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A NAVAL WEAPONS HANDLING LABORATORY

TECHNICAL REPORT

DYNAMIC AND STRUCTURAL ANALYSIS OF SHIPPING AND STORAGE CONTAINER FOR ENCAPSULATED HARPOON (RGM-84A)

DEPARTMENT OF THE NAVY
SEA SYSTEMS COMMAND

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(6) DYNAMIC AND STRUCTURAL ANALYSIS

OF

REUSABLE SHIPPING AND STORAGE CONTAINER

FOR

ENCAPSULATED HARPOON MISSILE (RGM-84A)

Abstract

A dynamic and structural analysis of a proposed reusable Shipping and Storage Container for the Encapsulated HARPOON Missile was made by the Naval Weapons Handling Laboratory as part of the design study for such container.

The subject design incorporates a freebreathing fiberglass pod containing the encapsulated weapon. The pod and weapon are suspended from a truss-like outer structure by elastomeric mounts configured in a laterally focalized fashion.

The analysis generates isolator parameters which attenuate the handling and transportation shock and vibration environment to safe levels for the weapon and verifies that the structural design concept can sustain the resulting loads.

Details of the analysis are presented herein.

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ILLUSTRATIONS

1. INTRODUCTION

1.1 The analysis which follows is based upon Specification XAS-3894B, dated 18 December 1974, for a reusable shipping and storage container for the Encapsulated HARPOON Missile (RGM-84A).

1.2 This version of the HARPOON has a support-attenuator (sabot) at Capsule Station 62 which greatly reduces the bending of the HARPOON when the Capsule/HARPOON is subjected to shock inputs. (See Figure 1). As a result, the resilient suspension system in the reusable container can be stiffened to a maximum of 12.3 Hz in vertical translational natural frequency.

1.3 A focused suspension system is proposed. The Capsule is supported at Capsule Stations 54 and 172 in cradles which are carried on two longitudinal beams. A free-breathing fiberglass shell encloses the capsule and cradles: it consists of a lower shell and upper cover closed by 1/4-turn fasteners. The beams are supported on focused resilient elements, symmetrical in the plan view about the combined C.G. of the assembly. The compression axes of the mounts at each mount station are focused so as to project the elastic center (EC) to a longitudinal axis through the center of gravity (CG) of the resiliently supported load. This concept so positions the mountings that it minimizes the width and height of the container assembly. It also results in a suspension that should decouple the modes: a translational lateral input should result in a translational response. Along the longitudinal axis the suspension provides a lower spring rate in shear to minimize loads from end impact: the links which restrain the HARPOON within the Capsule require this. Note that cradles are not symmetrical about the C.G. and so are not directly over the mount clusters.

2. PARAMETERS - DYNAMIC ANALYSIS

2.1 Vibration. Transmissibility of the resilient suspension system shall be four maximum in the vertical and axial directions of the container. XAS-3894B requires the natural frequency to be 7 to 10 Hz in the vertical mode. Measured in transverse direction, the missile response to conform to the maximum response as depicted on Figure 2 of XAS-3894B, when subjected to the vibration inputs of Table I of XAS-2381 (as modified). Natural frequency in the transverse direction to be less than 18 Hz.

2.1.1 Vibration Fatigue Test. To be conducted per para. 10.2.5.1 of OR-11. Input: cycled from 5 to 60 Hz in 15 minutes with a logarithmic sweep rate: input acceleration amplitude shall be a consistent 1 g. Resonant dwell at 1 g input for 15 minutes. Transmissibility shall be 4 or less: response per Fig. 2 of XAS-3894B.

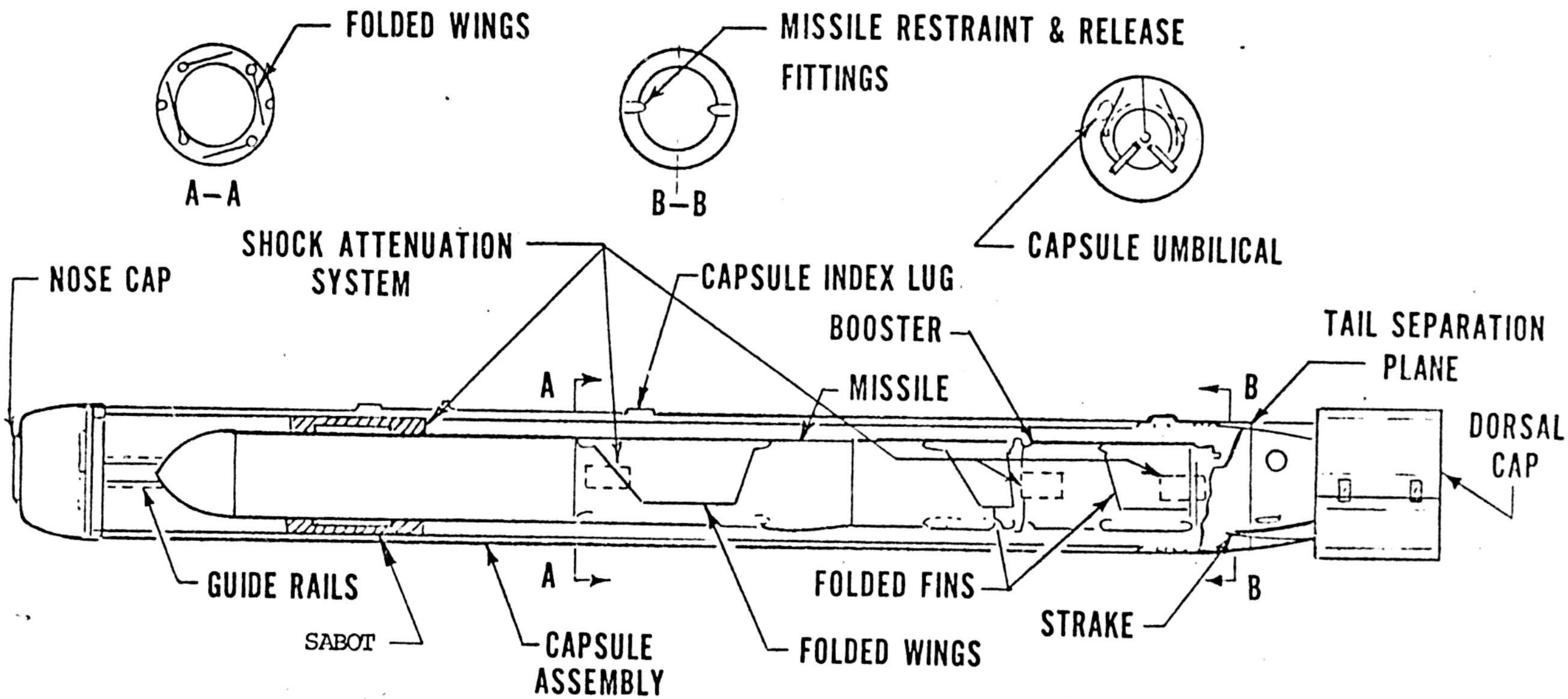
2.2 Shock Response. The undamped shock spectra of the missile response pulses in the three major axes are limited to the envelope depicted in Figure 1 of XAS-3894B, per para. 4.2.2.11.2, when the loaded container is subjected to the following shock attenuation tests:

2.2.1 Vertical Shock Test. Free fall flat drop from 18 inch height performed at 50° - 90°F.

2.2.2 Rotational Drop Test. Four (4) rotational corner drops from a height of 18 inches at temperatures of -20° ± 5°F., 50°-90°F., and 140° ± 5°F., for a total of 12 drops: 3 at each corner.

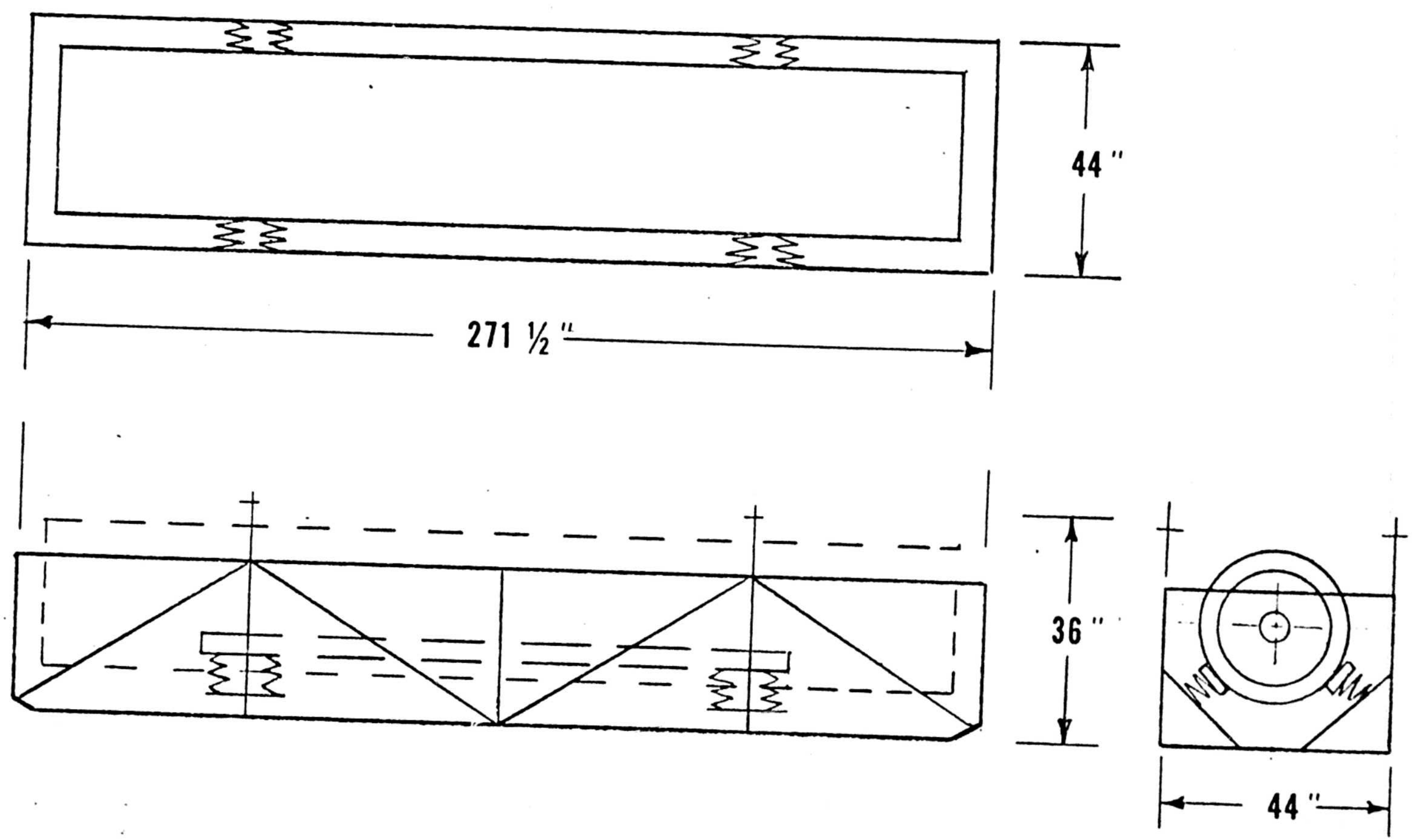
2.2.3 Impact Test. Incline-impact test on each end of the container with 10 f.p.s. velocity at temperatures of -20° ± 5°F., 50°-90°F., and 140° ± 5°F. Limit load on missile is 24,000 lbs. (or 16 g's) at -20°F.

