71-9144

PDR

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Dave Cramer
396SS

CHEM–NUCLEAR SYSTEMS INC.

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Charles E. MacDonald
Transportation Branch
Division of Fuel Cycle &
Material Safety
U. S. Nuclear Regulatory Commission
Washington D. C. 20555

Reference: Docket No. 71-9144

Dear Mr. MacDonald:

We are submitting herewith our reply to the various questions which we have discussed during recent telephone conversations. We trust that this submittal will permit you to make an early determination on our application for a Certificate of Compliance for the SGC-1 package. A revised list of effective pages is enclosed.

Please contact us if you have any questions in this matter. We will appreciate your prompt attention.

Sincerely,

CHEM-NUCLEAR SYSTEMS, INC.

Louis E Reynolds

Louis E. Reynolds
Director
Regulatory Affairs

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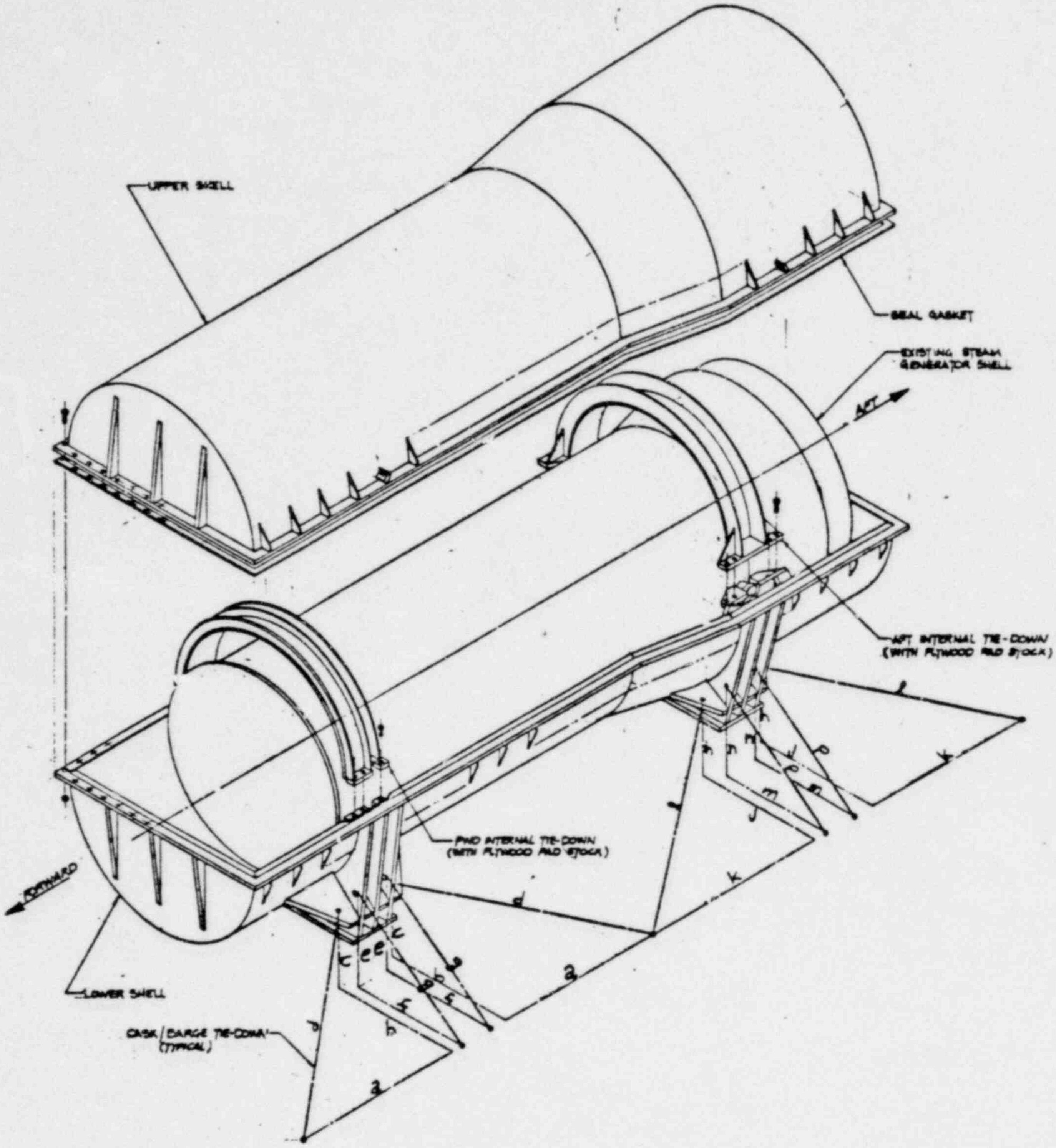

