh^3[1+0.462\left(\frac{b}{a}\right)^4]}{0.406(1-\nu^2)(a+b)^2 b^2}}$$

For numerical values

$W = 4000$ lb
$H = 30$ ft $= 360$ in
$E = 1.95 \cdot 10^6$ psi (Ref 4)
$h = 4\frac{1}{2}$ in
$\nu = 0.45$ (Ref 4, p 79)
$a = 48\frac{3}{4} - 3\frac{1}{2} = 45\frac{1}{4}$ in
$b = 36\frac{1}{2} - 3\frac{1}{2} = 33$ in

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PROJECT NO.

$$f = \sqrt{\frac{4000\,\text{lb} \cdot 360\,\text{in} \cdot 1.95 \cdot 10^6 \frac{\text{lb}}{\text{in}^2} (4.5\,\text{in})^3 \left[1 + 0.432 \left(\frac{33}{45.25\,\text{in}}\right)^4\right]}{0.406 \left[1-(0.45)^2\right] (78.25\,\text{in})^2 (33\,\text{in})^2}}$$

$$= \frac{2 \times 6 \times 4.5}{78.25 \times 33} 10^5 \frac{\text{lb}}{\text{in}} \sqrt{\frac{1.95 \times 4.5 \left[1 + 0.432 \overbrace{\left(\frac{33}{45.25}\right)^4}^{1.131}\right]}{0.406 \underbrace{\left[1-(0.45)^2\right]}_{0.798}}}$$

$$= 2090\,\frac{\text{lb}}{\text{in}} \overbrace{\sqrt{30.6}}^{5.51} = 11500\,\frac{\text{lb}}{\text{in}}$$

This load must be carried by the angle section of the exoskeleton, in bending.

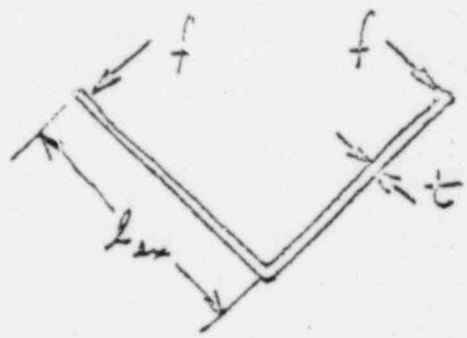


Fig Ex 5

$$\sigma = \frac{Mc}{I}$$

$$= \frac{f l_{ex} \frac{t}{2}}{\frac{t^3}{12}}$$

$$= 6 \frac{f l_{ex}}{t^2}$$

$$= 6 \frac{11500\,\frac{\text{lb}}{\text{in}}\, 3.5\,\text{in}}{(3/16\,\text{in})^2} = 686000\ \text{psi}$$

Too high by a factor of 100 or more.
A different support arrangement must be devised.

Ex

PROJECT NO.

Assuming that a satisfactory design can be devised, for holding the edge of the wood panels, what will be the stresses in the panels.

First, the shear stress in the panels, near the edge.

$$\tau = \frac{f}{h} = \frac{11500 \text{ lb/in}}{4.5 \text{ in}} = 2560 \text{ psi}$$

By Ref 4, p 75, Douglas-fir seems to have a shear strength of about 1100 psi. This stress thus seems high by a factor of about 2.3.

Part of the shear load will be carried by the steel sheathing.

$$\tau_{steel} = \frac{1}{2} S_u = \frac{1}{2}\, 63000 \text{ psi} = 31500 \text{ psi}$$

so

$$f_{steel} = h_{steel}\tau_{steel} = \frac{1}{8} \text{ in } 31500 \text{ psi}$$

$$\cong 4000 \text{ lb/in}$$

The net load to be carried by the wood panel then is

$$f_{wood} = f - f_{steel}$$

$$= 11500 \text{ lb/in} - 4000 \text{ lb/in}$$

$$= 7500 \text{ lb/in}$$

and the shear stress in the wood is

$$\tau_{wood} = \frac{f_{wood}}{h_{wood}} = \frac{7500}{4.5} = 1700 \text{ psi}$$

Still high.

W. K. COOPER

PROJECT NO.

The bending stress in the plate is a little harder to define. For a plate with a "concentrated" load, there would be an infinite stress at the load, hence the "concentrated" idealization must be abandoned. Roark, Ref 5, p 203, gives a relation for the bending stress in a plate which is loaded by a force uniformly distributed over a small area (circular) of radius r_0, at the center of the rectangular plate:

$$\sigma = \frac{3}{2}\left(\frac{F}{\sqrt{2}}\right)\frac{1}{\pi h^2}\left[(1+\nu)\ln\left(\frac{b}{2r_0}\right) + 1 + \frac{0.914}{1+1.6\left(\frac{b}{a}\right)^5} - 0.6\right]$$

$$= \frac{3F}{2\sqrt{2}\,\pi h^2}\left[0.40 + (1+\nu)\ln\left(\frac{b}{2r_0}\right) + \frac{0.914}{1+1.6\left(\frac{b}{a}\right)^5}\right]$$

where

r_0 = described in paragraph above

other symbols as on p 7.

Now, as on p 6,

$$F = \frac{a}{g}\,W$$

so

$$\sigma = \frac{3\,W}{2\sqrt{2}\,\pi h^2}\left[0.40 + (1+\nu)\ln\left(\frac{b}{2r_0}\right) + \frac{0.914}{1+1.6\left(\frac{b}{a}\right)^5}\right] \times \sqrt{\frac{4HEh^3\left[1+0.452\left(\frac{b}{a}\right)^4\right]}{0.203(1-\nu^2)\,W b^2}}$$

$$= \frac{3}{\pi}\left[0.40 + (1+\nu)\ln\left(\frac{b}{2r_0}\right) + \frac{0.914}{1+1.6\left(\frac{b}{a}\right)^5}\right]\sqrt{\frac{WHE\left[1+0.452\left(\frac{b}{a}\right)^4\right]}{0.203(1-\nu^2)\,h\,b^2}}$$

Ex C

PROJECT NO.

For a numerical value of σ, assume r_0 = 4 inches, an arbitrary estimate; other values as on p 9.

$$\sigma = \frac{3}{\pi}\left[0.40 + (1+0.45)\ln\left(\frac{33}{8}\right) + \frac{0.914}{1+1.6\left(\frac{33}{45\frac{1}{4}}\right)^5}\right]$$

$$\sqrt{\frac{4000\ \text{lb}\ 360\ \text{in}\ 1.95\cdot 10^6\ \frac{\text{lb}}{\text{in}^2}\left[1+0.462\left(\frac{33}{45\frac{1}{4}}\right)^9\right]}{0.406\left[1-(0.45)^2\right]\ 4.5\ \text{in}\ (33\ \text{in})^2}}$$

$$= 0.955\left[0.40 + 1.45 \cdot 1.42 + \frac{0.914}{1+0.33}\right]\frac{4\cdot 10^5\ \text{psi}}{11}\sqrt{1.07\ \frac{1+0.13}{1-0.202}}$$

$$= 34750\ \text{psi}\left[0.40 + 2.06 + 0.686\right]\sqrt{1.07\ \frac{1.13}{0.798}}$$

$$= 34750\ \text{psi}\left[3.146\right]1.231 = 135\,000\ \text{psi}$$

Compared with a modulus of rupture of about 12000 psi, (Ref 4, p 75) this is too high by a factor of about 11.

W. K. Dodson

With all 3 stresses too high,

σ	bending in wood panel high by factor	11
τ	shear " " " " " "	2 or 3
σ	bending " steel angle " " "	200

There is no reasonable way to make these stresses acceptable by changing dimensions of the present design.

The most promising approach seems to be to belt the container with steel straps, along the lines Fig C5, on p C 8 of the notes "Design of Container". These straps will take over the load of holding the container together, and will relieve all 3 of ~~the~~ the stresses at the top of the page.

The total force to be held, per panel, from p 9 & 10 is

$$F_p = 2(a+b)f = 2 \times 78\tfrac{1}{4}\text{ in } 11500 \tfrac{lb}{in} = 1800\,000\text{ lb}$$

Using low-carbon steel with an ultimate tensile strength of $S_{ut} = 63000$ psi (Ref 6, p 597) and a safety factor of $n = 1.25$, or design stress of

$$\sigma_{des} = \frac{S_{ut}}{n} = \frac{63\,000\text{ psi}}{1.25} = 50\,400 \text{ psi}$$

the total steel belt area required to hold the panel load is

$$\Sigma A_{belt} = \frac{F_p}{\sigma_{des}} = \frac{1800\,000 \text{ lb}}{50\,400 \frac{lb}{in^2}} = 35.7 \text{ in}^2$$

If the belt were distributed uniformly around the entire periphery of the panel, the metal thickness would have to be

$$t = \frac{\Sigma A}{\text{perimeter}} = \frac{35.7\,in^2}{2\left(48\frac{3}{4} + 36\frac{1}{2}\right) in} = \frac{35.7}{170.5}\,in$$

$$= 0.21\ in$$

The more probable arrangement is a set of belts of $\frac{1}{2}$ inch material; their total width would be 40% of the perimeter.

Bolt strength (tensile &/or shear) must also total to the strength of each belt, of course.

The analysis on the preceding pages has shown the impact force on the panel, and also the consequent bending and shear stresses, to be excessive. The prospects for reducing them any significant amount by changing the nominal dimensions of the shipping container are poor because the functional variations are as follows:

$$F_p \sim \sqrt{\frac{WHEh^3}{b^2}} \qquad p\ 9$$

$$\tau \sim \sqrt{\frac{WHEh}{b^2}} \qquad p\ 9,\ 10$$

$$\sigma_{wood} \sim \sqrt{\frac{WHE}{h b^2}} \qquad p\ 12$$

$$\sigma_{steel} \sim \frac{F_p}{t^2} \qquad p\ 10$$

No major change is available here. In fact a decrease in h (which is not possible anyway because of the fire requirement) would increase σ_{wood} while decreasing the other stresses.