2.2.4 Repetitive Shock Test. Method 5019 of FTMS 101B as modified by para. 10.2.2.3 of OR-11 at 50°-90°F. temperature.



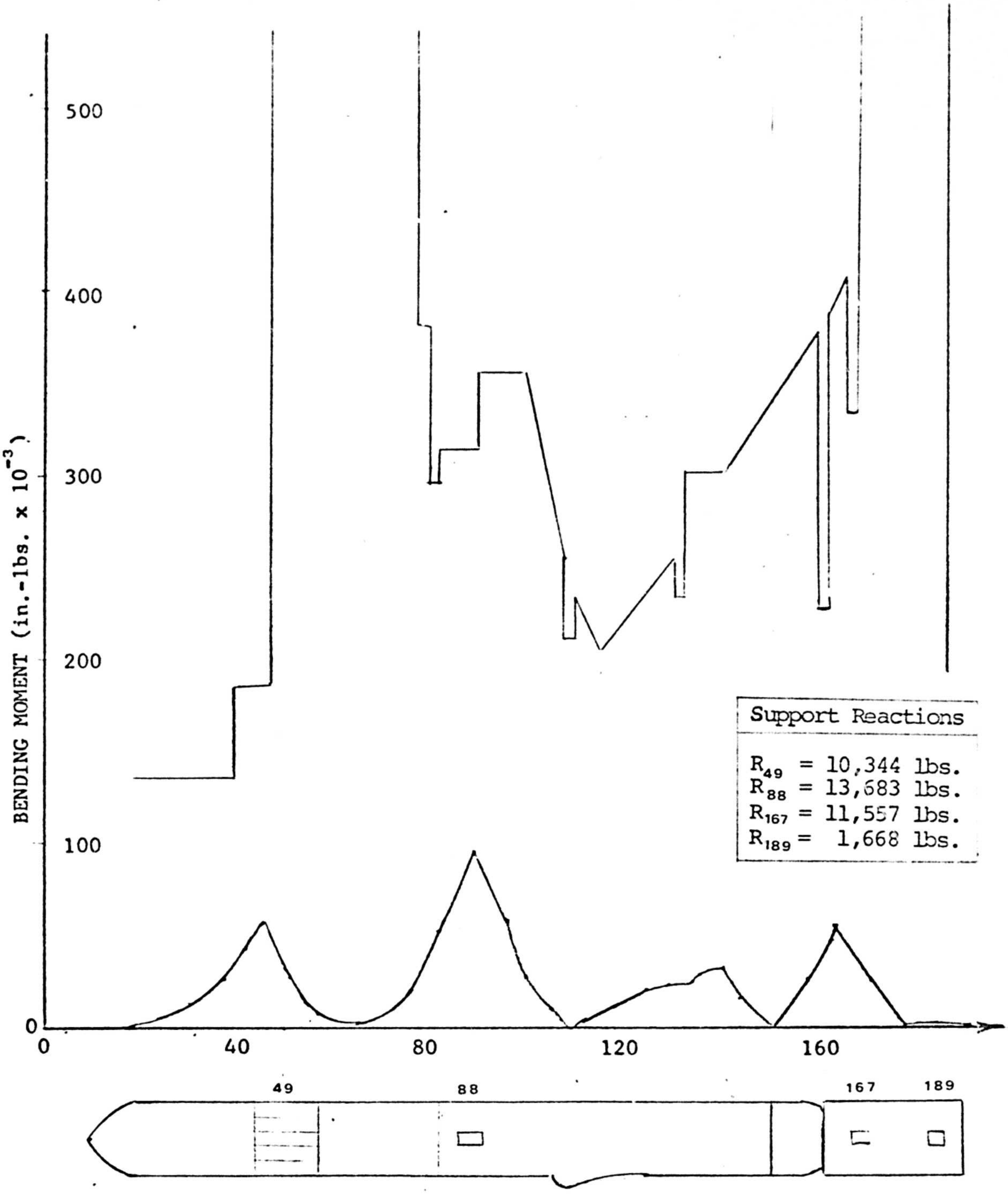
CAPSULE SUBSYSTEM CONFIGURATION

FIGURE 1



ENCAPSULATED HARPOON CONTAINER CONCEPT

FIGURE 2



BENDING MOMENTS AND REACTION LOADS AT SUPPORT POINTS FOR 18 INCH FLAT DROP

Calculated for Suspension System with f_n (Vertical Trans) = 12.3 Hz

Encapsulated Harpoon

Figure 3

3. SUMMARY

3.1 Summary of Dynamic Analysis. The resilient suspension of the mass weight of the capsule/missile assembly, protective shell, cradles and beams consists of a four-point spring (shear mount) system) symmetrical about the combined C.G. of the assembly. Two mounts are arranged at each support point to focus their elastic center (EC) to a point on the longitudinal axis through the CG of the assembly. This should effectively decouple the system in the presence of transverse excitation: also, the natural frequency in transverse direction should be about equal to the natural frequency in vertical translation. The capsule is secured in the two cradles by means of clamps tightened about the capsule at strong points defined by Table I of XAS-3894B. In the longitudinal direction the elastomeric shear mounts are loaded primarily in shear, which should keep forces along that axis well under the 24,000 pound (16G) limit established for the links which restrain the HARPOON within the Capsule.

3.1.1 Suspension System Natural Frequency. Estabilished as 10.5 Hz in vertical translation based upon a calculated maximum of 12.3 Hz which would keep the undamped shock spectrum of the resulting shock pulse within the envelope of Fig. 1 of XAS-3894B. The 10.5 Hz was chosen to provide a margin for low temperature stiffening of the resilient mounts. Note that the HARPOON is supported within the Capsule at CS63 and this additional support allows the use of higher suspension system natural frequency than allowed by XAS-3894B, Para. 3.2.1.1.

3.1.2 Resilient System Spring Rate. The dynamic vertical spring rate of the focused system at +70^{o}F. is K_v = 33,280 pounds/inch which resolves into static shear spring rate per mount = k_s = 800 lbs./inch.

3.1.2.1 Resilient Suspension Elements. The design of the bonded elastomer/metal shock and vibration attenuating mounts is covered in para. 6.1 and shown on Figure 8. Its design is based upon an elastomer having good low-temperature and damping properties expected to meet the requirements of XAS-3894B.

3.1.3 Acceleration Response. For free-fall and rotational cornerwise drops the resulting undamped shock spectrum should be within the allowed limits. For 10 fps end impact the resulting longitudinal force will be well under the established limit. Anticipated accelerations are shown on the calculation sheets.

3.1.4 Mount Station Spacing. Longitudinal half-spread of mount cluster stations to CG is 57.87 inches.

3.1.5 End Deflections. Maximum deflection of extreme end of the Capsule is 2.3 inches at +140^{o}F. in 18 inch rotational drop. The end impact results in longitudinal deflection of 3.8 inches maximum at +140^{o}F.

4. DYNAMIC ANALYSIS

4.1 SYMBOLS USED IN ANALYSIS:

(1)	A	Maximum Vertical Acceleration @ CG	in./sec.^{2}
(2)	B	Maximum Vertical Acceleration at other than CG due to rotation	in./sec.^{2}
(3)	b	Longitudinal distance to mounts from CG	inches
(4)	c	Distance to outer fiber from neutral axis	inches
(5)	CG	Center of Gravity of resiliently suspended equipment	
(6)	CS	Capsule Station	inches
(7)	D	Correction factor for energy losses due to damping	
(8)	d	Deflection	inches
(9)	d_1	Maximum vertical deflection at CG	inches
(10)	d_2	Maximum vertical deflection at end of unit due to rotation	inches
(11)	[illegible]	Dynamic deflection at resilient mountings	inches
(12)	d_r	Deflection; rebound	inches
(13)	d_{st}	Static deflection	inches
(14)	d_t	Dynamic Deflection - total at end of unit	inches
(15)	E	Modulus of Elasticity	lbs/in.^{2}
(16)	EC	Elastic Center of suspension system	
(17)	e	Vertical distance from longitudinal axis through CG to focal point of mounts' compression axes	inches
(18)	F.S.	Factor of Safety	
(19)	fn	Natural Frequency	Hz
(20)	f_{st}	Coefficient of friction	

4.1 SYMBOLS USED IN ANALYSIS - Continued

(21)	G	Unit of Gravity	
(22)	g	Acceleration of Gravity	386 in./sec^2
(23)	G_1	Maximum Vertical Acceleration @ CG @ CS124	Gravity Units
(24)	G_{ha}	Maximum Vertical Acceleration at HARPOON aft end at CS-205.4	Gravity Units
(25)	G_{hf}	Maximum Vertical Acceleration at HARPOON forward end at CS-25	Gravity Units
(26)	G_{ca}	Maximum Vertical Acceleration at Capsule aft end at CS-246	Gravity Units
(27)	G_{cf}	Maximum Vertical Acceleration at Capsule forward end	Gravity Units
(28)	G_r	Maximum Vertical Acceleration at C.G. due rebound	Gravity Units
(28)	G_T	Total Acceleration on Equipment due to Translation and Rotation	Gravity Units
(30)	h	Height of Drop	inches
(31)	h_1	Height of Container Pivot Point above Floor	inches
(32)	I	Moment of Inertia of Section	in.^{4}
(33)	I_{CG}	Moment of Inertia about CG	lb-in-sec^2
(34)	I_o	Moment of Inertia about Pivot Point "O"	lb-in-sec^2
(35)	J	Correction factor to account for damping in mounts	
(36)	K_V	System Dynamic Vertical Spring Rate	lbs./inch
(37)	K_H	System Dynamic Horizontal Spring Rate	lbs./inch
(38)	K_R	System Dynamic Rotational Spring Rate	
(39)	k_d	Single Mount Dynamic Shear Spring Rate	lbs./inch
(40)	k_s	Single Mount Static Shear Spring Rate	lbs./inch
(41)	KE	Kinetic Energy	in.-lbs.
(42)	L_o	Length of Container, overall	inches
(43)	L_1	Distance from CG to end of Capsule	inches
(44)	L_2	Distance from CG to end of HARPOON	inches

(45)	L'	Ratio $\frac{\text{compression k}}{\text{shear k}}$ of mountings	
(46)	ℓ	Unit length	inches
(47)	M	Moment	in-lbs.
(48)	m	Mass of suspended unit	lb-sec^2/inch
(49)	N	Clamping Load	lbs.
(50)	P	Load	lbs.
(51)	PE	Potential energy	inch-lbs.
(52)	P	Lateral horizontal distance from vertical plane through CG to mount's EC	inches
(53)	R	Distance from container pivot point "O" to CG	inches
(54)	R_A	Reaction load at point A	lbs.
(55)	R_B	Reaction load at point B	lbs.
(56)	r	Radius of gyration of suspended mass	inches
(57)	S	Stress	psi
(58)	T	Transmissibility	
(59)	v	Impact velocity	in./sec.
(60)	W	Weight of suspended equipment	lbs.
(61)	W_1	Weight of outer frame	lbs.
(62)	w	Weight per unit of length	lbs./inch
(63)	X	Horizontal distance from container pivot point "O" to unit CG	inches
(64)	X_f	Horizontal distance from CG to forward end of container assembly	inches
(65)	X_a	Horizontal distance from CG to aft end of container assembly	inches
(66)	Y	Vertical distance from container base to unit's CG (See Figure on p13)	inches
(67)	Z	Length of suspended unit	inches

4.1 SYMBOLS USED IN ANALYSIS - continued

(68)	θ_1	Angle between a line joining the CG and pivot point "O" and a vertical plane before drop (Outer end of unit raised "h" inches)	degrees
(69)	θ_2	Angle between a line joining the CG and pivot point "O" and a vertical after drop. (h = 0)	degrees
(70)	ω_0	Angular velocity of mounted unit about pivot point "O"	Radians/sec
(71)	ω_1	Vertical Translational Circular Frequency	Radians/sec
(72)	ω_2	Rotational Circular Frequency	Radians/sec
(73)	α	Angle between focused mounts' compression axis and horizontal	degrees
(74)	β	Angle between focused mounts' compression axis and vertical	degrees
(75)	ζ	% Critical damping (C/C_c)	
(76)	δ	Logarithmic decrement	

FACTORS USED IN ANALYSIS

$$\frac{\text{DYNAMIC SPRING RATE}}{\text{STATIC SPRING RATE}} = 1.3$$

$$\frac{\text{MOUNT SPRING RATE AT } -20^{\circ}\text{F.}}{\text{MOUNT SPRING RATE AT } +70^{\circ}\text{F.}} = 1.25$$

$$\frac{\text{MOUNT SPRING RATE AT } +140^{\circ}\text{F.}}{\text{MOUNT SPRING RATE AT } +70^{\circ}\text{F.}} = 0.9$$

Factors based upon data from mount vendors.

4.2 SUSPENDED MASS DATA

	I_{CG} In.-Lb.-Sec.^{2}	Weight Pounds
SHELL	3709.5	276
CRADLES	505	54
BEAMS	2319	386
MOUNTS (½ of weight of)	278	32
FASTENERS & GASKET	237.5	20
MISSILE	10240.2	1494.7
CAPSULE	10551	685
	27,840.2	2,948.7

5.0 DETERMINATION OF SUSPENSION SPRING RATES

Assume 10.5 Hz = f_n in Vertical Translational Mode

$K_v = W(f_n/3.13)^2 = 2{,}948(10.5/3.13)^2 = 33{,}175$ lbs/inch

Using 8 shear mounts with compression axis focused at 45° from vertical (β)

$K_v = 8k_d(L' \cos^2\beta + \sin^2\beta)$ and $L' = 7$

or $8\ k_d\ (4) = 33{,}175$

$k_d = 1036$ lb/in

$k_s = k_d/1.3 = 797$ lb/in

Using $k_s = 800$ lb/in for + 70°F.

+70°F.	$K_v = 8(800)(1.3)(4) = 33{,}280$ lb/in
-20°F.	Using stiffening factor of 1.25, $K_v = 41{,}600$ lb/in
+140°F.	Using stiffness reduction factor of 0.9, $K_v = 29{,}952$ lb/in

$K_H = 800(8)(1.3) = 8{,}320$ lbs./inch (Longitudinal)

5.1 18 INCH FLAT BOTTOM DROP

$+70^{\circ}F.$

Energy to be stored = K.E. = P.E. = wh

Wh = 2948 (18) = 53,064 in-lbs.