a = 121.625	e = 67.75	j = 92.375	m = 82.875
b = 91.75	f = 86.81	k = 121.625	p = 104.86
c = 52.56	g = 110.12	l = 161.91	
d = 161.16	h = 53.75	n = 64.25	

| 3

BARGE TIE DOWN SYSTEM

FIGURE 2.4.4-1

2.5 Standards for Type B and Large Quantity Packaging

2.5.1 Load Resistance

The cask if treated as a simple beam supported at its ends, can support a uniformly distributed load equal to five times its fully loaded weight.

Five times loaded weight = (5) 722,000
$= 3.61 \times 10^6$ lbs.

Length = 41 ft. 6 in. (excluding flanges) 3
= 498 in.

Load/in. = 7249 lb/in = w

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

For uniform load

$$M = \frac{WL^2}{8} = \frac{7249(494)^2}{8} = 2.247 \times 10^8$$

For conservatism assume the cask is a right cylinder of the small diameter.

$$I = \pi/64\ (D^4 - d^4) = \pi\, 1/64(149^4 - 144^4)$$
$$= 3.088 \times 10^6$$

$$\sigma = \frac{2.247 \times 10^8(74.3)}{3.088 \times 10^6} = 5422 \text{ psi}$$

$$MS = \frac{3800}{5422} - 1 = 6.0$$

2.5.2 External Pressure

The SGC-1 package will suffer no loss of contents if the package is subjected to an external pressure of 25 psig.

Hoop stress

$$\sigma_{\eta} = \frac{pr}{t}$$

Assume cylinder of the largest diameter found on the cask

D = 170 in

r = 85 in

$$\sigma_{\eta} = \frac{25(85)}{2.5} = 850 \text{ psi}$$

$$\text{M.S.} = \frac{38000}{850} - 1 = + \text{large}$$

The end plates can be modeled as a semi-circle with fixed edges.

From Roark and Young, "Formulas for Stress and Strain" 5th ed., McGraw-Hill, Table 24 page 371. 3

$$\sigma \text{ max.} = \frac{-0.42\ q\ a^2}{t^2}$$

q = pressure

a = radius

t = thickness

$$\sigma \text{ max.} = \frac{0.42(25)(85)^2}{2.75} = 27586 \text{ psi compression}$$

$$\text{M.S.} = \frac{38000}{27586} -1 = +.38$$

If considered one uniform plate

$$\sigma = \frac{6\ M}{t^2} \qquad M = \frac{q\ a^2}{8} = 22578 \text{ in.lbs.}$$

$$\sigma = \frac{6(22578)}{(2.75)^2} -1 = 2.12$$

The cask will adequately react 25 psig external pressure without loss of contents structurally.

The gasket is a full face flange gasket. The greater the external pressure on the shell the greater the gasket seating pressure and less likely the gasket is to leak.

3

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2.6 Normal Conditions of Transport

The Model SGC-1 packaging has been designed and the contents are so limited (described in Section 1.1.2 above) that the performance requirement specified in 10 CFR 71.35 will be met when the package is subjecte to the normal conditions of transport specified in Appendix A of 10 CFR 70 with the exception that the free drop will not be met due to the lack of applicability. The ability of the Model SGC-1 packaging to satisfaction withstand the normal conditions of transport has been assessed as described below:

2.6.1 Heat

The thermal evaluation for the analytical thermal model is reported in Section 3.4.