We have no control over W or H, and practically speaking no control over E.

The design of the shipping containers used in the Sandia ~~tests~~, Ref. 1, is very good in that it has no bending, no tension, & no shear - at least no first order effects of these types. The impact load is carried by compression of the container wall, or shell, between the cask and the ground. Crush strips are used to reduce the acceleration of the cask, and the magnitude of the consequent impact load. In effect, they reduce the actual load, W, to a smaller "effective" load, W_{eff}.

If we assume that the crush strips, in crushing, apply a ~~constant~~ deceleration to the cask, then we can say

$$V^2 = 2 a s_o$$

or

$$a = \frac{V^2}{2 s_o}$$

where

V = velocity of cask at time of initial impact on crush strip

s_o = amount of crush of crush strip
= motion of cask relative to container

a = acceleration (actually a deceleration) of cask due to crushing of crush strip

The velocity V came from a free fall from a height of H, so

PROJECT NO.

$$V = \sqrt{2gH}$$

or
$$a = \frac{2gH}{2s_o} = g\frac{H}{s_o}$$

or
$$\frac{a}{g} = \frac{H}{s_o}$$

this is the acceleration of the cask during crushing of a crush strip mounted on an unyielding surface.

The analysis on p (8) - (14) ignored any effect of the crush strips; this effect was to have been a safety margin. this led to (p (14))

$$F_p = 1800000 \text{ lb}$$

Since (p (9))

$$F_p = \frac{1}{\sqrt{2}} W \frac{a}{g}$$

the acceleration of W was

$$\frac{a}{g} = \frac{\sqrt{2} F_p}{W} = \frac{\sqrt{2}\ 1800000 \text{ lb}}{4000 \text{ lb}} = 636$$

A crush strip design that will yield $\frac{a}{g} << 636$ will give an effective weight, W_{eff}, much smaller than the actual W, leading to smaller stresses.

PROJECT NO.

The controlling relation is (p 12)

$$\frac{a}{g} = \frac{H}{s_0}$$

s should be made large. But how large?

The wood panel is, in effect, just a spring – as in Fig Ex 6 (a). The crush strip, similarly, is also a spring – Fig Ex 6 (b). When they both act together, they are in series, Fig Ex 6 (c). Then

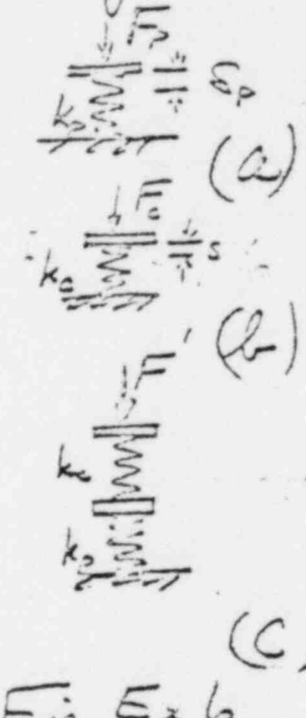


Fig Ex 6

$$F_p = k_p \delta_p$$

as on p 7, and

$$F_c = k_c s$$

when both are acting

$$\delta' = s + \delta_p' = \frac{F_c}{k_c} + \frac{F_p'}{k_p} = \frac{F_p'}{k_{equiv}}$$

and since

$$F_c = F_p'$$

$$\frac{1}{k_{equiv}} = \frac{1}{k_c} + \frac{1}{k_p}$$

or

$$k_{equiv} = \frac{1}{\frac{1}{k_c} + \frac{1}{k_p}} = \frac{k_p k_c}{k_p + k_c}$$

Ex

The purpose of the spring(s), in each case, is to absorb the energy of the cask. This energy is, in each case, assumed to be divided equally between 2 panels. So

$$\frac{1}{2} W H = \frac{1}{2} \frac{W V^2}{2g} = \frac{1}{2}\frac{1}{2} k_p \delta_p^2 = \frac{1}{2}\frac{1}{2} k_{equiv} (\delta')^2$$

or

$$\frac{\delta'}{\delta_p} = \sqrt{\frac{k_p}{k_{equiv}}} = \sqrt{\frac{k_p}{k_c} + \frac{k_p}{k_p}} = \sqrt{1 + \frac{k_p}{k_c}}$$

The impact force, in the two cases, is

$$\frac{F_p'}{F_p} = \frac{k_p \delta_p'}{k_p \delta_p} = \frac{\delta_p'}{\delta_p} = \frac{\delta' - s}{\delta_p} = \frac{\delta'}{\delta_p} - \frac{s}{\delta_p}$$

$$= \sqrt{1 + \frac{k_p}{k_c}} - \frac{F_c / k_c}{F_p / k_p} = \sqrt{1 + \frac{k_p}{k_c}} - \frac{F_p'}{F_p} \frac{k_p}{k_c}$$

or

$$\frac{F_p'}{F_p}\left[1 + \frac{k_p}{k_c}\right] = \sqrt{1 + \frac{k_p}{k_c}}$$

or

$$\frac{F_p'}{F_p} = \frac{\sqrt{1 + \frac{k_p}{k_c}}}{1 + \frac{k_p}{k_c}} = \frac{1}{\sqrt{1 + \frac{k_p}{k_c}}}$$

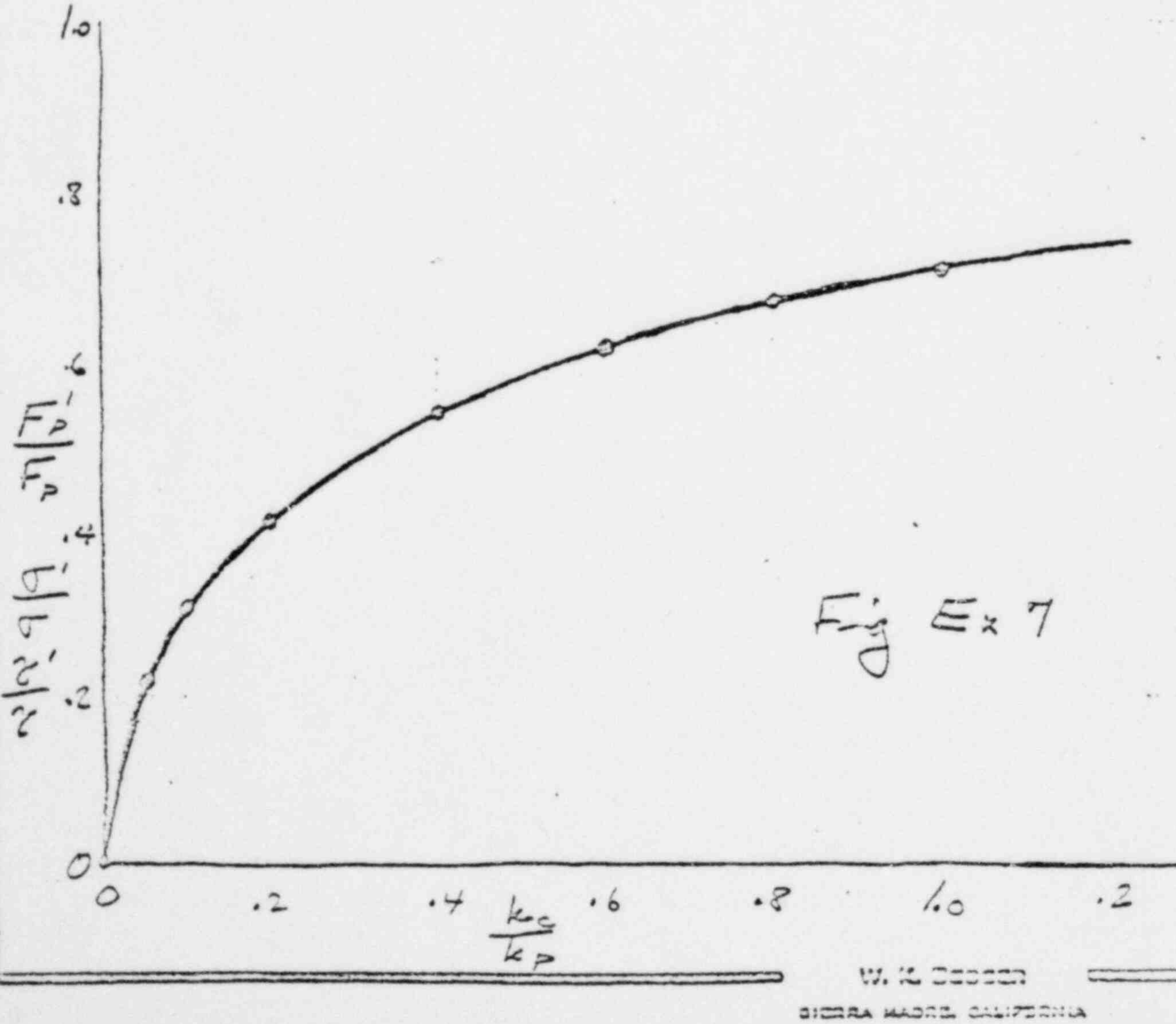
All stresses will be reduced in proportion to F_p'/F_p.

We can get k_p readily from the results on p ⑦. The next step is then to get k_c.

W. L. Dobson

To get an idea of the reduction to be obtained from k_c (ie the use of a crush strip) look at some numerical values.

$\frac{k_c}{k_p}$	0	.05	.1	.2	.4	.6	.8	1.0
$\frac{k_p}{k_c}$	∞	20	10	5	2.5	1.67	1.25	1.0
$1+\frac{k_p}{k_c}$	∞	21	11	6	3.5	2.67	2.25	2.0
$\sqrt{1+\frac{k_p}{k_c}}$	∞	4.58	3.32	2.45	1.87	1.63	1.50	1.41
$\frac{F_p'}{F_p}$	0	0.218	0.302	0.409	0.535	0.614	0.667	0.707



The curve in Fig Ex 7 tells us that, in order to accomplish anything worthwhile, we must get k_c/k_p below 0.2, and preferably below 0.1. Since little can be done about k_p, we assume that k_p will remain about constant and evaluate k_c.