$d^2 = 2KE/K_v = 2(53,064)/33,280 = 3.19$

d = 1.78 inches

$G_1 = dK_v/W = 1.78\ (33,280)/2948$

$G_1 = 20$ G's

5.2 END IMPACT at 10 ft/sec

$d^2 = Wv^2/8k_d g$ and $G = dK_d/W$

$-20^{\circ}F.$ $d^2 = 2948(120)^2/8(800)(1.3)(1.25)(386) = 10.57$

d = 3.2 inches

$G_1 = 3.2(8)(800)(1.3)(1.25)/2948$

$G_1 = 11.3$ G's vs 16G allowed

$+70^{\circ}F.$ d = 3.6 in. & $G_1 = 10.1$ G's

$+140^{\circ}F.$ $d^2 = 2948(120)^2/8(800)(1.3)(.9)(386) = 14.68$

d = 3.8 inches

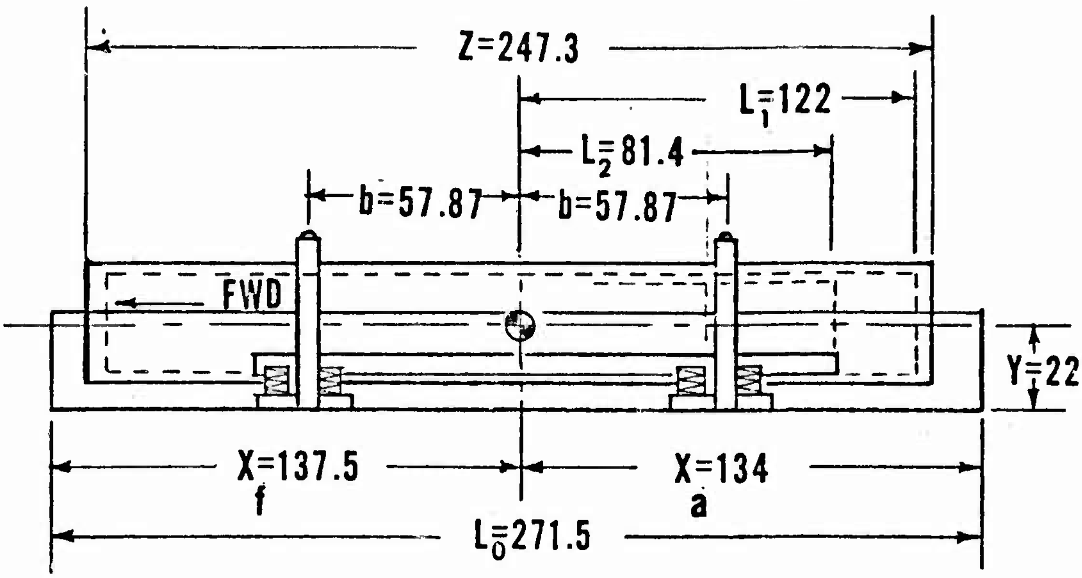
$G_1 = 3.8(8)(800)(1.3)(.9)/2948$

$G_1 = 9.6$ G's

5.3 DROP TEST CONFIGURATION

5.3 AFT END 18 INCH ROTATIONAL DROP

Ref: J. J. Goodill Paper "Flexible Suspension Systems" published in "Industrial Packaging" 10/15/56.



$$R^2 = X_f^2 + Y^2 = (137.5)^2 + (22)^2 = 19{,}390 \text{ in}^2$$

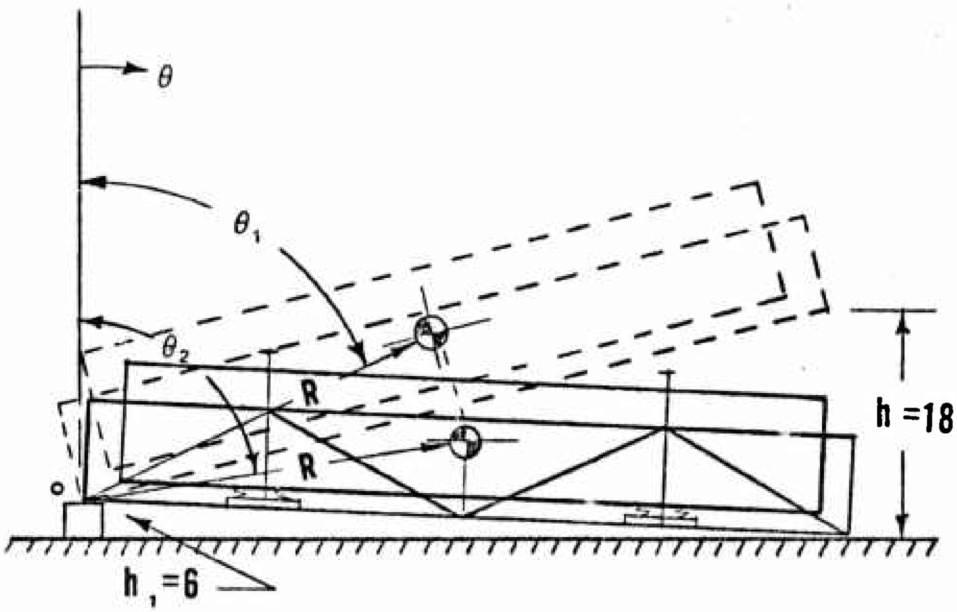
$$R = 139.25 \text{ inches}$$

$$m = \frac{2948}{386} = 7.637 \text{ lbs.-sec}^2 / \text{in.}$$

$$I_o = I_{CG} + mR^2 = 27{,}840 + 7.637(139.25)^2 = 175{,}930 \text{ in.-lb.-sec}^2$$

Radius of Gyration about Pivot Point "o" = r

$$r = \sqrt{\frac{I_o}{m}} = \sqrt{\frac{175{,}930}{7.637}} = \sqrt{23{,}035} = 151.77 \text{ inches}$$



5.3.1 DETERMINATION OF ANGLES θ_1 and θ_2

$$\theta_1 = 90^o - \tan^{-1}\left[\frac{Y}{X}\right] - \sin^{-1}\left[\frac{h - h_1}{L_o}\right]$$

$$\theta_1 = 90^o - \tan^{-1}\left[\frac{22}{137.5}\right] - \sin^{-1}\left[\frac{18 - 6}{271.5}\right]$$

$$\theta_1 = 90^o - 9.0903^o - 2.5332^o$$

$$\theta_1 = 78.3765^o$$

$$\cos\theta_1 = 0.20148$$

$$\theta_2 = 90^o - \tan^{-1}\left[\frac{Y}{X}\right] + \sin^{-1}\left[\frac{h_1}{L_o}\right]$$

$$\theta_2 = 90^o - 9.0903^o + 1.2633^o$$

$$\theta_2 = 82.176^o$$

$$\cos\theta_2 = 0.13613$$

5.3.2 AFT END 18 INCH ROTATIONAL DROP - continued

At impact, mounted unit's angular velocity = ω_0

$$\omega_0 = \left[\frac{2Rg(\cos\theta_1 - \cos\theta_2)}{r^2}\right]^{1/2}$$

$$\omega_0 = \left[\frac{2(139.25)(386)(.20148 - .13613)}{23{,}035}\right]^{1/2}$$

$\omega_0 = 0.55225$ Radians / second

5.3.3 18 INCH EDGEWISE DROP (Slightly more severe than cornerwise drop)

	Vertical Translational f_n	Rotational Circular f_n
	$\omega_1 = \left[\frac{K_v}{m}\right]^{1/2}$ Radians/second	$\omega_2 = \left[\frac{K_R}{I_{CG}}\right]^{1/2}$ Rad/sec.
-20°F	$\omega_1 = 73.8$	$\omega_2 = 70.74$
+70°F.	$\omega_1 = 66.0$	$\omega_2 = 63.27$
+140°F.	$\omega_1 = 62.6$	$\omega_2 = 60.03$

5.3.4 AFT END DROP -- Pivot Point "O" at Forward End:

-20°F.

$$A = \omega_1 R \omega_0 \cos \tan^{-1}\left[\frac{Y}{X}\right] \quad \text{in./sec.}^2$$

$$A = 73.8(139.25)(.55225)(.98744)$$

$$A = 5{,}604 \text{ in./sec.}^2 \div 386 = 14.5 = G_1$$

$$B = \omega_2 L \omega_0$$

$$B_1 = 70.74(81.4)(.55225)$$

$$B_1 = 3{,}179 \text{ in./sec.}^2 \qquad = 8.2 = G_{ha} \text{ due to rotation}$$

$$* \quad 22.7 \text{ G's} \doteq G_{ha}$$

$$B_2 = 70.74(122)(.55225) \qquad = 12.3 \text{ G's} = G_{ca} \text{ due to rotation}$$

$$+14.5$$

$$B_2 = 4{,}766 \text{ in./sec.}^2 \qquad = 26.8 \text{ G's} = G_{ca}$$

* ω_2 is only about 2.5% less than ω_1 so one can use the relationship $G_T = \frac{A}{g} + \frac{B}{g}$

+70°F.

$A = 66.0(139.25)(.55225)(.98744)$

$A = 5{,}011\ in./sec.^2 \quad = \quad 13.0\ G's = G_{CG}$

$B = 63.27(81.4)(.55225) \quad = \quad \underline{7.4}\ G's = G_{ha}$ due to rotation

$B = 2{,}844\ in./sec.^2 \quad G_{ha} = 20.4\ G's$

$B = 63.27(122)(.55225)$

$B = 4{,}236\ in./sec.^2 \quad = \quad 11.0\ G's = G_{ca}$ due to rotation

$+ \quad \underline{13.0}\ G's = G_{CG}$

$G_{ca} = 24.0\ G's$

$B = 63.27(57.87)(.55225)$

$B = 2{,}022\ in./sec.^2 \quad = \quad 5.2\ G's$ due to rotation

$+ \quad \underline{13.0}\ G's = G_{CG}$

AT OUTBOARD MOUNT $= 18.2\ G's$

+140°F.

$A = 62.6(139.25)(.55225)(.98744)$

$A = 4{,}753\ in./sec.^2 \quad = \quad 12.3\ G's = G_{CG}$

$B = 60.03(81.4)(.55225)$

$B = 2{,}698\ in./sec.^2 \quad = \quad \underline{7.0}\ G's = G_{ha}$ due to rotation

$G_{ha} = 19.3\ G's$

$B = 60.03(122)(.55225)$

$B = 4{,}044\ in./sec.^2 \quad = \quad 10.5\ G's = G_{ca}$ due to rotation

$+ \quad \underline{12.3}\ G's = G_{CG}$

$G_{ca} = 22.8\ G's$

5.3.4 AFT END DROP - Continued

5.3.4.1 Dynamic Deflection at Aft End of Capsule

+140°F.

$$d_t = \frac{A}{\omega_1^2} + \frac{B}{\omega_2^2}$$

$$d_t = \frac{4,753}{(62.6)^2} + \frac{4,044}{(60.03)^2}$$

$$d_t = 1.2 + 1.1 \text{ inches}$$

d_t = 2.3 inches at aft end of Capsule

5.3.4.2 DYNAMIC DEFLECTION at Outboard Mount Cluster

+140°F.

b = 57.87 inches

B = 60.03(57.87)(.55225)

B = 1,918 in./sec.^{2}

$$d_m = \frac{A}{\omega_1^2} + \frac{B}{\omega_2^2}$$

$$d_m = \frac{4,753}{(62.6)^2} + \frac{1,918}{(60.03)^2}$$

$$d_m = 1.2 + 0.5 \text{ inches}$$

d_m = 1.7 inches deflection at Mounts

+70°F.

$$f_n(\text{pitch}) = \frac{\omega_2}{2\pi} = \frac{63.27}{2\pi} = 10.07 \text{ Hz}$$

Deflection at Outboard Mounts:

$$d_m = \frac{A}{\omega_1^2} + \frac{B}{\omega_2^2}$$

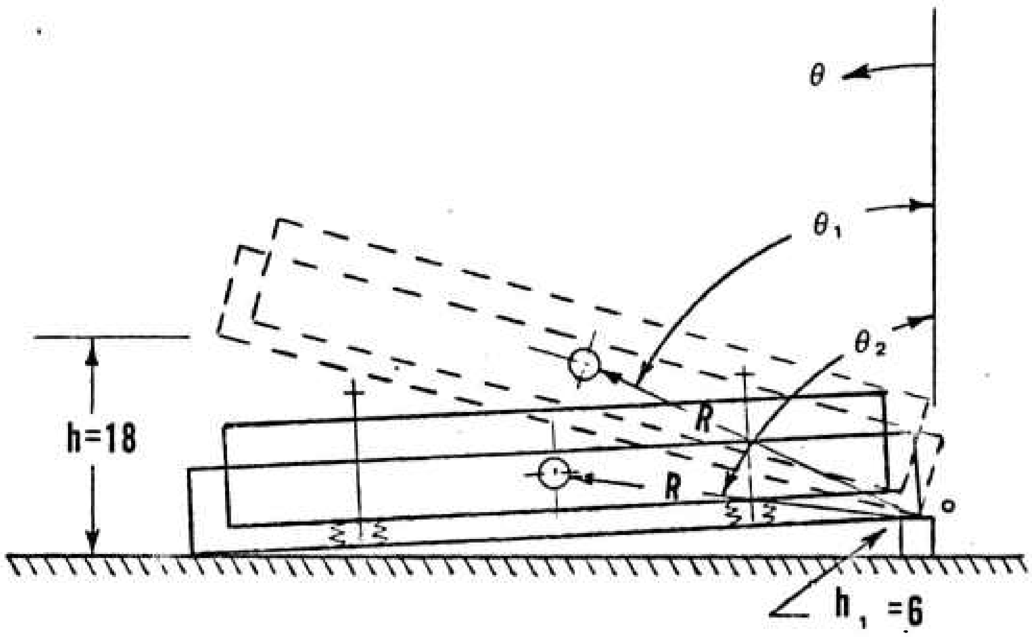
$$d_m = \frac{5,011}{(66.0)^2} + \frac{2,022}{(63.27)^2}$$

$$d_m = 1.15 + 0.50 \text{ inches}$$

d_m = 1.65 inches deflection at Outboard Mounts.

5.4 FORWARD END 18 INCH ROTATIONAL DROP

Pivot "o" at aft end



$$R^2 = X_a^2 + Y^2 = \overline{134}^2 + \overline{22}^2 = 18{,}440 \text{ in.}^2$$

$$R = 135.8 \text{ inches} \qquad m = \frac{2948}{386} = 7.637$$

$$I_o = I_{CG} + mR^2 = 27{,}840 + \frac{2948}{386}\left[135.8\right]^2 = 168{,}672 \text{ in.lb.sec.}^2$$

Radius of gyration about pivot point "o" = r

$$r = \sqrt{\frac{I_o}{m}} = \sqrt{\frac{168{,}672}{7.637}} = \sqrt{22{,}086}$$

r = 148.61 inches

5.4.1 DETERMINATION OF ANGLES θ_1 and θ_2

$$\theta_1 = 90^\circ - \tan^{-1}\left[\frac{Y}{X}\right] - \sin^{-1}\left[\frac{h - h_1}{L_o}\right]$$

$$\theta_1 = 90^\circ - 9.3236^\circ - 2.5332^\circ$$

$$\theta_1 = 78.1432^\circ$$

$$\cos\theta_1 = 0.20547$$

$$\theta_2 = 90^\circ - \tan^{-1}\left[\frac{Y}{X}\right] + \sin^{-1}\left[\frac{h_1}{L_o}\right]$$

$$\theta_2 = 90^\circ - 9.3236^\circ + 1.2633^\circ$$

$$\theta_2 = 81.9397^\circ$$

$$\cos\theta_2 = 0.14021$$

AT IMPACT, MOUNTED UNIT ANGULAR VELOCITY = ω_o

$$\omega_o = \left[\frac{2Rg(\cos\theta_1 - \cos\theta_2)}{r^2}\right]^{\frac{1}{2}}$$

$$\omega_o = \left[\frac{2(135.8)(386)(0.20547 - 0.14021)}{22086}\right]^{\frac{1}{2}}$$

$$\omega_o = 0.55657 \text{ rad./sec.}$$

SUSPENSION SPRING RATES - same as for aft end drops

ω_1 and ω_2 also the same.