2.6.1.1 Summary of Pressures and Temperatures

With a maximum solar heat load in 130°F air, the external maximum temperature rose to 168.7°F and the internal temperature rose to 149°F with an internal pressure of 5.49 psig. These conditions had no detrimental effect on the package.

2.6.2 Cold

The materials of construction in this package as described in Section 2.3 have the ability to withstand a standard Charpy V-notch test with a minimum of 15 ft-lbs impact energy at -40°F. Based on that criteria, it is safe to conclude that cold will not substantially reduce the effectiveness of the package.

2.6.3 Pressure

A differential pressure of .5 atmosphere will be reacted by the cask and its closures. Loads on the closure bolts are calculated as follows: The longitudinal stress and maximum hoop stress in the cylinder are:

$f_h = PR/t = 14.7/2\ (85/2.5) = 249.90$ psi

$f_l = PR/2t = (14.7/2)\ 85/(2)2.5 = 124.95$ psi

Assuming these biaxial stresses are additive.

$f_{max} = 374.85$ psi

APPENDIX C

DYNAMIC RESPONSE OF THE MARINE TIEDOWN SYSTEM

The cask with its contents is secured to the trailer. This complete system is secured to the barge by 16 tiedowns as shown in figure C-1 and figure 1.3.3-10. To determine the time dynamic response of the system would require a detailed and complex model. An approximate response can be determined by looking at the system as a lumped mass system with three hypothetical springs representing the summation of the spring forces of the tiedowns.

This model is based on several assumptions:

1) The combined mass of the steam generator, the cask and the trailer can be lumped as a single mass. 3

2) The mass described in 1) is a point mass.

3) The barge beneath the trailer has the same spring coefficient as the vertical components of the tiedowns.

4) The tiedown components in the plane of the barge on opposite sides/ends of the cask act equally and opposite.

5) The order of magnitude of the natural frequency of the system is not affected by the coupling terms.

6) The tiedowns are considered massless in relation to the cask and trailer. (Each tiedown weighs less than 3000 lbs.)

7) The barge connections are considered rigid.

8) The relative position of the barge/tiedown connections are considered fixed in relation to one another.

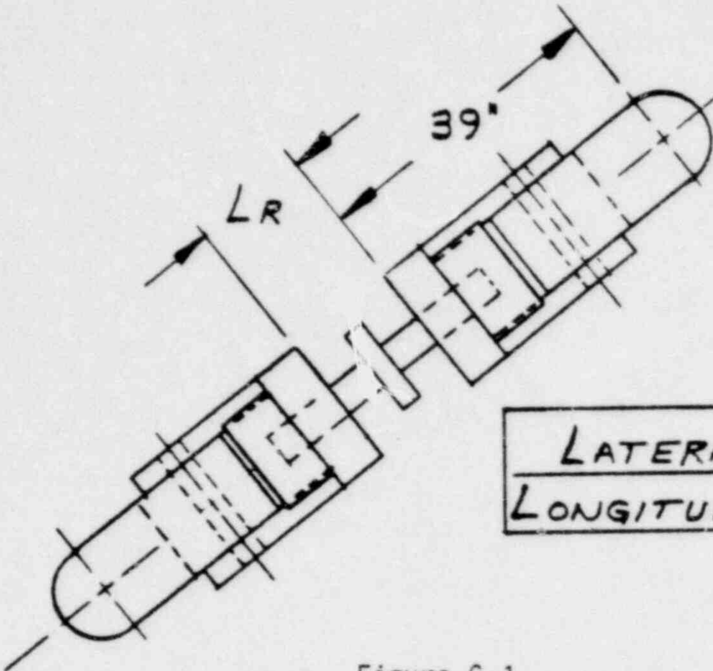
The above assumptions permit the system to be modeled as a mass and three springs as shown in figure C-2. This results in the following simplified equations of motion:

$$m\ddot{x} + k_x x = 0$$
$$m\ddot{y} + k_y y = 0$$
$$m\ddot{z} + k_z z = 0$$

Defining the terms:

M = combined mass of the steam generator, the cask and the trailer =

$$\frac{872{,}000\ \text{lbs.}}{(32.2)\ \text{ft/sec}^2} \times \frac{1}{12\ \text{in./ft.}} = 2.26 \times 10^3\ \text{lb in}^{-1}\ \text{sec}^2$$