To establish a target value for k_c, first evaluate k_p (p 7).

$$k_p = \frac{2Eh^3\left[1+0.482\left(\frac{b}{a}\right)^4\right]}{0.203(1-\nu^2)b^2}$$

Using numerical values on p 9,

$$k_p = \frac{2\times 1.95\cdot 10^6\,\frac{lb}{in^2}\left[1+0.482\left(\frac{33}{40\ in}\right)^4\right]\,(4.5\,in)^3}{0.203\,\left[1-(0.45)^2\right]\,(33\ in)^2}$$

$$= \frac{\overset{1.3}{\cancel{3.9}}\cdot 10^6[1.13]\ 4.5\times 4.5\times 4.5}{0.203[0.798]\,33\times \underset{11}{\cancel{33}}}\ \frac{lb}{in}$$

$$= 2.28\cdot 10^6\ \frac{lb}{in}$$

Our target then is

$$k_c \ll 2.28\cdot 10^6\ \frac{lb}{in}$$

say

$$k_c < 2.28\cdot 10^5\ \frac{lb}{in}$$

The deflection (ie the crushing) of the crush strip, in the direction normal to the plane of the panel, will be (p 19)

$$s = \frac{F_c}{k_c} = \frac{F_p'}{k_c}$$

For comparison, the deflection δ_p, normal to the plane of the panel, without considering the crush strip, is

$$\delta_p = \frac{F_p}{k_p} = \frac{1800000 \text{ lb}}{2280000 \frac{\text{lb}}{\text{in}}} = \frac{1.8}{2.28} \text{ in} = 0.79 \text{ in}$$

The amount of crushing of the crush strip will be

$$\frac{s}{\delta_p} = \frac{F_p'/k_c}{F_p/k_p} = \frac{F_p'}{F_p}\,\frac{k_p}{k_c}$$

$$= \frac{\frac{k_p}{k_c}}{\sqrt{1+\frac{k_p}{k_c}}} = \frac{1}{\sqrt{\left(\frac{k_c}{k_p}\right)^2 + \frac{k_c}{k_p}}}$$

$\frac{k_c}{k_p}$	0	.05	.1	.2	.4	.6	.8	1.0
$\left(\frac{k_c}{k_p}\right)^2$	0	.0025	.01	.04	.16	.36	.64	1.0
$\frac{k_c}{k_p}+\left(\frac{k_c}{k_p}\right)^2$	0	.0525	.11	.24	.56	.96	1.44	2.0
$\sqrt{}$ "	0	.229	.332	.490	.748	.98	1.20	1.41
$\frac{1}{\sqrt{}}$	∞	4.36	3.02	2.04	1.34	1.02	0.833	0.707

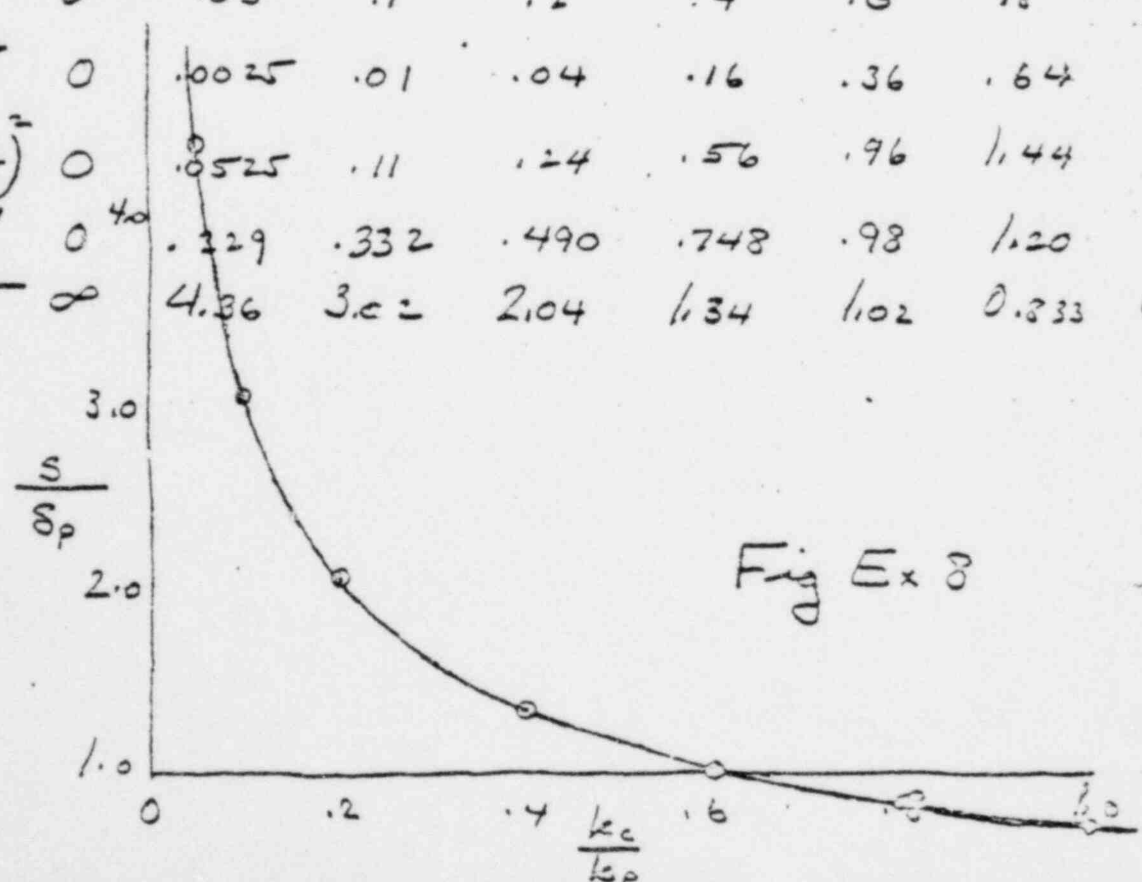


Fig Ex 8

Considering that $\delta_p = 0.79$ inch, this gives

$\frac{k_c}{k_p}$	0	.05	0.1	0.2	0.4	0.6
$\frac{s}{\delta_p}$	∞	4.36	3.02	2.04	1.34	1.02
s	∞	3.45	2.38	1.61	1.06	0.206

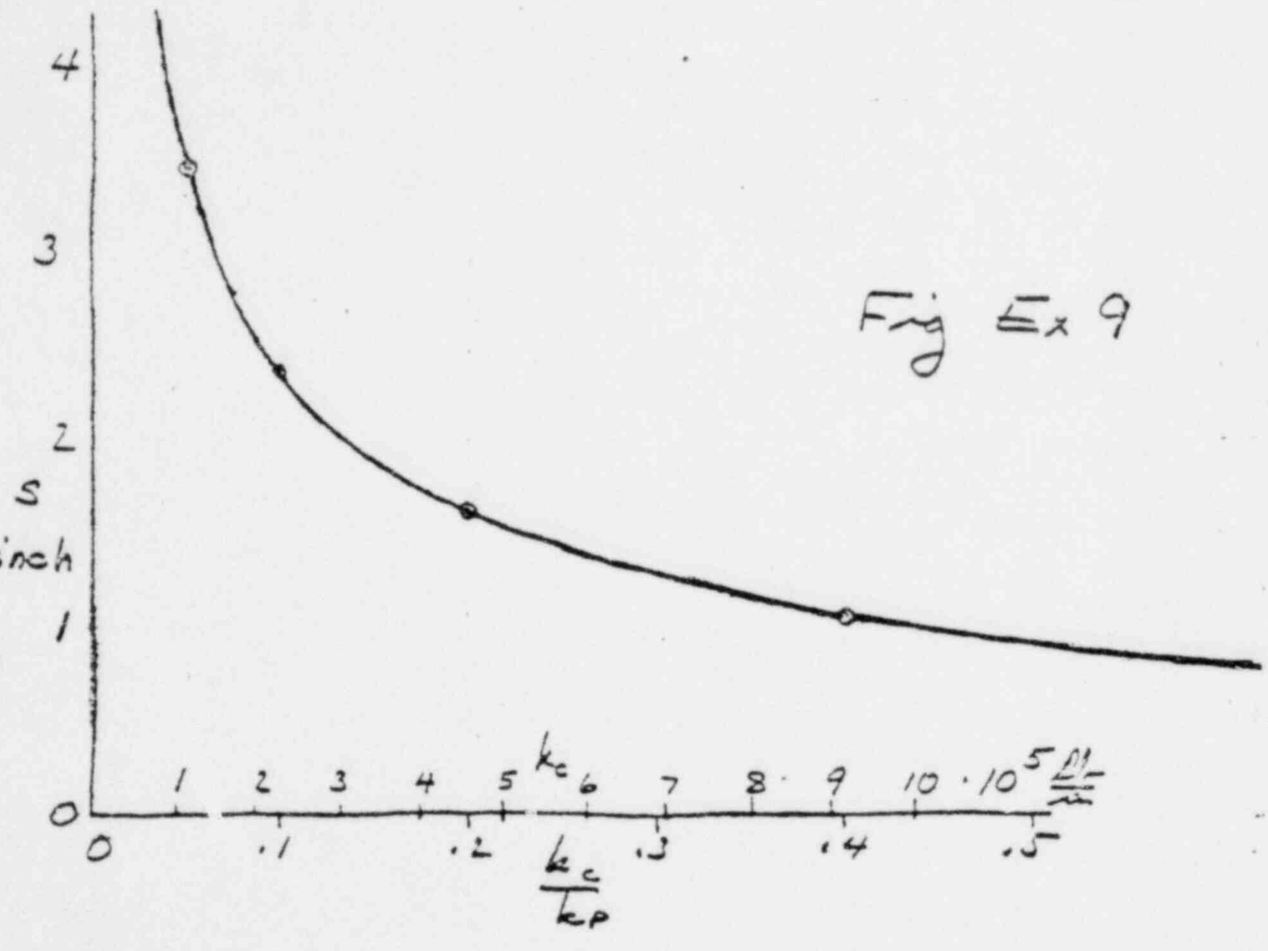


Fig Ex 9

It may be more informative to cross-plot F_p'/F_p from Fig Ex 7 vs s from Fig Ex 9. This means just eliminate the k_c/k_p parameter.