5.4.2 FORWARD END DROP - PIVOT POINT "o" AT AFT END.

-20°F. $A = \omega_1 R\omega_o \cos\tan^{-1}\left[\frac{Y}{X}\right]$ and, $\cos\tan^{-1}\left[\frac{22}{134}\right] = 0.98679$

$$A = 73.8(135.8)(.55657)(.98679)$$

$$A = 5{,}504 \text{ in./sec.}^2 = 14.2 \text{ G's @ C.G.}$$

$$B_1 = \omega_2 L_1 \omega_o$$

$$B_1 = 70.74(98.97)(.55657)$$

$$B_1 = 3{,}897 \text{ in./sec.}^2 = \underline{10.1} \text{ G's} = G_{hf} \text{ due to rotation}$$

$$G_{hf} = 24.3 \text{ G's total}$$

$$\text{vs} \quad G_{ha} = 22.7 \text{ G's total.}$$

5.5 DEFLECTION

Aft End of Capsule during rotational drops:

140°F.

$$d = \frac{A}{\omega_1^2} + \frac{B}{\omega_2^2}$$

$$d = \frac{4753}{(62.6)^2} + \frac{4{,}044}{(60.03)^2}$$

$$d = 1.2 + 1.1$$

$$d = 2.3 \text{ inches}$$

Capsule during 18 inch Flat Drop @ +70°F.

$$d = 1.78 \text{ inches}$$

Capsule during 10 ft./sec. End Impact @ +140°F.

$$d = 3.8 \text{ inches}$$

5.6 SHOCK REBOUND

5.6.1 Rebound load and deflection at mounts as related to loads and deflections due to initial impact:

REF: Dynamics of Package Cushioning, R. D. Mindlin Bell Telephone Laboratories 1945 pp. 44-47

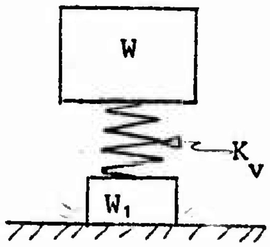
$$G_1 = \sqrt{\frac{2hK_v}{W}}$$

$$G_1 = \sqrt{\frac{2(18)(33280)}{2950}}$$

$$G_1 = 20$$

$$\frac{G_r}{G_1} = \frac{W_1}{W + W_1}\sqrt{1 - \frac{W_1}{2hK_v}}$$

$$G_r = \frac{20(550)}{2950 + 550}\sqrt{1 - \frac{550}{2(18)(33280)}}$$



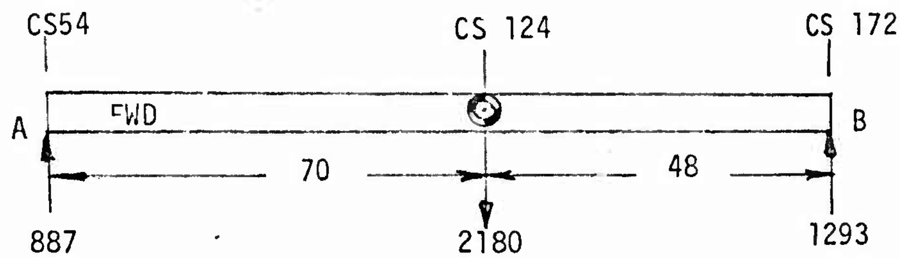
$$G_r = 3.14$$

$$d_r = \frac{G_r W}{K_v} = \frac{3.14(2950)}{33280}$$

$$d_r = 0.28 \text{ inch}$$

6.1 LOADS - CRADLES

1G distribution of Capsule/HARPOON load on cradles:



$M_A = 2180(70) - R_B(118) = 0$

$R_B = 1293$ LB. $R_A = 887$ LB.

With EC at CG, upon impact due to aft end rotational drop the unit will revolve about the CG-EC, resulting in displacement and load due to rotation being proportional to distance from CG-EC. Adding loads and displacements due to translation gives total G forces at any point.

Thus, to determine load at CS-172 Cradle, there is a load of 7.4 G's at CS-205.4, so on cradle at CS-172 there will be:

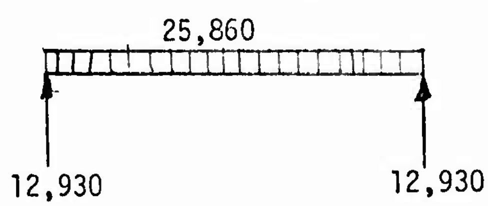
$$G_{172} = 13.0 + \frac{(172 - 124)}{(205.4 - 124)}(7.4) = 13.0 + 4.4 = 17.4 \text{ G's}$$

$$P_{172} = 17.4(1293) = 22{,}498 \text{ lbs.}$$

In 18 inch flat drop there is load of 20 G's x 1293 = 25,860 lbs.

Cradle load is most severe in the flat drop.

This 25,860 lb. load is carried 1/2 on each end of the cradle.



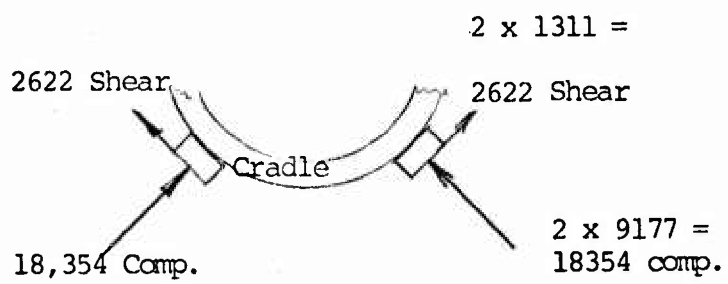
6.2 LOADS ACTING ON MAIN SUPPORT BEAMS

Bending across the 3 inch depth of beam is considered most critical and the 18 inch flat drop loading will be maximum.

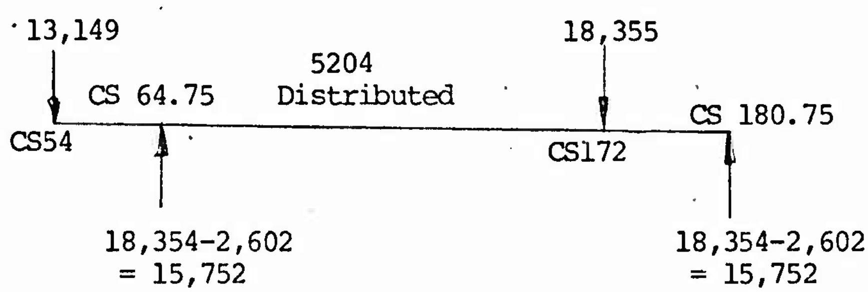
Acting across this 3 inch depth are loads at CS54 and CS172 (where cradles are attached) and reaction loads from mounts at 64.75 and 180.75 as shown on diagram. Loads on beam at cradles must also include 296 pounds of fiberglass shell, which is clamped between beams and cradles, and 54 pounds of two cradles. Inlcuded in the total 2948 pound suspended load is 418 pounds for beams and 1/2 of weight of mounts.

Mount reaction loads are based upon deflection which takes no account of energy dissipated in mounts during deflection.

Forces exerted by mounts along their compression axes, centered at CS 64.75 and CS 180.75 are 2 (9177) = 18,354 lbs. (See paragraph 6.4) Of this, $\frac{418}{2948}$ (18,354) = 2602 lbs. at each mount cluster is due to weight of beam and 1/2 mounts.



Loads acting across the 3 inch depth of beam are shown below:



$$\sum M_{CS\ 54} = 10.75\ (15,752) + \chi\ (118) - 126.75\ (15,752) = 0$$

$$\chi = 18,355 \text{ lbs.}$$

6.3 LOADS AT RESILIENT MOUNT SUPPORTS

Rotational drop of aft end

Forces at mounting stations

Total at CS182 --- $P = \dfrac{2948(18.2)}{2} = 26,827$ lbs.

Total at CS66 --- $P = \dfrac{2948(13 - 5.3)}{2} = 11,497$ lbs.

Above loads act on outer frame during rotational drops.

18 inch flat drop results in 20 G's, so is more severe loading on pedestals carrying resilient mounts:

$P = 2948(20)/12 = 29,480$ lbs. at each mount station.

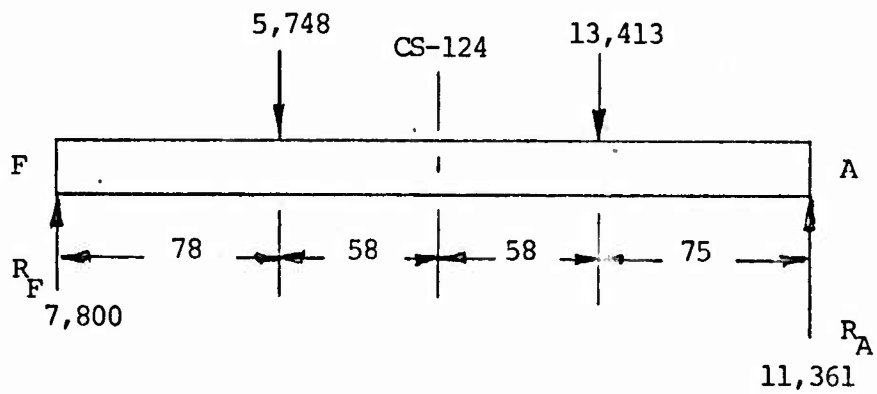
LOADS ACTING ON OUTER FRAME ON EACH SIDE, during rotational drops, causing corner loading.

DIVIDING by 2 the vertical loads derived above:

NOTE: Mount loads are applied at bottom of outer frame.

$$M_F = 5748(78) + 13413(194) - R_A(268.5) = 0$$

$R_A = 11,361$ lbs. $R_F = 7,800$ lbs.

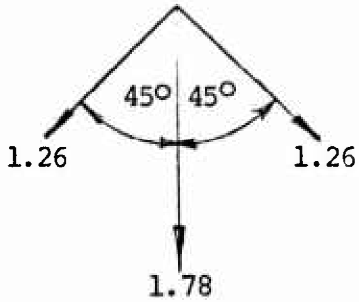


2.6

6.4 LOADS ACTING AT MOUNT SUPPORT POINTS AND REACTING ON BEAMS

18 inch flat drop at 70° F.

With mounts at 45° angle, a vertical deflection of 1.78 inches results in displacements of 1.26 inches along compression and shear axes of mounts.

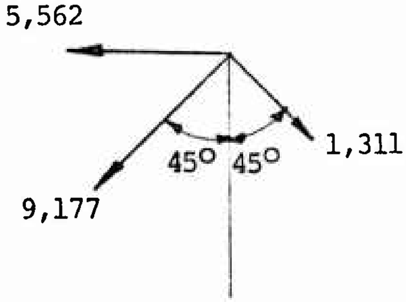


Force along shear axis of mount =
$k_d(d) = 800(1.3)\ (1.26) = 1{,}311$ lbs.

Force along compression axis of mount =
$k_d(d)\ L' = 1{,}311(7) = 9{,}177$ lbs.

At each support point there are two mounts. Each mount is supported on a short column which in turn is supported by a cross member. On top of the columns is a single plate to which the two mounts are bolted. This plate is welded to the stacking member and to the two cross members, for increased rigidity.

6.5 LOADS (LATERAL) ON MOUNT PEDESTALS



$P_H = 9177 \cos 45^{\circ} - 1311 \cos 45^{\circ} = 0$

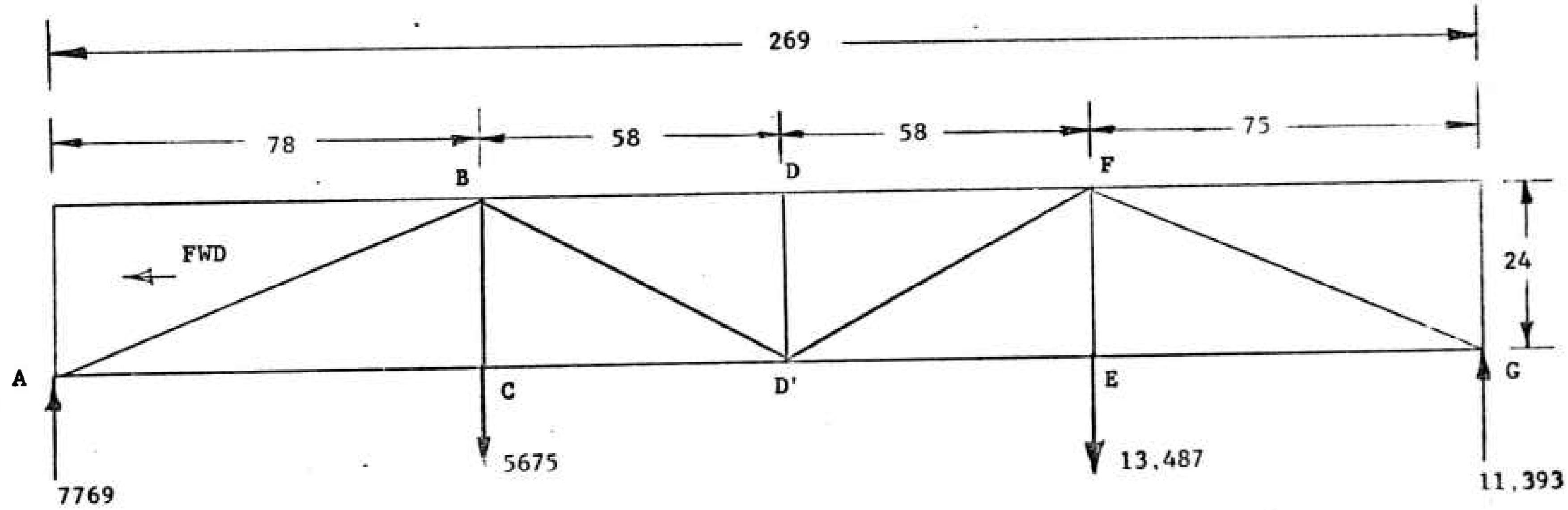
$P_H = 5{,}562$ lbs. acting laterally

6.6 LOADS (LONGITUDINAL) ON MOUNT PEDESTALS

The 10 ft/sec end impact results in a maximum of 11.3 G's on 2948 lb. suspended mass, or a load of 11.3(2948) = 33,312 pounds distributed over 8 pedestals, for unit pedestal load = 4164 lbs., which is lower than lateral load of 5562 lbs.