	L_R
LATERAL	14.5
LONGITUDINAL	56

Figure C-1

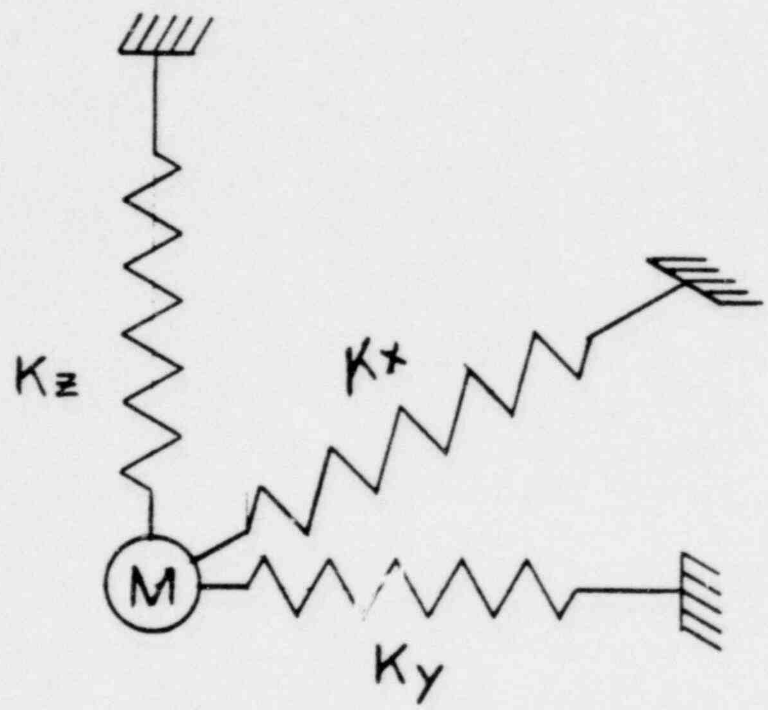


Figure C-2

3

For a bar $K = \frac{AE}{L}$

A = metal area
E = modulus of elasticity
L = length

The tiedown can be considered actually three springs, two end springs that are identical and a spring representing a take-up rod.

The ends are approximately 39 inches in length, with a metal surface area of about 60 square inches and a modulus of elasticity of about 29×10^6 psi. The rod varies in length from approximately 14.5 active inches in length for the lateral tiedown to about 56 active inches in length for the longitudinal tiedown. The threaded rod is 4¼ inches in diameter with a stress area of about 13.8 square inches. The modulus of elasticity for the rod is about 29×10^6 psi.

$$K_E = \frac{AE}{L_E} = \frac{(60)(29 \times 10^6)}{39} = 4.46 \times 10^7 \text{ 16/in.}$$

$$K_{R\,lateral} = \frac{A_R E_R}{L_{R\,lateral}} = \frac{(13.8)(29 \times 10^6)}{14.5} = 2.76 \times 10^7 \text{ lb/in.}$$

$$K_{R\,longitudinal} = \frac{A_R E_R}{L_{R\,longitudinal}} = \frac{(13.8)(29 \times 10^6)}{56} = 7.15 \times 10^6 \text{ lb/in.}$$

$$K_{lateral} = \frac{1}{\frac{1}{K_E} + \frac{1}{K_E} + \frac{1}{K_{R\,lateral}}} = 1.23 \times 10^7 \text{ lb/in.}$$

$$K_{longitudinal} = \frac{1}{\frac{1}{K_E} + \frac{1}{K_E} + \frac{1}{K_{R\,long.}}} = 5.41 \times 10^6 \text{ lb/in.}$$

From figure 2.4.4.1 the tiedowns can be resolved into components in the x,h,z directions. In the x and y directions, only one-half of the tiedowns are reacting since the tiedowns are hinged allowing only tensile loads.

$$K_x = K_{long.} = 4\ K_{long.} \frac{121.625}{161.16} = 1.63 \times 10^7 \text{ lb/in.}$$

$$K_y = K_{lat.} = 4\ K_{long.} \frac{91.75}{161.16} + 4\ K_{lat.} \frac{86.81}{110.12} = 5.11 \times 10^7 \text{ lb/in.}$$