$\frac{k_c}{k_p}$	0	.05	.1	.2	.4	.6
$\frac{F_p'}{F_p}$	0	.218	.302	.409	.535	.614
s	∞	3.45	2.38	1.61	1.06	0.806 inch

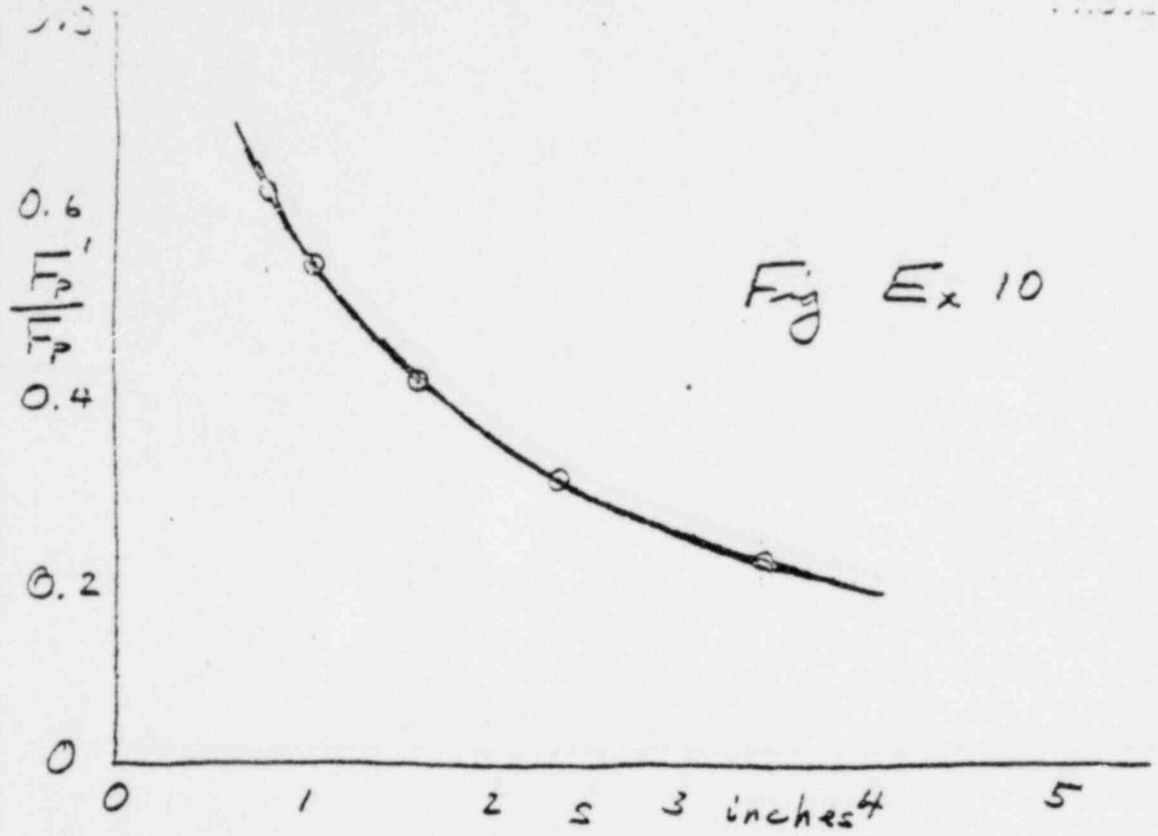


Fig Ex 10

Fig Ex 10 suggests that crush strips should be designed for a crushing of

$s > 2$ inches

and preferably

$s > 3$ inches

This would obviously mean that

crush strip thickness > 2 inches in the one ca-
" " " > 3 " " " other "

Comparison of Fig 21 and Fig 6 of Ref 1 indicates that the 2 x 2 inch crush rings crushed completely in the Sandia tests. The crushed portion was the entire segment of the annular crush ring.

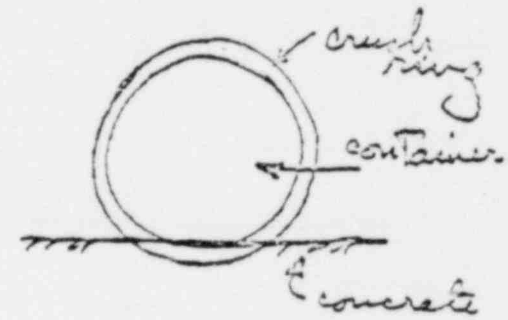


Fig Ex 11

In the present design, the construction is different in that the crush strip is a straight piece on the inner surface of the structural panel. The geometry of crushing, however is almost identical as seen in Fig Ex 12. We can therefore assume that the crush strips can be crushed completely. This means

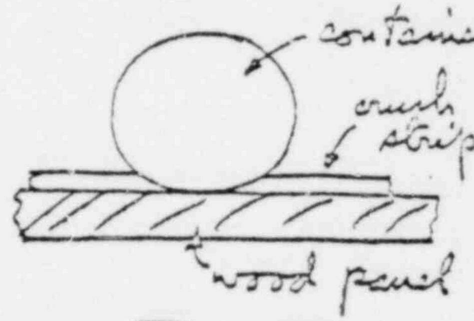


Fig Ex 12

crush strip thickness = s

Determination of k_c is another matter. In the classical, or elastic, case – as in Fig Ex 13(a) – k_c follows from

$$\delta = \frac{P l}{A E}$$

or

$$k = \frac{P}{\delta} = \frac{A E}{l}$$

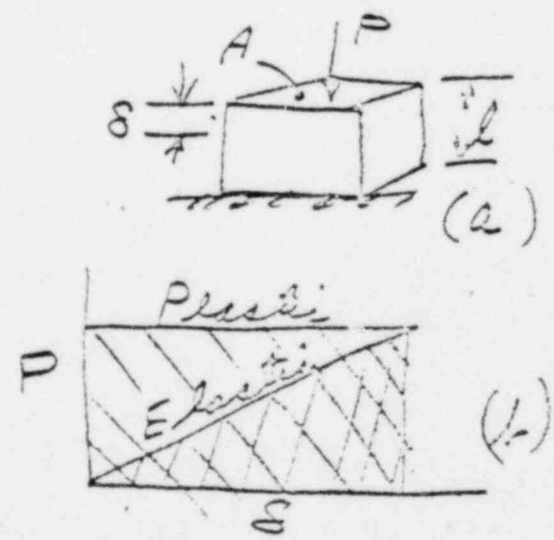


Fig Ex 13

However in crushing there is no E. The P–δ curve is like the plastic curve of Fig E_x 13 (b). The work required to cause the deformation, however, is

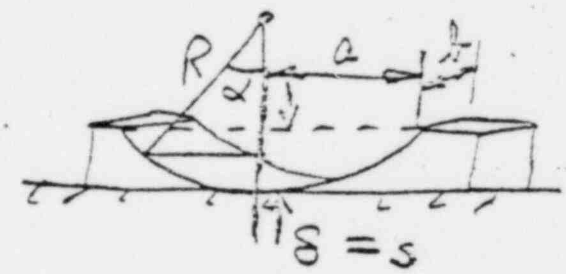
$$\text{Work} = \tfrac{1}{2} k \delta^2 \qquad \text{elastic}$$
$$= P\delta \qquad \text{plastic or crushing}$$

thus

$$k_c = \frac{P\delta}{\frac{1}{2}\delta^2} = \frac{2P}{\delta}$$
$$= \frac{2A\,\sigma_{crush}}{\delta}$$
$$= \frac{2A\,\sigma_{crush}}{s}$$

since the entire thickness of the crush strip is the amplitude of crushing motion.

The area, A, comes from



$$s = R - R\cos\alpha$$
$$= R[1-\cos\alpha]$$
$$= R\left[1-(1-\tfrac{1}{2}\alpha^2+\dots)\right]$$
$$= \tfrac{1}{2}R\alpha^2$$
$$a = R\sin\alpha = R\alpha$$
$$s = \tfrac{1}{2}R\left(\frac{a}{R}\right)^2$$
$$= \frac{a^2}{2R}$$

or

$$a = \sqrt{2Rs}$$
$$A = 2ab = 2\sqrt{2}\,b\sqrt{Rs}$$

Then

$$k_c = \frac{2\sqrt{2}\, b \sqrt{Rs}}{s}\, \sigma_{crush}$$

$$= 4\sqrt{2}\sqrt{\frac{R}{s}}\, b\, \sigma_{crush}$$

$$= 4\sqrt{2}\sqrt{\frac{D}{2s}}\, b\, \sigma_{crush}$$

$$= 4\sqrt{\frac{D}{s}}\, b\, \sigma_{crush}$$

where

k_c = equivalent spring "constant" of crush strip

R, D = radius & diameter of cask

s = amplitude of crushing

b = width of crush strip

σ_{crush} = crushing strength of crush strip

Data on σ_{crush} are not readily available. The Wood Handbook (Ref 4, Table 12) has some data, however:

	Compression Parallel to Grain Prop. Limit	Compression Parallel to Grain Max Crush	Perpendicular to Gr. Prop. Limit
Douglas Fir	5850	7430	870
Redwood	4560	6150	860

The "parallel to grain" loading is virtually a buckling type load, hence the maximum load is only a little more than the initial failure. The "perpendicular to grain" loading is the first yielding of a soft material that gets "harder" as it deforms. Since we are concerned with very large deformation — very large compressive strain at least — it seems reasonable to assume the same value for maximum compression perpendicular to the grain as parallel to it. Use, as a design figure

$$\sigma_{crush} = 6000 \text{ psi}$$

For numerical values use

D = 24 inches (Dwg F00100-1)

b = 4 inches (2 strips, each 2") (Dwg D00103-1)

s = 2 inches " "