6.7 STRAP ASSEMBLY LOADS

Rebound load = $G_r W = 3.14(2950) = 9{,}263$ lbs. total



OUTER FRAME
Figure 4

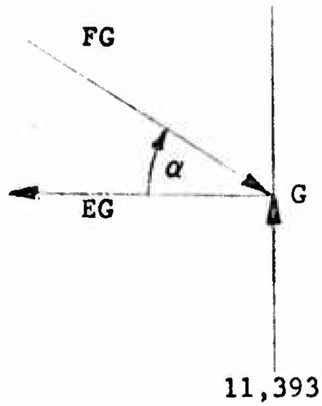
6.8 LOADS ON OUTER TRUSS MEMBERS

Aft End 18 inch Rotational Drop

NOTE: Mount loads are applied at bottom of frame.

Rotation about Point A (Fwd end)

AT POINT G:



$\tan\alpha = \frac{24}{75} = 0.32 \qquad \alpha = 17.745^{\circ}$

$\sin\alpha = 0.30478$

$\cos\alpha = 0.95242$

$\Sigma P_Y = -11{,}393 + FG \sin\alpha = 0 \qquad FG = 37{,}381$

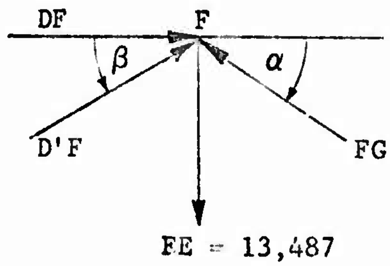
$\Sigma P_X = FG\cos\alpha = EG = 0 \qquad EG = 35{,}602$

AT POINT F:

$\tan\beta = \frac{24}{58} = 0.41379$

BY INSPECTION:

FE = 13,487



$\sin\beta = 0.38235$

$\cos\beta = 0.92402$

$\Sigma P_Y = D'F \sin\beta + GF \sin\alpha - 13{,}487 = 0$

$D'F = 5{,}477$

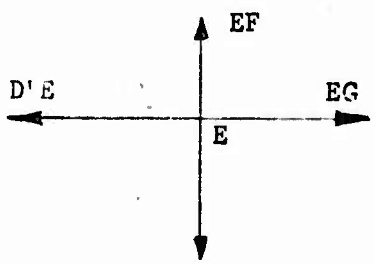
$\Sigma P_X = -FG \cos\alpha + D'F \cos\beta + DF = 0$

$DF = 30{,}542$

AT POINT E:

$\Sigma P_Y = -13487 + EF = 0 \qquad EF = 13{,}487$

$\Sigma P_X = EG - D'E = 0 \qquad D'E = 35{,}602$



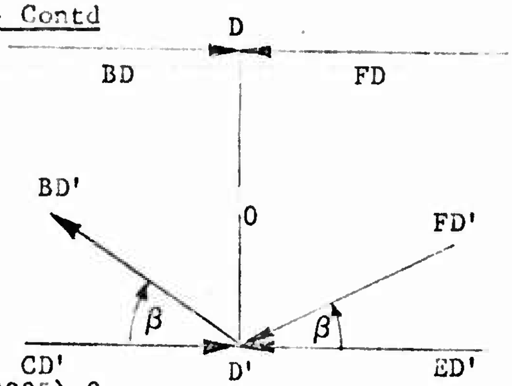
6.8 LOADS ON OUTER TRUSS MEMBERS - Contd

AT POINT D:

$\Sigma P_Y = DD' = 0$

$\Sigma P_X = BD - FD = 0$

$BD = 30,542$



AT POINT D':

$\Sigma P_Y = D'D + BD' \sin \beta - FD' \sin \beta$

$= 0 + BD'(0.38235) - 5477(0.38235) = 0$

$BD' = 5477$

$\Sigma P_X = CD' - ED' = 0 \qquad CD' = 35,602$

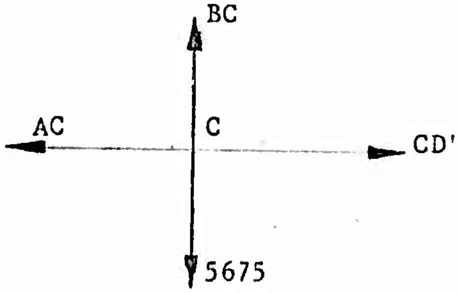
AT POINT C:

$\Sigma P_Y = BC - 5675 = 0$

$BC = 5675$

$\Sigma P_X = CD' - AC = 0$

$AC = 35,602$



AT POINT B:

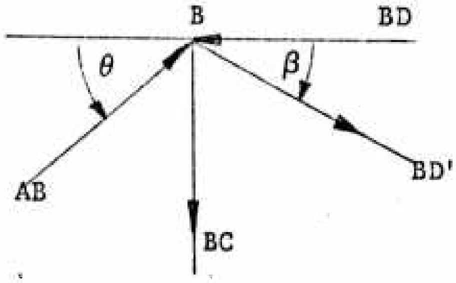
$\operatorname{Tan} \theta = \dfrac{24}{78} = 0.30769$

$\operatorname{Sin} \theta = 0.29409$

$\operatorname{Cos} \theta = 0.95578$

$\Sigma P_Y = AB \sin \theta - BD' \sin \beta - BC = 0$

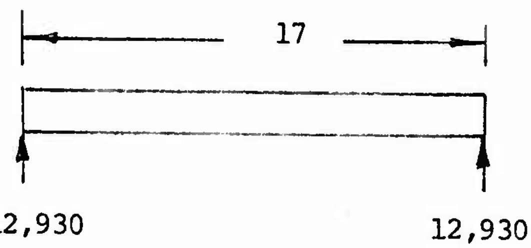
$AB \doteq 26,415$



6.9 STRESS

6.9.1 CRADLE

Cradle is curved beam of relatively large diameter so for simplicity is considered as a uniformly-loaded simple beam of horizontal projected length of 17 inches.



$M = w\ell^2/8$ and $w = 25{,}860/17$
(Ref. Para. 6.1)

$M = 25{,}860(17)^2/(17)(8) = 54{,}952$ in-lb.

$$Y_{max} = \frac{5W\ell^3}{384EI} = \frac{5(25{,}860(17)^3}{384(10)^7(I)} = \frac{0.165}{I}$$

and $I = 4.93\ in^4$ for section shown in Loads Section

$Y_{max} = 0.165/4.93 = 0.033$ in.

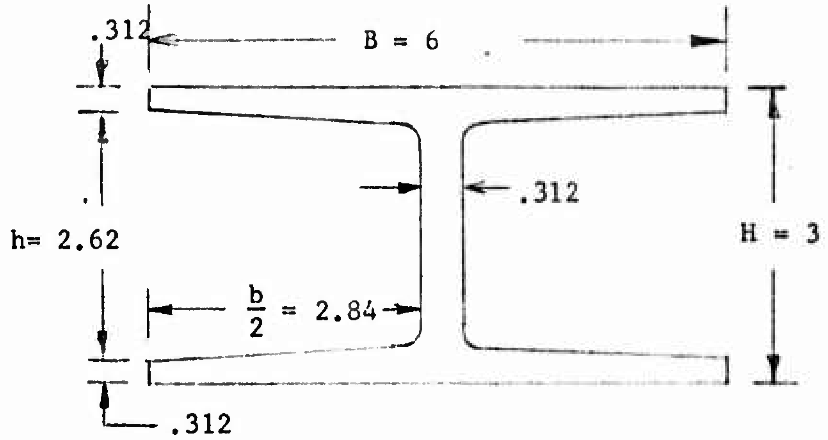
Using 220-T4 Aluminum Casting, Y.S. = 22,000 psi

$$S = \frac{Mc}{I} = \frac{54{,}952(1.5)}{4.93} = 16{,}719 \text{ psi}$$

$$F.S. = \frac{22{,}000}{16{,}719} = 1.32$$

As flanges will be thicker than the 0.31 used to derive I, stresses and deflection should be lower than calculated.

6.9.1 CRADLE - STRESS (continued)



$$I = \frac{BH^3 - bh^3}{12}$$ (Ref: Marks Handbook)

$$I = \frac{6(3)^3 - 5.6875(2.625)^3}{12}$$

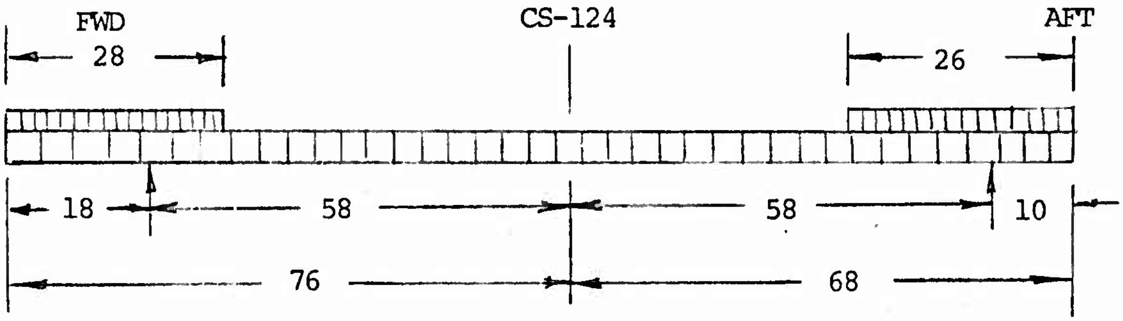
$$I = \frac{162 - 102.87}{12}$$

$$I = 4.93 \text{ in.}^4$$

$$I/c = \frac{BH^3 - bh^3}{6H} = 3.285 \quad \text{or} \quad \frac{4.93}{1.5} = 3.286$$

6.9.2.1 MAIN BEAM BENDING - STRESS AND DEFLECTION

Beam weighs 127 pounds uniformly distributed along 144 inch length and 34 pounds distributed along 28 inches at forward end and 32 pounds distributed along 26 inches at aft end. This is shown below:



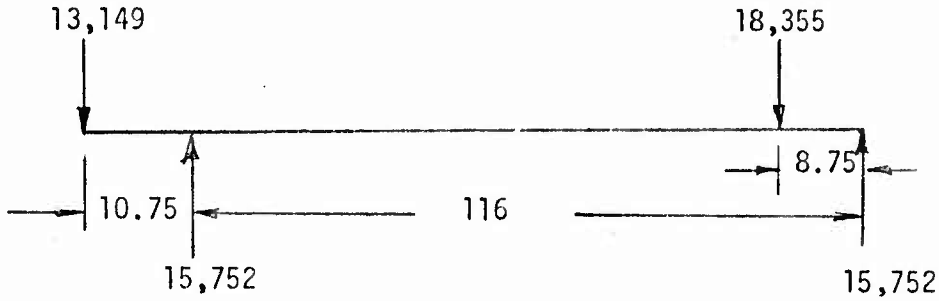
We can make simplifying assumptions (conservative in nature) to approximate deflection and stresses due to bending caused by loading as shown.

(a) Assume loads at ends due to weight of stiffeners act directly down on supporting mounts so they do not contribute to beam deflection or bending stress.

(b) Assume that bending is caused only by the 102 pound uniformly distributed load of 116 inch beam length between supporting mounts.