3

In the vertical direction all tiedowns react upward, and the barge deck reacts the downward movement.

$$K_Z = K_{vertical} = 8\ K_{long.} \frac{52.56}{161.16} + 8\ K_{lat.} \frac{67.75}{116.12} = 7.08 \times 10^7 \text{ lb/in.}$$

From the uncoupled equations of motion:

$$\omega_1 = \sqrt{\frac{K_x}{m}} = \sqrt{\frac{1.63 \times 10^7}{2.26 \times 10^3}} = 84.93 \text{ rad/sec} = 13.52 \text{ cycles/sec}$$

$$\omega_2 = \sqrt{\frac{K_y}{m}} = \sqrt{\frac{5.05 \times 10^7}{2.26 \times 10^3}} = 149.48 \text{ rad/sec} = 23.79 \text{ cycles/sec}$$

$$\omega_3 = \sqrt{\frac{K_Z}{m}} = \sqrt{\frac{7.08 \times 10^7}{2.26 \times 10^3}} = 177.00 \text{ rad/sec} = 28.17 \text{ cycles/sec}$$

The forcing functions are the pitch, heave and roll of the barge. The period for the pitch and roll is about 20 seconds (ANSI 14 N 522.) This results in a frequency of .05 cycles/second. Another impulse force is that induced by slams. 3

From "A Study of the Single Voyage Risk Levels Associated with Extreme Motion Values for [illegible] Voyage from Norfolk, Virginia, to Astoria, Oregon," L.R. Glosten & Associates, Inc., the maximum slam frequency is 86 slams per hour. This results in [illegible] lower frequency than the other periodic loadings.

The magnification factor that results from the forcing function is:

$$MF = \frac{1}{1-\left(\frac{W}{W_n}\right)^2}$$

$$MF = \frac{1}{1-\left(\frac{.05}{13.52}\right)^2} = 1$$

The natural frequency of the system is considerably higher than any of the forcing frequencies.

It is recognized that the values given above are not the true natural frequencies of the various modes of the system. However, the values given represent relative orders of magnitude of the system. Even if the true natural frequency of the system was two orders of magnitude less than the calculated, the magnification factor would be on the order of 15%.

The damping of the system is large. The assumption that the barge is rigid is very conservative because the barge flexture will dampen the system. Also, any motion of the barge is dampened by the sea. However, if one can assume the barge to be rigid in relation to the tiedowns, the cask itself is further dampened by friction. Treating the cask as a body with Colamb dampening due to friction, the minimum friction force which prevents vibration at resonance is:

$$F_f = .79\ Ku_o$$

where K = spring constant in the direction of motion

u_o = displacement in

C.M. Harris & C.E. Crede, Shock and Vibration Handbook, 2nd ed. McGraw Hill, 1976. Page 30-8.

On the deck of the barge the cask generates the following frictional force, assuming the coefficient of friction between wood and steel to be equal to .4.

$$F_f = 872{,}000\ (.4) = 348{,}800 \text{ lbs.}$$

This gives a maximum displacement of

$$u_o = \frac{348{,}800}{(.79)(5.11 \times 10^7)} = .01 \text{ in.}$$

Assuming that the majority of the elongation occurs in the tie rod, the required loading can be calculated for the longitudinal or longest tiedown. From section 2.4.4.1 page 2-22

$$u_o = \frac{PL}{A_E} = \frac{435310\ (56)}{13.8\ 29 \times 10^6} = .06 \text{ in.}$$

for 1.5_g loading.

The .01 in displacement is equivalent to:

$$P_g = \frac{1.5\ (0.01)}{.06} = .25 \text{ g loading.}$$

From the L.R. Glosten report it is evident that the high accelerations are infrequent and random. The damping of the system would prevent resonance amplification of the higher frequency loadings.

In conclusion, the rigidity of the tiedowns create a considerably higher natural frequency for the system than the expected forcing frequencies. The system as a whole is dampened so that only the higher accelerations will cause vibration.