σ_{crush} = 6000 psi

Then

$$k_c = 4\sqrt{\frac{D}{s}}\, b\, \sigma_{crush}$$

$$= 4\sqrt{\frac{24 \text{ in}}{2 \text{ in}}}\; 4 \text{ in}\; 6000 \frac{\text{lb}}{\text{in}^2}$$

$$= 333\,000 \frac{\text{lb}}{\text{in}}$$

Comparing this result with Fig Ex 9, we see that

$$s = 1.9 \text{ inch}$$

This agrees well enough. This corresponds to

$$\frac{k_c}{k_p} = 0.15 \;,\; \frac{s}{\delta_p} = 2.4 \;,\; \frac{F_p'}{F_p} = 0.36$$

This gives a basis for correcting the [illegible] on p 10, 11, 13, 14, 15, 18, 23

Page	Item	Original	Factor	Actual	Allowable	High by [illegible]
10	f	11500 $\frac{lb}{in}$	0.36	4140 $\frac{lb}{in}$	—	—
10	τ_{steel}	6860000 psi	↓	2470000 psi	63000 psi	39.2
11	τ	2560 psi		923 psi	1100 psi	OK
13	σ	135000 psi		48600 psi	12000 psi	4.05
14	F_p	1800000 lb		649000 lb	—	—
14	ΣA_{bolt}	35.7 in²		12.9 in²	—	—
15	t_{avg}	0.21 in		0.0756 in	—	—
18	$\frac{a}{g}$	636 –		229 –	—	—
23	δ_p	0.790 in		0.284 in	—	—

This looks much better, but not yet good enough.

We can get a still better result by using a crush strip 3 inches deep (ie s = 3 in) by 2 in wide.

$$k_c = 4\sqrt{\frac{24}{3}}\, 4 \times 6000 = 272\,000 \ \frac{lb}{in}$$

From Fig Ex 9, s = 2.1 in.

This does not match at all. Use crush strips 3 in wide by 1 in wide.

$$k_c = \frac{1}{2} \times 272\,000 \ \frac{lb}{in} = 136\,000 \ \frac{lb}{in}$$

From Fig Ex 9, s = 3.1 in. Good enough.

So

$$\frac{k_c}{k_p} = 0.0595 \ , \ \frac{s}{\delta_p} = 4.0 \ , \ \frac{F_p'}{F_p} = 0.23$$

Modifying the basic results again

Page	Item	Original	Factor	Actual	Allowable	High by
10	f	11500 lb/in	0.23	2640 lb/in	—	—
12	σ_{steel}	686000 psi		158000 psi	6300 psi	25.1
11	τ	2560 psi		590 psi	1100 psi	OK
13	σ	135000 psi		31000 psi	12500 psi	2.53
14	F_p	180000 lb		41500 lb	—	—
14	ΣA_{bolt}	35.7 in²		8.21 in²	—	—
15	t_{avg}	0.21 in		0.0483 in	—	—
18	$\frac{a}{g}$	636 -		147 -	—	—
23	δ_p	0.790 in		0.182 in.	—	—

Still better, but not good enough.

W. K. DODGE

PROJECT NO.

The simplest solution to this design problem is to make several design changes:

1. Weld the 1/8 inch steel sheets to the angle members of the exoskeleton, to make better use of the strength of these sheets. The 8 upper angle members and 5 of the sheets will be welded together to form a single unit. This unit, with the 5 wood panels, will be the cover.
2. Use high-strength heat-treated steel, instead of low-carbon steel, to get high strength with light weight. The assembly will be heat treated after welding.
3. The 5 wood panels will be permanently assembled in the steel box, but will not be closely attached to the 1/8 inch steel sheets as was previously planned.

The steel to be used will be

AMS 6350

Bought and fabricated in annealed form, it can be heat treated to

R_c 42-44, avg 43

Properties are, at R_c 43

Tensile ultimate	S_u = 201 000 psi
" yield	S_y = 175 000 psi
Elongation	10%
Reduction of area	42%

PROJECT NO.

Referring to p 30, we use the data for the case of the original 2×2 crush strips.

The "opening up" load of the panels

$$f = 4140 \text{ lb/in}$$

will be carried by direct tension in the $\frac{1}{8}$ inch steel sheet. This gives

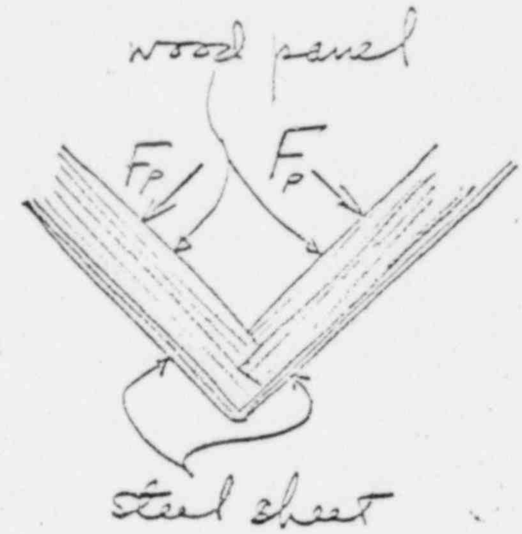


Fig Ex 15

$$\sigma_{steel} = \frac{f}{t}$$

$$= \frac{4140 \text{ lb/in}}{1/8 \text{ in}}$$

$$= 33120 \text{ psi.}$$

Compared with 175000 psi yield, the safety factor is

$$n = \frac{175000}{33120} = 5.28$$

Far more than adequate.

Bending in the wood panel was excessive by a factor of 4.05. The wood panel will break and the load must be carried by the steel sheet. Energy will be absorbed by the breaking of the wood panel – the amount can be estimated (roughly) quite easily.

Ref 4, table 12, gives

Work to Maximum Load, in static bending, $= 9.8 \frac{\text{in lb}}{\text{in}^3}$

If we assume the entire volume of each of the wood panels absorbs this amount of work uniformly we get (data from p 9)

$$\text{Volume of wood} = 2\,a\,b\,h$$

$$= 2 \times 45\tfrac{1}{4}\,\text{in}\ 33\,\text{in}\ 4\tfrac{1}{2}\,\text{in}$$

$$= 13400\,\text{in}^3$$

$$\text{Work} = 9.8\,\frac{\text{in lb}}{\text{in}^3}\ 13400\,\text{in}^3$$

$$= 131000\ \text{in lb} = 10900\ \text{ft lb}$$

A 4000 lb weight falling 30 feet has an energy of

$$U = WH = 4000\,\text{lb}\ 30\,\text{ft} = 120000\ \text{ft lb}$$

The breaking of the wood panel thus accounts for less than 10 % of the energy to be accounted for. This can be neglected.

PROJECT NO.

The cask will have to be held by the steel sheet acting as a membrane. For a rough analysis of this, assume that the sheet bulges and that it acts like a belt whose width is about ½ the length of the box – ie the belt goes around the girth of the box – as in Fig Ex 16. The membrane force will come from

$$2 \times F_{mem} \times \frac{\delta}{\frac{1}{2}b} = F_p$$

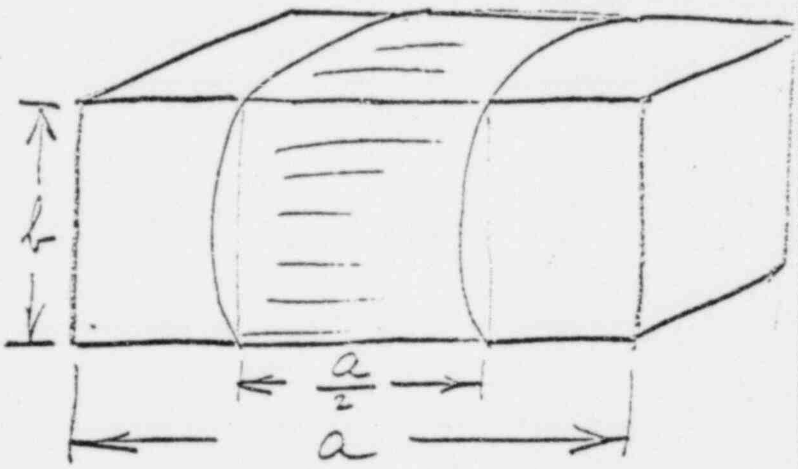


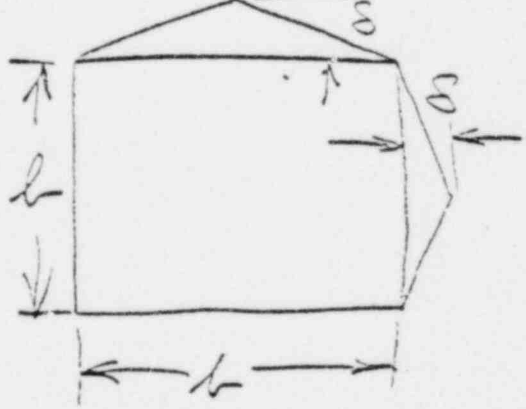
Fig Ex 16

or

$$\delta = \frac{F_p\, b}{4\, F_{mem}}$$

$$= \frac{F_p\, b}{4 \frac{a}{2} t\, \sigma_{mem}}$$

If we now say the sheet will yield at a constant stress, $\sigma_{mem} = S_y$, then



$$\delta = \frac{649\,000\ lb \cdot 33\ in}{2 \times 45\tfrac{1}{4}\ in\ \tfrac{1}{8}\ in\ 17500\ \frac{lb}{in^2}}$$

$$= 10.8\ in$$

Fig Ex 17

This is excessive.

W. K. BODGER

PROJECT NO.

The next question is — Is there enough energy to cause this deformation?