6.9.2.2 STRESS AND DEFLECTION OF MAIN BEAMS

(a) Due to point loads as shown



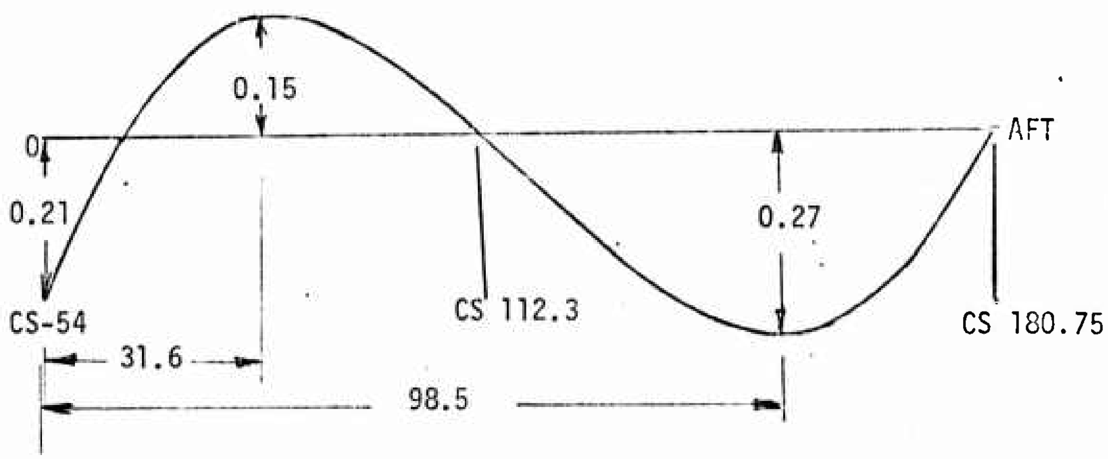
Using the method of double integration, main beam deflection equations were determined to be:

$$y_1\,(EI) = -2191.5\,X^3 + 3138531.9X - 31016723.7 \qquad \text{for } 0 \leqslant X \leqslant 10.75$$

$$y_2\,(EI) = 433.8\,X^3 - 84667\,X^2 + 4048702.1\,X - 34278125.7 \qquad \text{for } 10.75 \leqslant X \leqslant 118$$

$$y_3\,(EI) = -2625.3\,X^3 + 998278\,X^2 - 123738807.9\,X + 4991921064 \qquad \text{for } 118 \leqslant X \leqslant 126.75$$

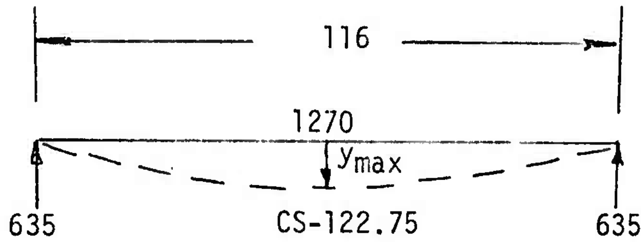
which yield the deflection curve shown below:



6.9.2.2 STRESS AND DEFLECTION OF MAIN BEAMS - Cont'd

Of the 418 pound beam weight, 102 pounds are considered as causing bending due to weight of beam between mountings. Thus,

$\frac{102}{418}$ (5204) or 1270 pounds uniformly distributed, causes bending across the three inch depth of beam. (Ref para 6.2 on page 25).



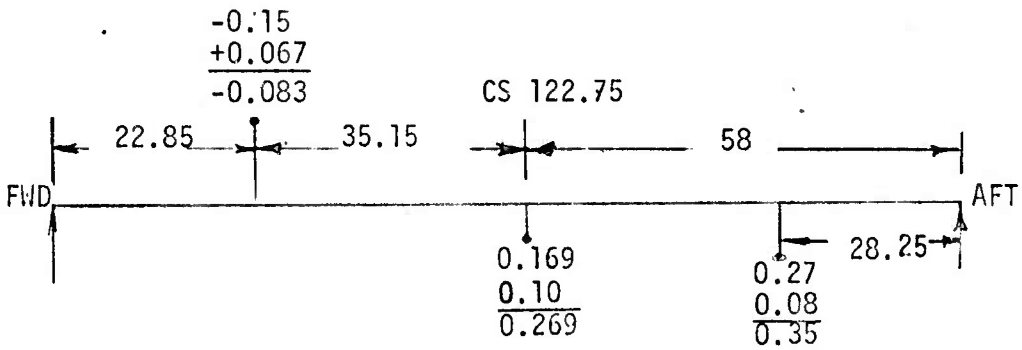
$$y_{max} = \frac{5WL^3}{384EI} \text{ at center of span}$$

$$y_{max} = \frac{5\ (1270)(116)^3}{384(30 \times 10^6)\ \ (5.08)} = 0.169 \text{ inch}$$

At point where deflection is 0.15 upward, which is 31.6 inches from forward end of beam or 31.6 - 8.75 = 22.85 from forward mount point at CS 64.75, beam weight causes downward deflection of (22.85/58)(0.169) = 0.067.

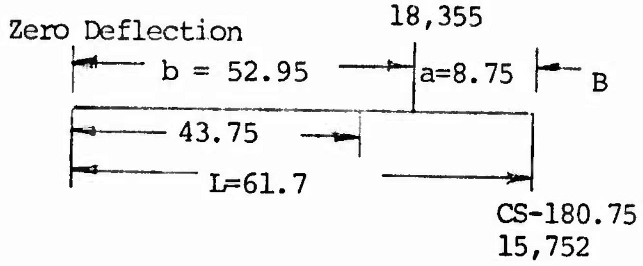
Similarly, at point of 0.27 deflection there is added deflection of (28.25/58)(0.169) = 0.08 inch due to beam weight.

6.9.2.3 TOTAL DEFLECTIONS



6.9.2.4 STRESS IN BEAM:

For aft end



$$M_{max} = \frac{Pab}{L} \text{ at } a = 8.75$$

$$M_{max} = \frac{18,355\ (8.75)\ (52.95)}{61.7}$$

$$M_{max} = 137,830$$

$$S = \frac{Mc}{I} = \frac{137,830\ (1.5)}{17.9} = 11,550 \text{ PSI}$$

S = 11,550 PSI vs nominal 30,000 PSI allowable; however, maximum stress will occur 6 inches forward on the beam where the stiffening plates (at the end) terminate.

$$M = \frac{43.75}{52.95}\ (137,830) = 113,882$$

$$S_{max} = \frac{113,882}{5.02} = 34,028$$

Although the calculated S_{max} exceeds the nominal allowable it is not considered to be significant because the anlaysis is conservative in that it doesn't include the positive effect of the following:

1. Damping in the mounts
2. Elastic deformation of the outer frame.
3. Local plastic deformation at the impact point.
4. Negative moments developed in the cradles.
5. Stiffening effect of the fiberglass shell.

6.9.2.5 STRESS IN MAIN BEAMS - Continued

This approximation is conservative because:

(a) distributed loads have been considered as point loads, and

(b) the effects of 3/8 inch thick reinforcing plates on ends have not been taken into consideration. ($I = 17.9$ in.^{4} at ends).

By measuring beam deflections during drop tests, it can be determined if it is desirable to stiffen the center section of the beams, although this will add weight.

6.9.3 STRESS IN MOST HEAVILY LOADED FRAME MEMBERS

(a) Compression load of 26,415 lbs.

$$S_c = \frac{P}{A} = \frac{26,415}{2.1146} = 12,492 \text{ p.s.i vs } 30,000 \text{ p.s.i for 6061-T6}$$

Diagonal considered is approximately 75 inches long and $\ell/k = 75/1.1492$

$= 65$,

which is less than 80 so need not be considered as a column liable to buckling.

(b) Tension load of 35,602 lbs.

$$S_t = \frac{P}{A} = \frac{35,602}{2.1146} = 16,836 \text{ p.s.i. vs } 30,000 \text{ psi for 6061-T6}$$

6.9.4 STRESS IN STACKING LEGS

Five containers high x 3 ft. = 15 feet.

Legs on bottom unit carry 3,500 pounds each

$$S_c = \frac{P}{A} = \frac{3,500}{2.1146} = 1,655 \text{ p.s.i. vs } 30,000 \text{ p.s.i. for 6061-T6}$$

6.9.5 STRESS IN HOLD-DOWN STRAPS AND ELEMENTS DUE TO CLAMPING FORCE

Required restraining force = P = GW = 12.2(2180) = 26,600 lbs.
during end impact

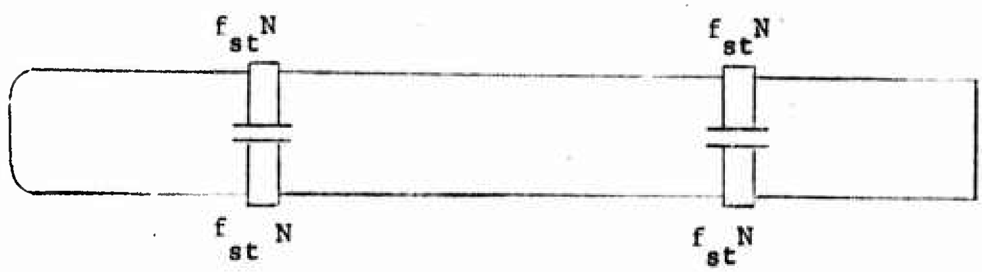
Using f_{st} = 0.5 = static coefficient of friction
for rubber on painted aluminum

There are two cradles with clamping force at top and at bottom:

$f_{st} \doteq 0.5$ is conservative: Ref: MIL-HDBK-149A, p.18.

$$\sum P_x = 4\, f_{st} N - 26,600 = 0$$

$$N = \frac{26,600}{4(0.5)} = 13,300 \text{ lbs.}$$



Tension in strap is:

$P_1 + P_2 = N$ and as $P_1 = P_2$, $N = 2P_1$

$P = \frac{N}{2} = \frac{13,300}{2}$ = 6,650 lbs. and there are two bolts per strap

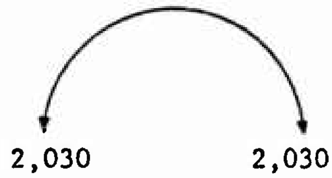
Tightening torque = T_b = 0.2dW = 0.2(0.375)(6650)/2 = 250 in.-lbs.

Stress in strap = $\frac{6650}{.06(6)}$ = 18,472 psi vs 110,000 Y.P. in 301SS ½ hard

F.S. ~ 6

6.9.5 STRESS IN HOLD-DOWN STRAPS - continued

18 inch flat drop is most severe condition



Rebound load = $G_r W$ = 3.14(2130)

= 6,845 lbs.

At C.S.-172:

Rebound Load = 3.14(1,293) = 4,060 lbs., with ½ carried on 2 bolts.

At C.S.-54:

Rebound Load = 3.14(887) = 2,785 lbs.

Straps are SS301 ½ hard, with allowable stress = 110,000 P.S.I.

Straps are 0.06 x 6 inch = 0.36 in.^{2}

$$S = \frac{2,030}{0.36} = 5,639 \text{ P.S.I. vs } 110,000 \text{ P.S.I.} \quad \text{Y.P.}$$

F.S. ~ 20

6.9.6 TEE BOLT STRESS

Minimum cross sectional area = πr^2 = $3.14(0.322/2)^2$

= 0.086 in.^{2} per bolt

Two (2) bolts carry load of 2,030 lbs. or 1,015 lbs. per bolt

S = 1,015/0.086 = 11,802 p.s.i.

Tee bolts are 4130 alloy steel at 125,000 p.s.i.

F.S. = 125,000/11,802 = 10.6

6.9.7 HINGE PIN SHEAR STRESS

Pin diameter = 0.401 inch and shear load is carried on 2 shear areas:

$$S = \frac{P}{A} = \frac{2,030}{2\pi(.200)^2} = \frac{2,030}{.2526} = 8,036 \text{ p.s.i.}$$

F.S. ~ 9

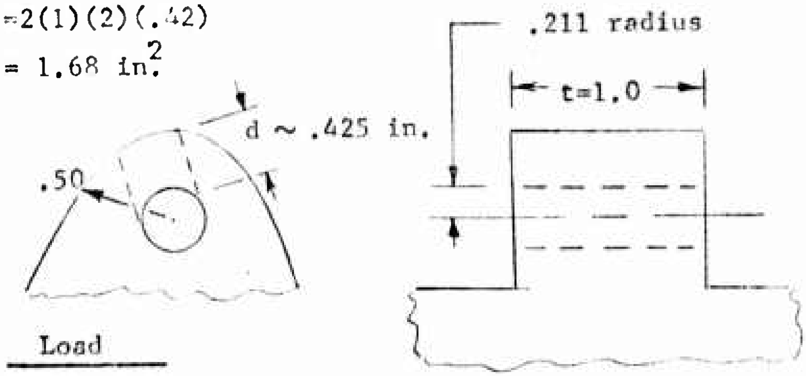
6.9.8 STRESS - PIN, CRADLE TO STRAP

Ref: Alcoa Handbook, p.263 220T4 Shear Strength = 34,000 psi

Shear Area = 2(t)(2d)

= 2(1)(2)(.42)

= 1.68 in.^{2}



$$\text{Stress} = \frac{\text{Load}}{\text{Shear Area}}$$

$$S = \frac{6782}{1.68} = 4{,}036 \text{ p.s.i.}$$

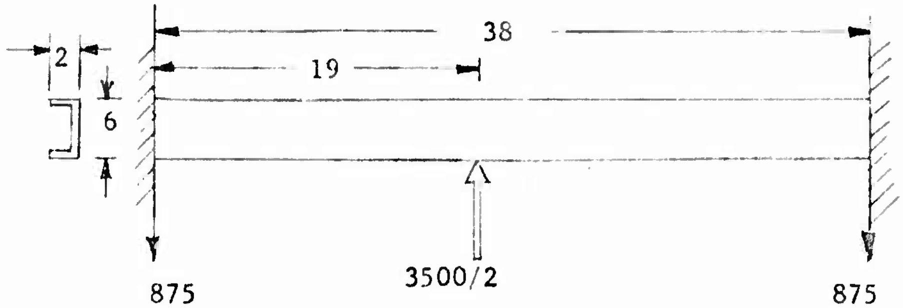
F.S. = 34,000/4,036 = 8.4

6.9.9 STRESS IN LATERAL CHANNEL DUE TO HAND TRUCK HANDLING

$I = 9.007$

$c = 3$

$I/c = 3.002$



Consider channel as fixed beam: concentrated load at center.

$$M = \frac{P\ell}{8} = \frac{1750(38)}{8}$$

$M = 8{,}312$ in-lb

$$\text{Deflection } y = \frac{W\ell^3}{192EI} = \frac{1750(38)^3}{192(10)^7(9.007)} = 0.0055 \text{ inch}$$

$$S = \frac{Mc}{I} = \frac{8{,}312(3)}{9.007} = 2{,}768 \text{ p.s.i.}$$

MIL-STD 648, Para. 4.17.7a, page 19, requires
Load = 7,500 lbs. be carried at single end fitting:

$$M = \frac{7500(38)}{8} = 35{,}625 \text{ in-lb}$$

$$y = \frac{7500(38)^3}{192(10)^7(9.007)} = 0.024 \text{ inch}$$

$$S = \frac{Mc}{I} = \frac{35{,}625(3)}{9.007} = 11{,}865 \text{ p.s.i.}$$

6.9.10 STRESS - HOISTING RING

Worst condition is lifting of full load on one point, as required by OR-11.

6061-T6 has Y.P. = 20,000 p.s.i. (Shear)

For load of 3500 lbs., minimum shear area required = $\frac{3{,}500 \text{ lbs.}}{20{,}000 \text{ lbs/in}^2}$

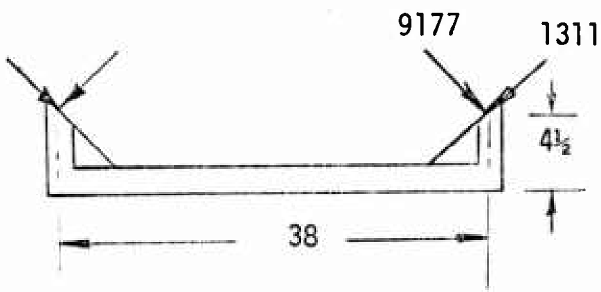
$\text{Area}_{\text{min.}} = 0.175 \text{ in}^2$ minimum

Using 1/2 inch thick plate with 1 inch minimum distance to hole from edge gives 0.5 sq.in. and for shear it would be 1.0 sq.in. so

$$F.S. = \frac{1.0}{0.175} = 5.7$$

6.9.11 BENDING IN 3 X 3 X 3/16 LATERAL FRAME MEMBERS:

Consider load from individual mount on pedestal on each side as acting horizontally at point 4.5 inches above neutral axis of cross member. Resolving mount's compression and shear vectors into horizontal forces:



$$\Sigma P = 9177 \cos 45^\circ - 1311 \cos 45^\circ = 0$$

$$= 5562 \text{ lbs. acting outward}$$

With moment = 5562 (4.5) = 25,029 in.-lbs at each end of 35 inch wide lateral,

$$\text{Maximum deflection} = \frac{M_o L^2}{8\ EI} = \frac{25{,}029\ (35)^2}{8(10)^7(2.7993)} = 0.137 \text{ inch}$$

7.0 RESILIENT SUSPENSION DESIGN

Because of limit on loads during end impact, it is desirable to have a relatively low spring rate along longitudinal axis and to have large deflection capability. Elastomeric shear mounts having shear axes parallel to length of weapon are desirable.

The system can be stiffer in the vertical direction due to additional support given HARPOON by Sabot support. Laterally, system may be even stiffer but it is important to minimize the rocking response, particularly during lateral vibration.

A resilient elastomeric sandwich mount suspension was conceived, with mounts located on each side of and below the longitudinal axis of the assembly, having their compression axes focused to a point above the longitudinal axis through the C.G. of the assembly, thus projecting the elastic center of the mounts to that axis.

Focal angle, vertical distance from horizontal plane through C.G. to mount's elastic center, distance from vertical plane through C.G. to mounts' E.C., and focal angle θ are inter-related with the "L'" ratio of the mount. This is shown graphically on Figures 5 and 6.

A 45° focal angle was chosen, with an estimated ratio of compression to shear spring rate of L' = 7.

Mounts are in shear along longitudinal direction for end impact. Length of mount is greater than the anticipated deflection, to insure that an adequate column of elastomer carries the vertical load during deflection in shear.

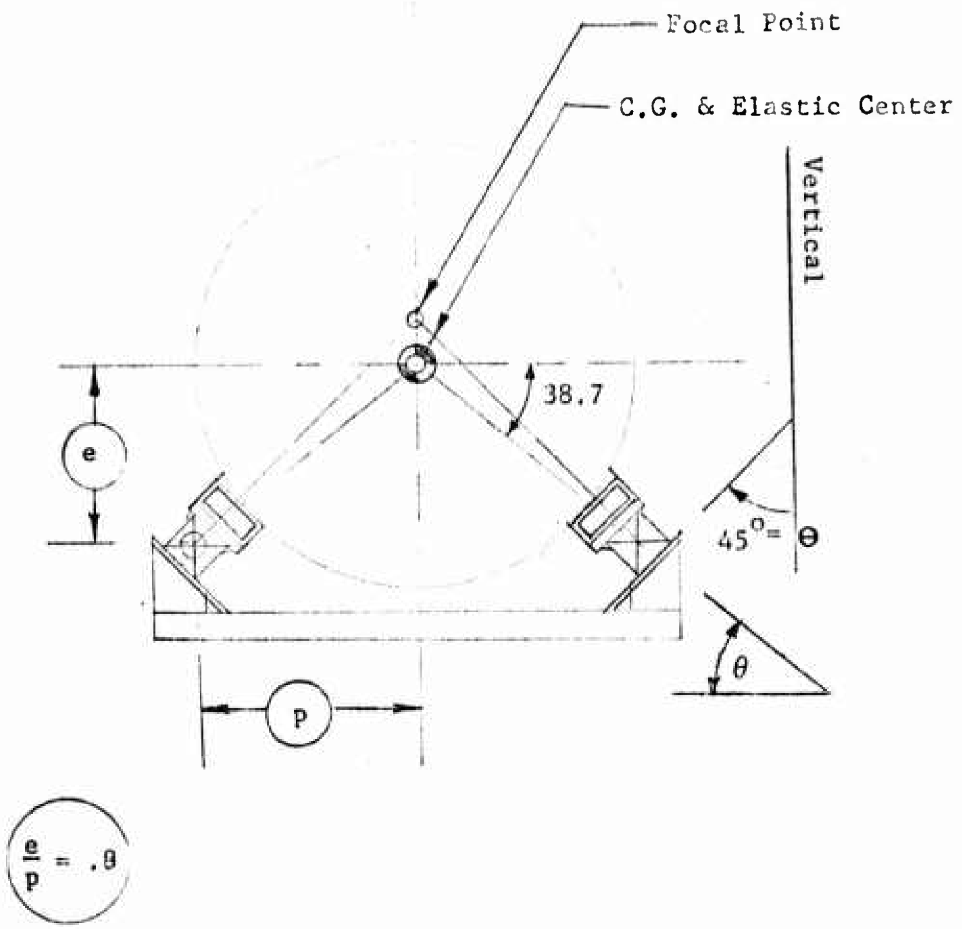
Because the stipulated support points on the Capsule are not symmetrical about the C.G., a structure is required to support the Capsule on the upper side and to pick up mounts on the lower side. Hollow rectangular steel tubes were selected, with cast aluminum saddles for the weapon, this assembly being supported on mounts. Mount spacing was predicated on keeping bending in the tubes to a minimum while providing a pitch natural frequency in the 10 Hz region.

As the weight of the saddles and tubes is below the C.G. of the weapon assembly, the overall C.G. is lowered slightly below the longitudinal axis of the weapon assembly.

Until mounts are made and tested, their spring rates and L' value won't be known accurately. When such information is derived, consideration will be given to moving mounts slightly to modify the e/p ratio.

If L' decreases to 5.5 for vibration at resonance, and unacceptable rocking modes are revealed during testing, decoupling might be accomplished by moving mounts outwards and upwards approximately 0.4 inch on the supporting structure and on the beams.

Again, measured spring rates are essential to this calculation.



MOUNT LATERAL SPACING DETERMINATION

FIGURE 5

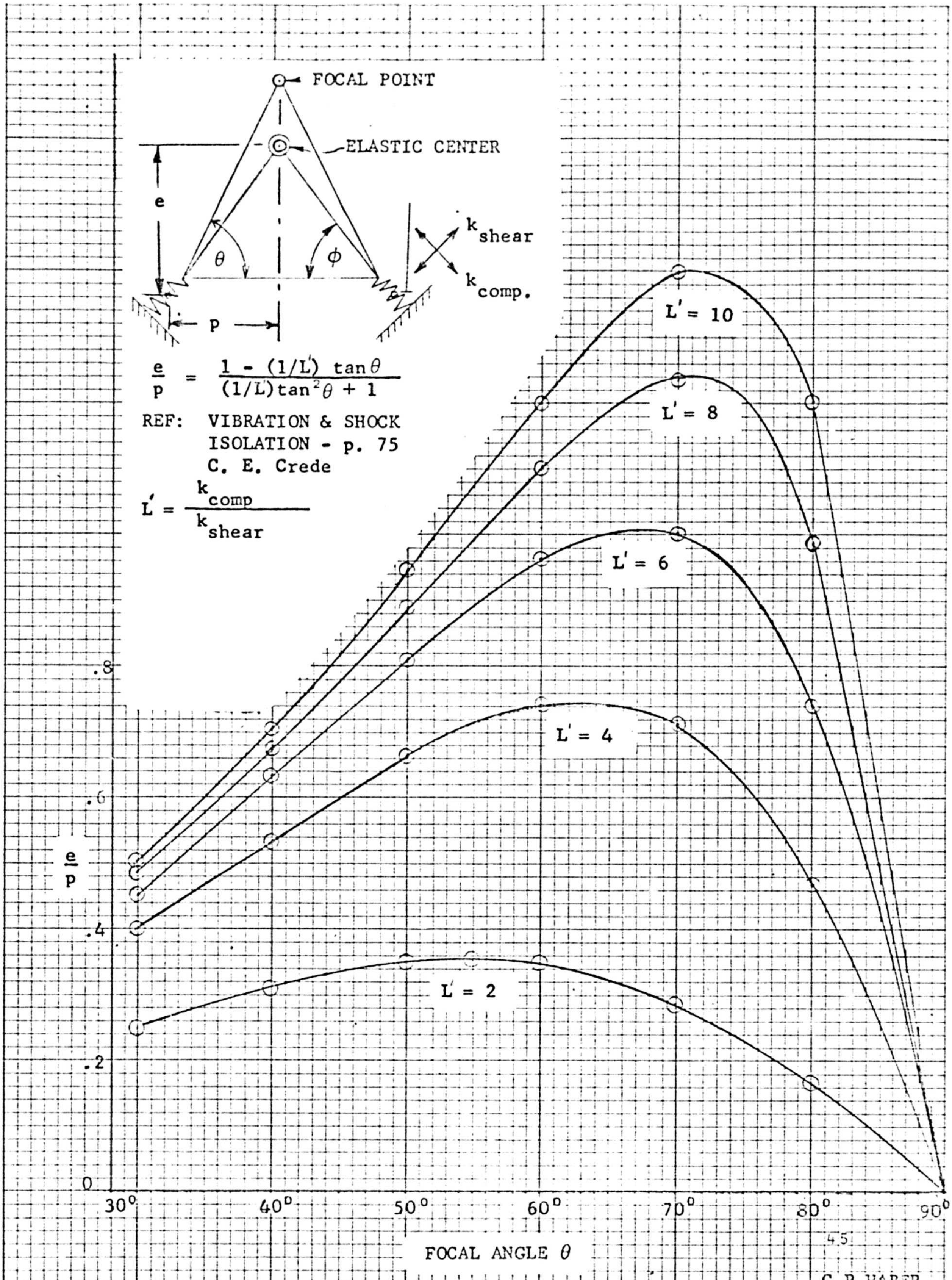


FIGURE 6

C.P. HABER
8-21-75

7.1 RESILIENT SUSPENSION ELEMENT DESIGN

Mount design is based on suspension system requirements determined by analysis of a focalized suspension system in which bonded metal/elastomer/metal sandwich mounts are loaded in both compression and shear simultaneously. A nominal static shear spring rate of 800 - 825 pounds per inch per mounting and a compression to shear spring rate ratio of seven was required. Also needed was damping adequate to keep transmissibility at resonance to four maximum, which requires a fairly hard elastomer (nominally 65 durometer was selected).

Mounting configuration selected is similar to a component used previously elsewhere but increased in thickness to four inches. Unit is hollow, to help provide linear spring rate characteristics along compression axis, and to permit changes in spring rates by varying size of mold insert, rather than by changing stiffness of elastomer. The need for a stiff elastomer recommends this approach.

With a 1.25 inch rubber wall thickness and a three inch column height being compressed, there exists a possibility of some buckling of the mount along its compression axis. However, this should not occur until deflections are approximately the calculated 1.2 inch along the compression axis. Final determination of mounting buckling characteristics should be made when mounts become available.

In the calculations it was assumed that mounts had linear load-deflection characteristics, but this is only an approximation, as shown by the calculated L-D curves. Data from parts should be reviewed as soon as it is available.

7.1 RESILIENT SUSPENSION ELEMENT DESIGN (Cont'd)

The ratio of compression to shear spring rate used in drop calculations is 7:1 (L' = 7). For this value and the relationship between mounting half-spread (lateral) and height of CG above mounts elastic center, with mounts' compression axis angled 45° from the vertical, a decoupled system is provided. Thus a lateral translational input should result in a translational response. The mounts' lateral spacing may require modification after initial testing. The L-D curves are not linear and this results in lowering of the "L'" value for small deflections.

The elastomer chosen retains its properties quite well over the temperature range that the suspension system is required to meet. Factors used in calculations are based upon vendor data and are as follows:

$$\frac{\text{Dynamic Spring Rate}}{\text{Static Spring Rate}} = 1.3$$

$$\frac{\text{Stiffness at } -20^{\circ}\text{F}}{\text{Stiffness at } +70^{\circ}\text{F}} = 1.25^{*}$$

$$\frac{\text{Stiffness at } +140^{\circ}\text{F}}{\text{Stiffness at } +70^{\circ}\text{F}} = 0.9$$

Required is $C/C_c = 0.125$ minimum to limit T to 4 or less.