If the force $F_p = 649000$ lbs remains constant and acts through a distance δ, while deforming the sheet plastically, what will be the magnitude of δ?

$$F_p \delta = \frac{1}{2} W H \qquad \text{for one panel}$$

so

$$\delta = \frac{WH}{2F_p}$$

$$= \frac{4000 \text{ lb } 360 \text{ in}}{2 \times 649000 \text{ lb}}$$

$$= 1.11 \text{ inch}$$

This is a perfectly acceptable amount of bulging. Just to be safe, what is the strain?

$$\varepsilon = \frac{\sqrt{\left(\frac{b}{2}\right)^2 + \delta^2}}{\frac{b}{2}} - 1$$

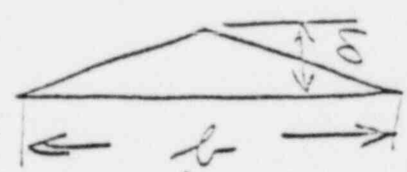


Fig Ex 18

$$= \sqrt{1 + \left(\frac{2\delta}{b}\right)^2} - 1$$

$$\cong 1 + \frac{1}{2}\left(\frac{2\delta}{b}\right)^2 - 1$$

$$= 1 + 2\left(\frac{\delta}{b}\right)^2 - 1 = 2\left(\frac{\delta}{b}\right)^2 = 2\left(\frac{1.11 \text{ in}}{33 \text{ in}}\right)^2 = \frac{2}{900} = 0.0022$$

This is OK.

PROJECT NO.

A better way to look at this is to say that the sheet is yielding plastically, and then to find the component of force that it can support in the direction of F_n. The integral of this force in the direction of yielding is the work done. This force is F_n in Fig Ex 18a:

Equilibrium gives

$$\frac{1}{2} F_n \cong \frac{\delta}{b/2} F_{mem}$$

or

$$F_n \cong \frac{4\delta}{b} F_{mem}$$

$$= \frac{4\delta}{b} \frac{a}{2} t S_y$$

$$= \frac{2 a t \delta}{b} S_y$$

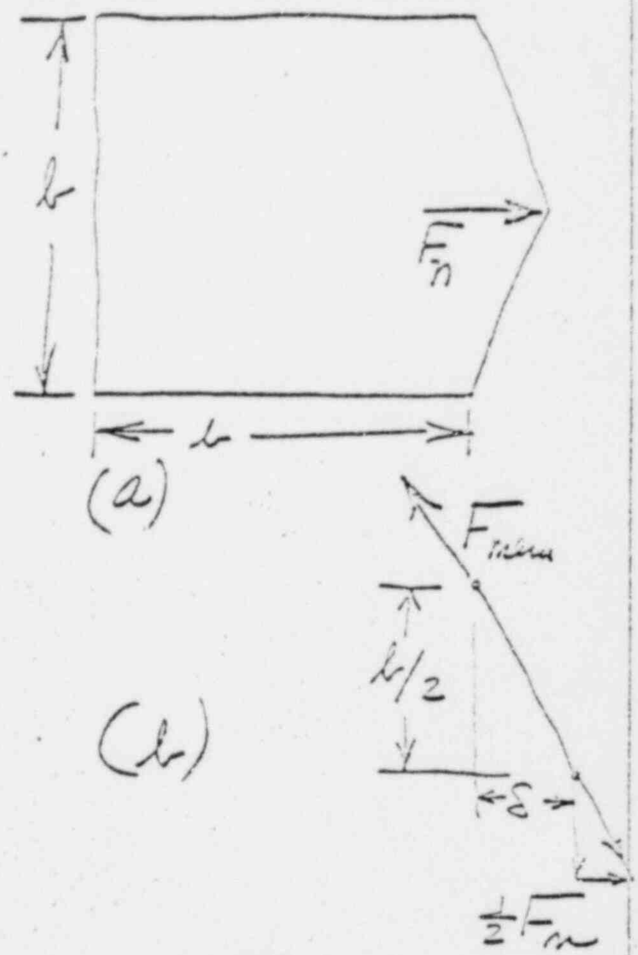


Fig Ex 18a

Total work done by F_n is

$$Work = \int F_n \, d\delta$$

$$= \frac{2 a t S_y}{b} \int_0^{\delta} \delta \, d\delta$$

$$= \frac{a t S_y \delta^2}{b}$$

or

$$\delta = \sqrt{\frac{b \times Work}{a t S_y}} = \sqrt{\frac{b \times U}{a t S_y}} = \sqrt{\frac{\frac{1}{2} W H b}{a t S_y}}$$

PROJECT NO.

$$\delta = \sqrt{\frac{4005\ \text{lb}\ 360\ \text{in}\ 36.5\ \text{in}}{48.7\ \text{in}\ 0.125\ \text{in}\ 175\,000\ \frac{\text{lb}}{\text{in}^2}}} = \sqrt{49.5}\ \text{in}$$

$$= 7.01\ \text{in}$$

This is a large deflection. Considering the conservative assumptions on which it is based, however, it is acceptable. Fracturing of the wood will absorb some energy, and probably more than half of the dimension "a" will be effective. Also there will be some work done in response to bowing measured the other way — ie interchanging "a" and "b" above. All in all, it seems likely that the maximum deflection will be about

$$\delta = 5\ \text{in}$$

The corresponding strain will be (see p 36)

$$\varepsilon = 2\left(\frac{\delta}{l}\right)^2 = 2\left(\frac{5}{33}\right)^2 = 0.046 = 4.6\ \%$$

This is OK.

The foregoing covers the welded-up steel box. However there still remains the question of attaching the cover to the base.

This can be done efficiently with the construction in Fig Ex 19. A steel angle is welded around the perimeter of the steel sheath on the underside of the base. A floating steel strip is welded around the perimeter of the bottom edge of the steel sheath on the outside of the cover.

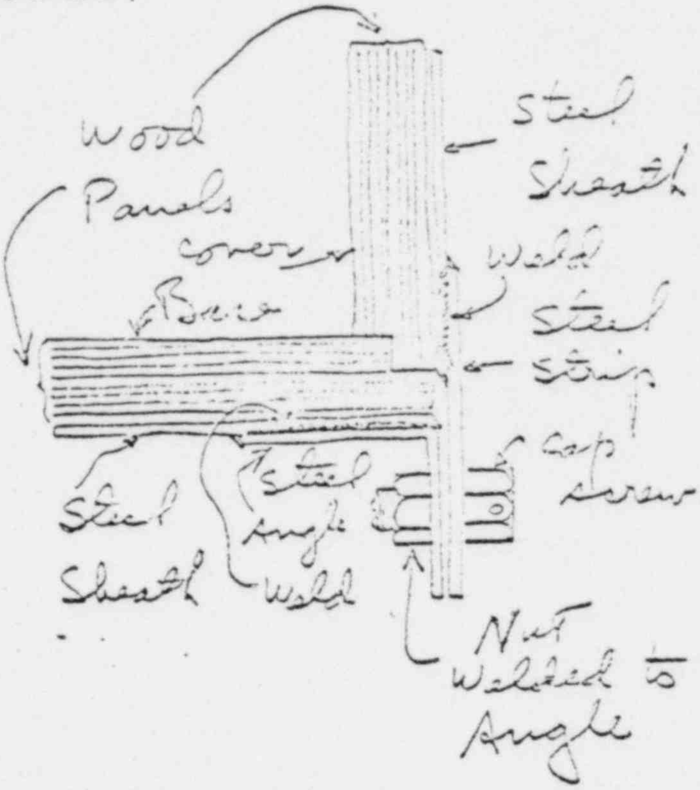


Fig Ex 19

When the cover is assembled to the base, the steel strip is bolted to the steel angle. The strip, the angle, and the bolts must carry the load. All will be of heat-treated high-strength steel.

PROJECT NO.

Each load, F_P, will necessarily be applied at approximately the center of a panel. As illustrated in Fig Ex 20 this means a shear of $\frac{1}{2}F_P$ and a tension of $\frac{1}{2}F_P$ simultaneously, to be carried by a line of bolts.

If there are N bolts along one side, then the load per bolt will be

$$_SF_B = \frac{\frac{1}{2}F_P}{N} = \frac{F_P}{2N} \text{ shear}$$

and

$$_TF_B = \frac{\frac{1}{2}F_P}{N} = \frac{F_P}{2N} \text{ tension}$$

carried simultaneously.

The cap screws should be as in Fig Ex 21 — i e the shank should be the full nominal diameter, not relieved. Then load-carrying areas are

Shear $A_S = \frac{\pi}{4}d^2$

Tension $A_T = \frac{\pi}{4}d_\sigma^2$

where d_σ = effective tensile stress diam.

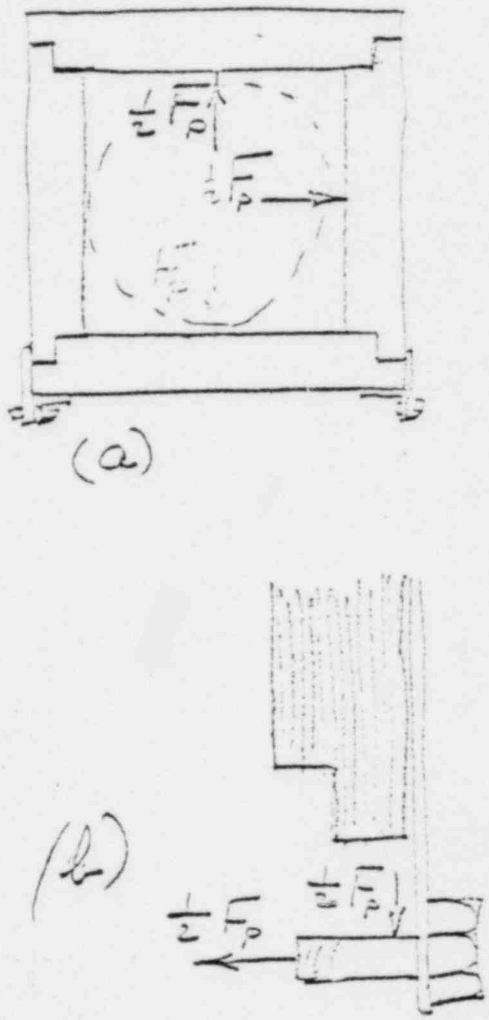


Fig. Ex 20

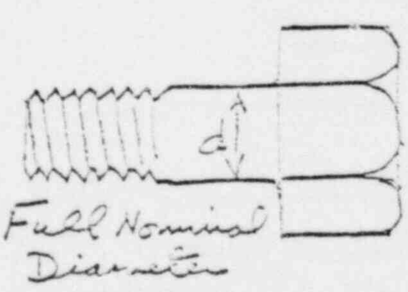


Fig Ex 21

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At the thread, the stress is purely tensile -

$${}_{th}\sigma_B = \frac{{}_T F_B}{A_T} = \frac{F_P}{2 N A_T}$$

At the shank, there is both a shear stress

$${}_{sh}\tau_B = \frac{{}_s F_B}{A_s} = \frac{F_P}{2 N A_s}$$

and a tensile stress

$${}_{sh}\sigma_B = \frac{{}_T F_B}{A_s} = \frac{F_P}{2 N A_s}$$

The two stresses at the shank will add in Mohr's Circle, Fig Ex 22.

The max shear is

$$\tau_{max} = \sqrt{{}_{sh}\tau_B^{\,2} + \left(\tfrac{1}{2}\,{}_{sh}\sigma_B\right)^2}$$

$$= {}_{sh}\tau_B \sqrt{1 + \left(\tfrac{1}{2}\right)^2}$$

$$= 1.118\ {}_{sh}\tau_B$$

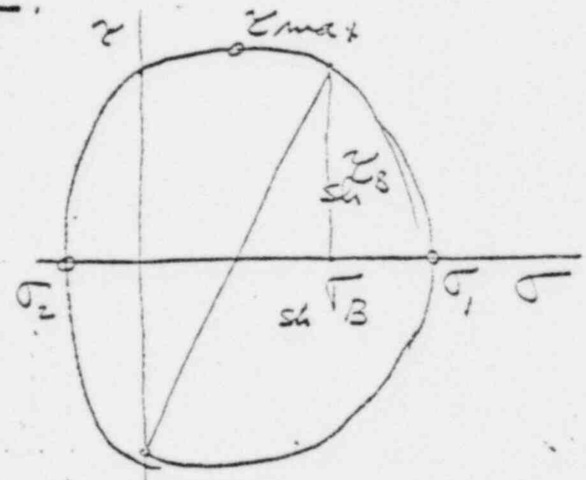

Fig Ex 22

The larger principal stress will be

$$\sigma_1 = \tfrac{1}{2}\,{}_{sh}\sigma_B + \tau_{max}$$

$$= 1.618\ {}_{sh}\tau_B$$

PROJECT NO.

We can assume limit conditions — ie failure — to be any of the following

$$\left.\begin{aligned} {}_{th}\sigma_B &= S_{ult} = 201\,000 \text{ psi} \\ \sigma_1 &= S_{ult} = 201\,000 \text{ psi} \\ \tau_{max} &= \tfrac{1}{2} S_{ult} = 100\,500 \text{ psi} \end{aligned}\right\} \text{see p (32)}$$

3-3

Rearranging the equations on the preceding pages, we can solve for the required size or number of bolts. Also, all threads should be fine, ie UNFC, in order to keep the ratio d_r/d, and hence A/A_s, as large as possible.

∴ the number of bolts may be determined by any of ${}_{th}\sigma_B$ or σ_1 or τ_{max}.

For σ_1

$$N = \frac{F_p}{2A_s\,{}_{sh}\tau_B} = \frac{F_p}{2\frac{\pi}{4}d^2\sigma_1/1.618} = \frac{3.236}{\pi}\,\frac{F_p}{S_{ult}\,d^2}$$

$$= 1.03\,\frac{F_p}{S_{ult}\,d^2} = \frac{1.03 \times 649\,000 \text{ lb}}{201\,000 \frac{\text{lb}}{\text{in}^2}\,d^2} = \frac{3.32 \text{ in}^2}{d^2}$$

For τ_{max}

$$N_{\tau max} = \frac{F_p}{2A_s\,{}_{sh}\tau_B} = \frac{F_p}{2\frac{\pi}{4}d^2\tau_{max}/1.118} = \frac{2.236}{\pi}\,\frac{F_p}{\frac{1}{2}S_{ult}\,d^2}$$

$$= \frac{4.472}{\pi}\,\frac{F_p}{S_{ult}\,d^2} = \frac{4.472 \times 649\,000 \text{ lb}}{\pi\, 201\,000 \frac{\text{lb}}{\text{in}^2}\,d^2} = \frac{4.59 \text{ in}^2}{d^2}$$

For ${}_{th}\sigma_B$

$$N = \frac{F_p}{2A_T\,{}_{th}\sigma_B} = \frac{F_p}{2\frac{\pi}{4}d_r^2\,{}_{th}\sigma_B} = \frac{2}{\pi}\,\frac{F_p}{S_{ult}\,d_r^2} = \frac{2 \times 649\,000 \text{ lb}}{\pi\, 201\,000 \frac{\text{lb}}{\text{in}^2}\,d_r^2}$$

$$= \frac{2.15 \text{ in}^2}{d_r^2}$$

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d	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	in
d_r	.451	.571	.689	.805	.919	1.04	1.17	in
d^2	.250	.390	.562	.765	1.000	1.26	1.56	in^2
d_r^2	.203	.326	.474	.648	.844	1.08	1.36	in^2
$(N)_{\tau_{max}}$	18.4	11.8	8.16	6.00	4.59	3.64	2.94	–
$(N)_{\sigma_{TE}}$	10.6	6.60	4.54	3.32	2.55	1.99	1.58	–
Use N	19	12	9	6	5	4	3	–

The last row above is the number of bolts to use, per side, if bolt strength is the design criterion. Clearly shear of the bolt is the failure criterion, as far as bolt design is concerned.

There is also a question of tearing out a piece of the apron, or steel strip along the bottom edge of the cover.

In Fig Ex 23, we can take the width of the "tear out" piece, below the hole, to be "d" wide (ie nominal hole diameter) by "e" high, from the edge of the apron to the near-side tangent of the hole.

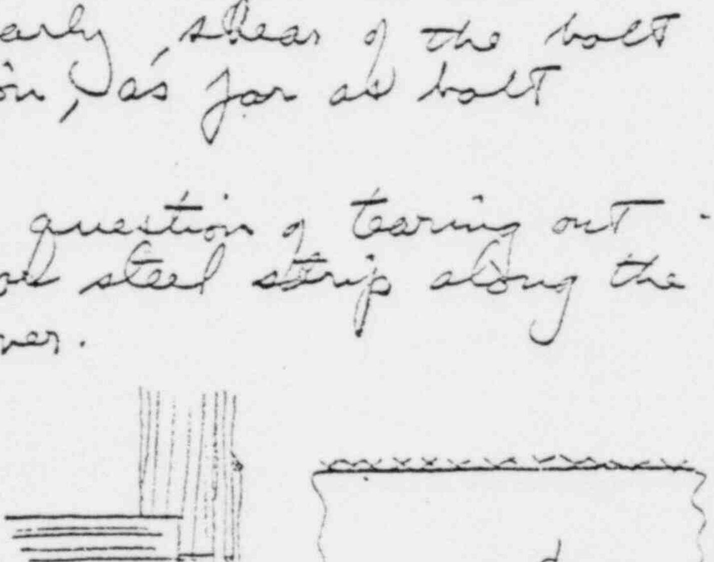

Fig Ex 23

Then

$$\tau = \frac{\frac{1}{2} F_p}{N\, 2 e t}$$

or

$$e = \frac{\frac{1}{2} F_p}{2 N \tau t}$$

Using the failure criterion of p (40), we can say

$$e = \frac{\frac{1}{2} F_p}{2N \frac{1}{2} S_{ucr}\, t} = \frac{F_p}{2N S_{ucr}\, t}.$$

In addition, there is a lateral, or bending, load on the apron, as in Fig Ex 24. At any one bolt, the load is (see p (38))

$$_T F_B = \frac{F_p}{2N}$$

where, as before, there are N bolts along one side. The apron must be strong enough that the bolt head can not pull through.

Fig Ex 24

For simplicity, consider the loading to be a punch-type shear load along the circle inscribed in the hexagonal bolt head. Then

$$\tau = \frac{\frac{1}{2} F_p}{N \pi d_h t}$$

or

$$t = \frac{F_p}{2\pi N d_h \tau} = \frac{F_p}{2\pi N d_h \frac{1}{2} S_{ucr}} = \frac{F_p}{\pi N d_h S_{ucr}}$$

PROJECT NO.

Numerically

$$t = \frac{649\,000\ \mathrm{lb}}{\pi N d_h\, 201\,000 \frac{\mathrm{lb}}{\mathrm{in}^2}} = \frac{1.03\ \mathrm{in}^2}{N d_h}$$

d	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	in
d_h	$\frac{3}{4}$	15/16	$1\frac{1}{8}$	$1\frac{5}{16}$	$1\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{7}{8}$	in
N	19	12	9	6	5	4	3	–
t	0.072	0.0915	0.102	0.131	0.138	0.153	0.183	in

For the "tear-out" at the bottom of the apron (p 42)

$$e = \frac{F_p}{2 N t S_{wet}} = \frac{649\,000\ \mathrm{lb}}{2 N t\, 201\,000 \frac{\mathrm{lb}}{\mathrm{in}^2}} = \frac{1.61\ \mathrm{in}^2}{N t}$$

d	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	in
N	19	12	9	6	5	4	3	–
et	0.0847	0.134	0.179	0.268	0.322	0.402	0.536	in^2
$t=\frac{1}{8}$, e	0.678	1.072	1.43	2.14	2.58	3.22	4.30	in
$\frac{3}{16}$	0.452	0.715	0.955	1.43	1.72	2.24	2.86	in
$\frac{1}{4}$	0.339	0.536	0.717	1.072	1.29	1.61	2.15	in

Both "t" and "e" will apply equally to the apron and the angle section on the underside of the base panel; the "tear-out" failure can not occur on the angle, however. The nuts (Fig Ex 19 p 37 or Fig Ex 23 p 41) should be welded to the back side of the angle, as near to the horizontal leg as possible, but still keeping the nut axis straight.