*For shock: Will be lower for vibration.

It should be noted that damping constant is dependent upon elastomer blending but that, in general, the stiffer the elastomer the higher the damping, and the lower the elongation capability.

7.2 MOUNT DESIGN

USING 60H DATA FROM GOODYEAR HANDBOOK

COMPRESSION

Deflection Inch	% Compression	Compression Stress PSI	Force Lbs.	Spring Rate Lbs./inch
.25	.07	43	894	3577
.50	.14	92	1913	3827
.75	.21	150	3120	4160
1.00	.28	240	4990	4990
1.25	.35	350	7280	5825
1.50	.42	510	10600	7070

SHEAR

d/t	Deflection Inch	S PSI	$k_s = \frac{SA}{t}$ Lbs./in.	Force/Defl. Lbs. @ In.	$\frac{\text{Comp. k}}{\text{Shear k}} = L$
.1	.356	140	818	205 @ .25	4.4
.2	.712	132	771	380 @ .50	5.0
.3	1.069	122	713	575 @ .75	5.4
				665 @ 1.0 est.	7.4
.4	1.424	113	660	940 @ 1.4	
.5	1.78	108	631	1123 @ 1.78	
.6	2.14	103	602	1288 @ 2.14	

7.2 MOUNT DESIGN

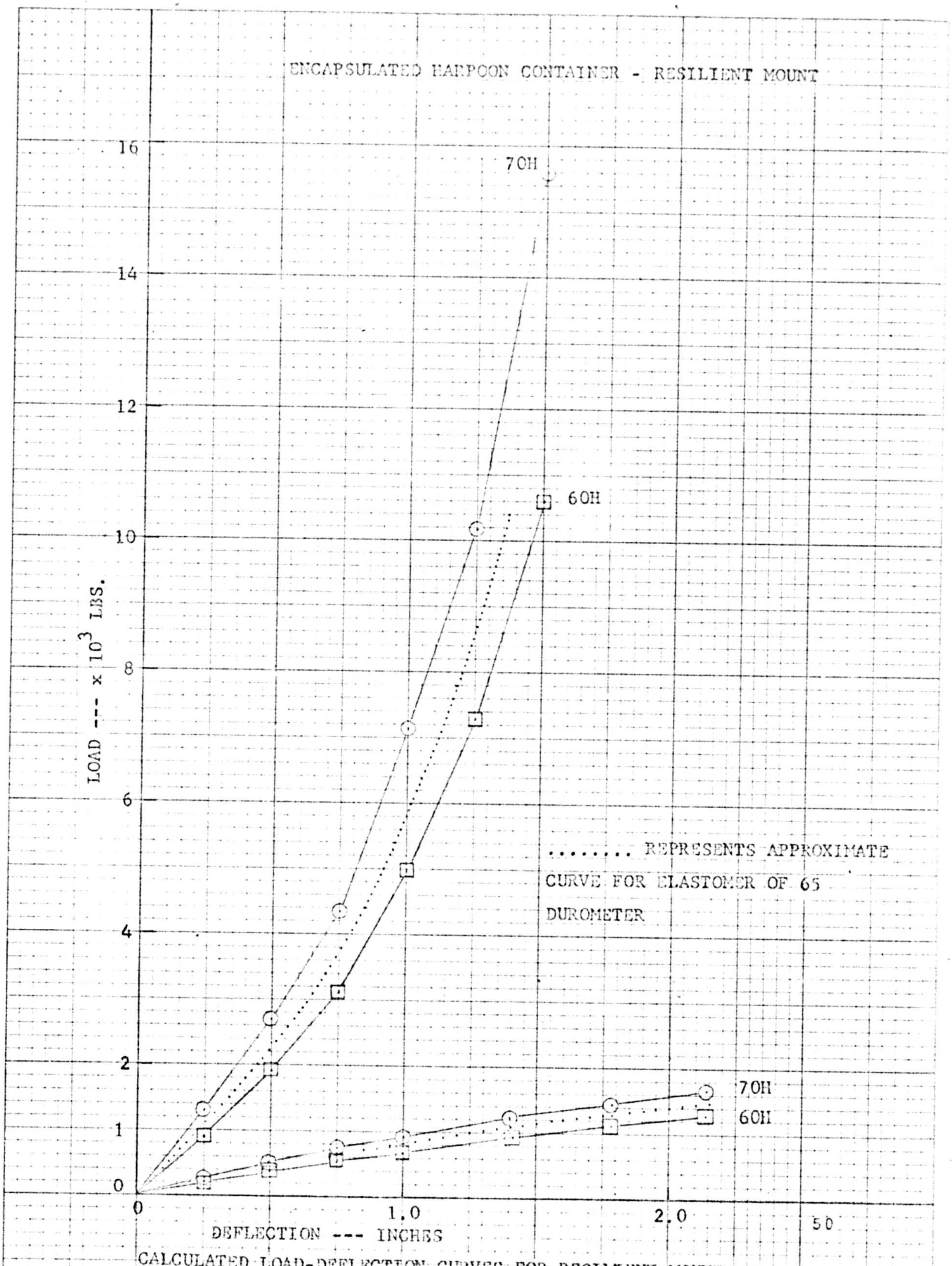
USING 70L DATA FROM GOODYEAR HANDBOOK

COMPRESSION

Deflection Inch	% Compression	Compression Stress PSI	Force Lbs.	Spring Rate Lbs./inch
.25	.07	60	1248	4992
.50	.14	130	2704	5408
.75	.21	210	4368	5824
1.00	.28	345	7176	7176
1.25	.35	490	10192	8154
1.50	.42	750	15600	10400

SHEAR

d/t	Deflection Inch	S PSI	$k_s = \frac{SA}{t}$ Lbs./in.	Force/Defl. Lbs. @ In.	$\frac{\text{Comp. k}}{\text{Shear k}} = L$
				283 @ .25	4.4
.1	.356	194	1133	403 @ .356	
				506 @ .50	5.3
.2	.712	173	1011	720 @ .71	
				758 @ .75	5.75
.3	1.069	158	923	987 @ 1.07	
				923 @ 1.0	7.77
.4	1.424	145	847	1206 @ 1.4	
.5	1.78	137	800	1425 @ 1.78	
.6	2.14	131	765	1637 @ 2.14	



CALCULATED LOAD-DEFLECTION CURVES FOR RESILIENT MOUNT

REF: Dwg. NWHL-7201

FIGURE 7

COMPRESSION/SHEAR MOUNTING

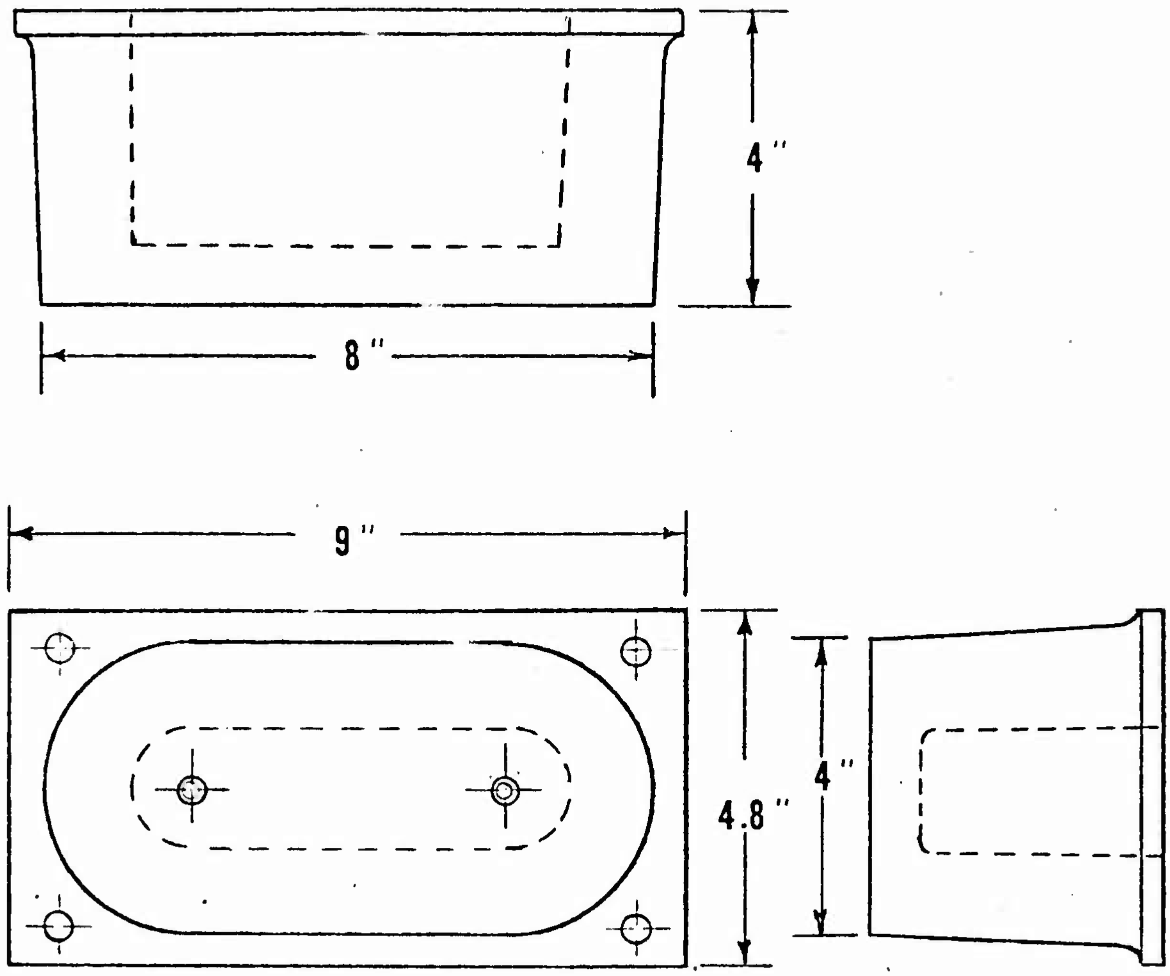


FIGURE 8

7.3 RESILIENT MOUNT DESIGN

Although it is anticipated that mounts will have adequate damping to limit transmissibility to 4, there is a possibility that a lower transmissibility might be desirable. To obtain this would require that a silicone elastomer be used in the mountings.

Silicones have lower tensile strength and lower elongation capability when used as an alternate elastomer in a particular configuration. Direct substitution is rarely possible unless the possible need has been anticipated.

Normal design limits for these materials are:

	Organic	Silicone
Static Shear Stress	25 psi	10 psi
Dynamic Shear Stress	250 psi	160 psi
Dynamic Shear Strain	250%	200%

End Impact @ 10 ft/sec @ +140°F.
Strain = 3.9 in. defl./3.62 in. thickness = 108%

$$\text{Stress} = \frac{3.7\text{ in. defl. (Ambient)} \times 966\text{ lbs/in.}}{20.8\text{ sq. in.}} = 172\text{ psi}$$

To limit stress to 160 psi entails increasing bond area to 22.4 sq. in. which is readily accomplished by a slight reduction in the width of the cavity in the mount.

After vibration tests have been run the effects of the highly-damped non-linear butyl mounts between HARPOON and Capsule can be considered. Also, the lower ratio of compression/shear spring rates' effect on lateral response can be considered: a small change in focal angle may be desirable.

7.4 NOTES AND CONCLUSIONS:

.....The HARPOON is supported within the Capsule on resiliently-mounted rails plus a precompressed 8-segment "Sabot" clamped to forward section of HARPOON. This Sabot support reduces bending by reducing the cantilevered load. Characteristics of the resilient Sabot are not known and the rail mounts are reportedly about 40 Hz natural frequency both vertically and laterally. All such internal mounts are compression-type elastomeric mounts which have non-linear spring characteristics after small initial deflection. Lack of data on these internal mounts, precluded any analysis taking these mounts' effects into account.

During testing phase it is anticipated that both Capsule and HARPOON will be instrumented. Data will be examined to determine if there are any adverse effects resulting from internal suspension and, if so, what action to take. As is usual practice, dynamic analysis assumes container resilient mounts to have linear characteristics. This is not quite true for either shear or compression loading: at small displacements in shear, the mounts' apparent shear rate is higher than for larger displacements. Conversely, in compression the spring rate increases as displacement increases. At the anticipated SHOCK displacement the ratio of compression spring rate to shear spring rate is calculated to be approximately 7:1. As displacement decreases, this ratio will also decrease. This is shown in the mount design data. Adjustment is possible by small movement of mounts outward and upward without change of the 45° focal angle. By reference to the curves relating L' to the focal angle and e/p ratio, keeping the focal angle fixed at 45°, it will be seen that if L' was as low as 5.5, the required e/p ratio becomes 0.69. Moving the mounts laterally out and up a distance of 0.40 inch results in e/p = 0.69. Thus, some adjustment is possible by a relatively simple change in location of mounts.

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17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

None Apply

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Container Dynamics
Reusable Container

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A dynamic and structural analysis of a proposed reusable Shipping and Storage Container for the Encapsulated HARPOON Missile was made by the Naval Weapons Handling Laboratory as part of the design study for such container.

The subject design incorporates a freebreathing fiberglass pod containing the encapsulated weapon. The pod and weapon are suspended from a truss-like

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered) outer structure by elastomeric mounts configured in a laterally focalized fashion.

The analysis generates isolator parameters which attenuate the handling and transportation shock and vibration environment to safe levels for the weapon and verifies that the structural design concept can sustain the resulting loads.