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Ex

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The base of the container is mounted on a pair of 4 in x 6 in skids. These skids should give adequate clearance for the fork of a fork-lift truck to go under the base to lift the assembly. This clearance will be the basic 6 inches, less the total drop of the apron and/or the angle section. Assuming that the fillet will prevent the nut from being closer than 1/4 inch from the horizontal member of the angle, the usable clearance will be

$$\text{clearance} = 6\text{ in} - \tfrac{1}{2}d_h - \tfrac{1}{2}d - e - \tfrac{1}{4}\text{ in}$$
$$= 5.75\text{ in} - \left[e + \frac{d+d_h}{2}\right]$$

From p 43

	d	$\frac{1}{2}$	$\frac{5}{8}$	$3/4$	$7/8$	1	$1\frac{1}{8}$	$1\frac{1}{4}$ in
	d_h	$3/4$	$15/16$	$1\frac{1}{8}$	$1\frac{5}{16}$	$1\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{7}{8}$ in
	cl + e	5.125	4.97	4.81	4.65	4.50	4.54	4.18 in
$t=\frac{1}{8}$,	e	0.678	1.072	1.43	2.14	2.58	3.22	4.30 in
	cl	4.54	3.90	3.38	2.51	1.92	1.12	— in
$\frac{3}{16}$	e	0.452	0.715	0.955	1.43	1.72	2.24	2.86 in
	cl	4.67	4.22	3.85	3.22	2.78	2.10	1.32 in
$\frac{1}{4}$	e	0.339	0.536	0.717	1.072	1.29	1.61	2.15 in
	cl	4.78	4.43	4.09 ✓	3.52	3.21	2.73	2.03 in

From the mechanical design point of view, any of the foregoing combinations is OK. The need for fork-lift clearance puts an upper limit on d. Assembly & disassembly time puts a lower limit on d.

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All welds should be full fillet welds, at both edges, as in Fig Ex 25 — i.e. a double-filleted lap joint.

The stress in the weld can be taken as

$$\tau = \frac{\frac{1}{2} F_p}{2 \times \frac{1}{\sqrt{2}} t b}$$

$$= \frac{\sqrt{2}}{4} \frac{F_p}{t b}$$

$$= \frac{\sqrt{2}}{4} \frac{649000 \text{ lb}}{36\frac{1}{2} \text{ in } t}$$

$$= \frac{6290 \text{ lb/in}}{t}$$

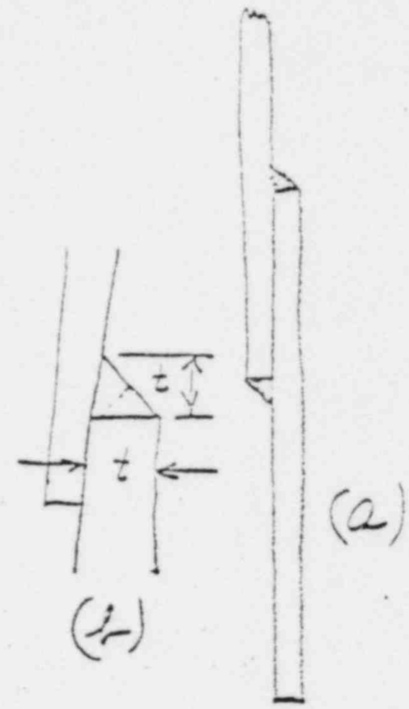

Fig Ex 25

$t = \frac{1}{8} \quad \frac{3}{16} \quad \frac{1}{4}$ in

$\tau = 50300 \quad 33500 \quad 25100$ psi

Since the apron, in each case, is attaching to a $\frac{1}{8}$ in thick box, the $t = \frac{1}{8}$ inch case applies most directly. With a 50% weld efficiency this is still OK.

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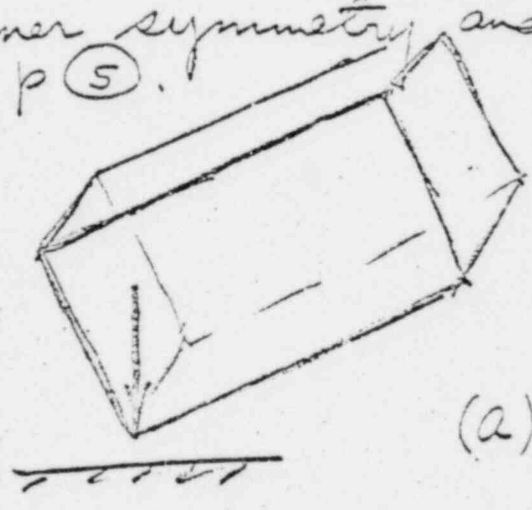
C. Corner Contact

The ideal case, or the simple one to analyze, is the symmetric case - ie each of the 3 panels involved makes an equal angle with the vertical; or gravity direction. Any case between this and (1.) symmetric edge contact will be intermediate in intensity, as the impacting force can be resolved into two components, one with corner symmetry and one with edge symmetry as on p(6), or (2.) side contact will be intermediate in intensity as the impacting force can be resolved into two components, one with corner symmetry and one with side contact as on p(5).

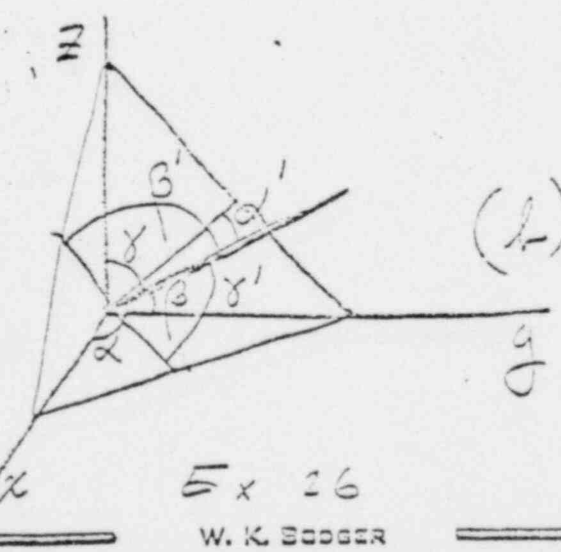


The ideal corner contact is sketched in Fig Ex 26(a). The relationship of the striking force to the corner axes is in Fig Ex 26 (b).

A well known relation in mathematics is

$$\cos^2\alpha + \cos^2\beta + \cos^2\gamma = 1$$

which comes from



Ex 26

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$$a^2 + b^2 = (d')^2 = (\text{diagonal on a plane})^2$$

and $(d')^2 + c^2 = d^2$

or $a^2 + b^2 + c^2 = d^2$

or
$$a = d \cos\alpha$$
$$b = d \cos\beta$$
$$c = d \cos\gamma$$

so $d^2\cos^2\alpha + d^2\cos^2\beta + d^2\cos^2\gamma = d^2$

or $\cos^2\alpha + \cos^2\beta + \cos^2\gamma = 1$

Since from symmetry we have

$$\alpha = \beta = \gamma$$

then $3\cos^2\alpha = 1$

or $\cos^2\alpha = \frac{1}{3}$

or $\cos\alpha = \frac{1}{\sqrt{3}} = 0.577$

or $\alpha = \beta = \gamma = \cos^{-1} 0.577 = 54.75°$

As seen in Fig Ex 26 (b)

$$\alpha + \alpha' = \beta + \beta' = \gamma + \gamma' = \frac{\pi}{2} = 90°$$

This is always true – with or without symmetry.

So $\alpha' = \beta' = \gamma' = 90° - 54.75° = 35.25°$

for our symmetric case.

PROJECT NO.

The impact force will be resolved into 3 equal components; one of these is shown in Fig Ex 27.

The angle between the vertical (ie the total impact force) and one plane is α'. Hence the angle between the impact force F and the component ${}_cF_p$ is α. Thus

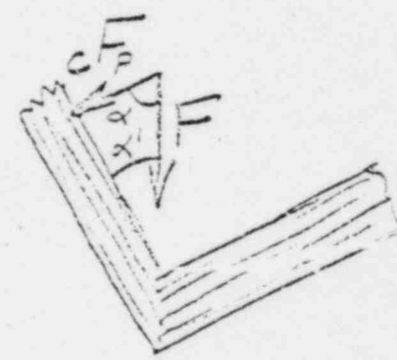

Fig Ex 27

$$ {}_cF_p = F\cos\alpha $$

and

$$ ({}_cF_p)_\alpha \cos\alpha + ({}_cF_p)_\beta \cos\beta + ({}_cF_p)_\gamma \cos\gamma $$

$$ = F\cos^2\alpha + F\cos^2\beta + F\cos^2\gamma = F $$

The important result is

$$ {}_cF_p = F\cos\alpha = \frac{F}{\sqrt{3}} $$

This is only $\sqrt{\frac{2}{3}} = 0.816 = 81.6\%$ as large as F_p for the edge contact case (see p(6)).

Since

$$ {}_cF_p < F_p $$

The stresses, etc, in the corner contact case are smaller than those in the edge contact case and they need not be considered further here.

3-4

PROJECT NO.

3. Fire Resistance

The original exoskeleton design had an exposed, essentially separate, frame. As the design has evolved, the frame is no longer separate, although still exposed.

The fire resistance of the present design will be at least equal, in all respects, to the original design. The completely welded-up box will eliminate any possibility that the frame members will warp and lose their ability to perform their function.

The steel sheathing over the wood panels will still offer some fire resistance to the wood. The welded-up design will probably perform this function a little better than the loose-piece design; data on this [illegible] are not readily available, however. The [illegible] rely bolted junction of the co[illegible] [illegible] the base will [illegible]event direct access of the flames to the wood panels; however charring will still occur, of course.

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