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Mr. Charles E. MacDonald, Chief
Transportation Certification Branch
Division of Fuel Cycle and Material Safety
Nuclear Regulatory Commission
Washington, D.C. 20555

Subject: Amendment to Certificate of Compliance Application, Docket No. 71-9153

Gentlemen:

Teledyne Energy Systems herewith submits an amendment to our license application, TES-3147, "Sentinel 1S Radiation, Structural and Thermal Evaluation." The enclosed ten packages of seven pages each, replace pages 2-22 through 2-24 of Section 2.7.1 "Free Drop," submitted previously.

Sincerely,

R. G. Hannah
Program Manager
Terrestrial Power Systems

dw

Enclosures: 10

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SUBJECT: DISTRIBUTION OF CHANGE - PAGES 2-22 thru 2-24.d

SENTINEL IS

RADIATION, STRUCTURAL AND THERMAL EVALUATION

TES-3147

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An impact of the generator on its side is similar to the case above where some of the available energy is converted to strain energy of the Min-K insulation. In reality the cooling fins will attain an instability and absorb significant strain energy in bending. For this analysis the latter energy dissipator will be ignored. It can be shown that for this orientation, the minimum crush-effective volume of insulation is about 27.3 $in.^3$. At 9.5 in-lbs/$in.^3$ the strain energy is 259 in-lbs.

A similar situation exists for an impact on the top or upper end of the generator with the exception that there is no effective volume of insulation for that direction. For this orientation, the material between the housing and shield is the thermoelectric module, a relatively brittle semiconductor alloy providing negligible strain energy. For both the side and top impact there is no concern for the shield lid bolts. From Figure 2.7-4, it is shown that the shield lid protrudes well into the shield cavity such that bolt deformation is limited in the shear plane corresponding to the side impact orientation. For the upper end impact the bolts are not loaded beyond their initial torque.

The radiation shield, fabricated from uranium - 3/4% titanium, is evaluated for direct impact response for two attitudes, side-on and top end-on. For both orientations energy is absorbed in bending of the cooling fins that are integral with the housing that surrounds the shield. It can be shown that the energy absorbed in bending is approximately 4000 in-lbs in each direction. However, this magnitude is insufficient since the available energy associated with a 50 lb shield and fuel capsule dropping 30 feet is,

$$E_a = 50\,(30)\,(12) = 18{,}000 \text{ in-lbs}$$

Conservatively assuming that the shield impacts a rigid surface with no loss of energy due to surrounding structure, a shock rise time is derived and the resulting stresses are obtained. From the reference cited below,* the peak deceleration is obtained from the equation,

$$G = \frac{72}{t} \sqrt{h}$$

where: G = deceleration, g's

t = shock rise time, milliseconds

h = drop height, inches

For an impact on a flat face the shock rise time for rigid steel against concrete is one (1) ms. This value should be slightly larger when impacting U - 3/4 Ti since its modulus is 26×10^6 psi versus 29×10^6 psi for steel. It is also important to note that an impact against an edge or point the shock rise time is greater which implies lower deceleration. For $t = 1$ ms,

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* Reference 6: "Design for Shock Resistance" by R. T. Magner, Product Engineering 1962.

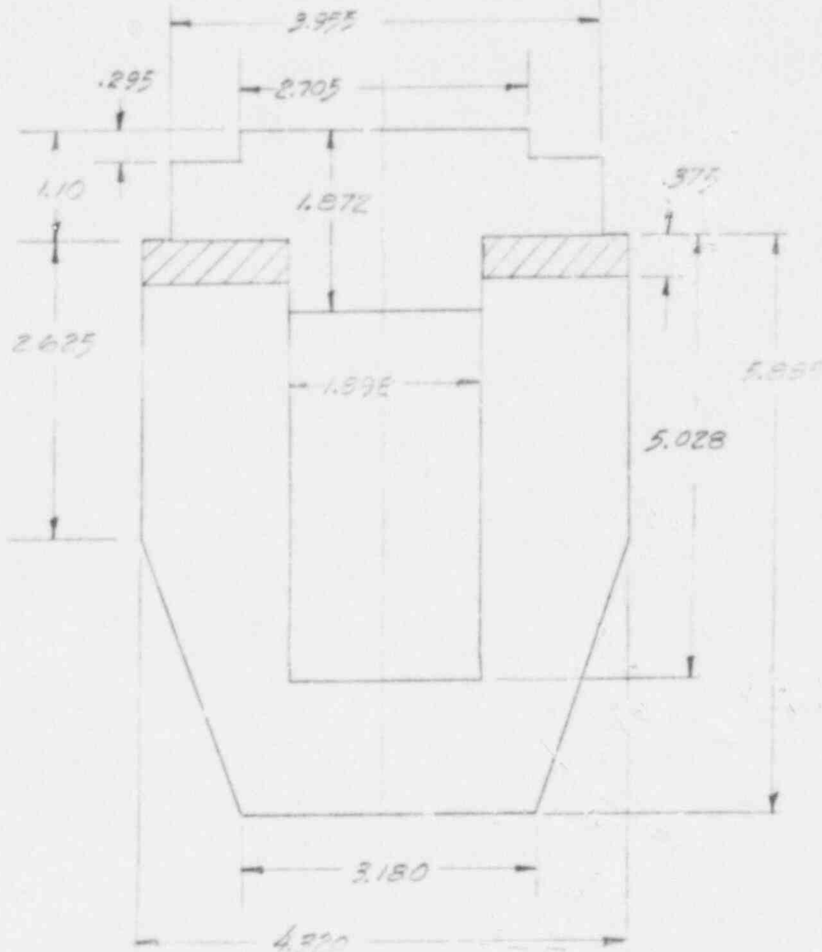


FIGURE 2.7-4
SENTINEL IS SHIELD GEOMETRY

$$g = \frac{72}{1} \sqrt{30 (12)} = 1366 \text{ g's}$$

For the end-on (top) impact the maximum stresses can be easily derived.

At the upper surface of the lid, the inertia force is,

$$P = 1366 (2.85 + 30.98 + 1.29 + 9.25)$$

$$= 68,805 \text{ lbs}$$

$$A = \frac{\pi}{4} (2.705)^2 = 5.747 \text{ in}^2$$

$$\therefore \sigma = \frac{68805}{5.747} = 11,972 \text{ psi}$$

The stainless steel ring has a minimum yield strength of 30,000 psi. The minimum area is,

$$A = \frac{\pi}{4} \left[(3.955)^2 - (1.898)^2 \right] = 9.456 \text{ in}^2$$

The inertia force is 1366 (36.98 + 1.29) = 52277 lbs. Therefore, the stress is,

$$\sigma = \frac{52277}{9.456} = 5528 \text{ psi}$$

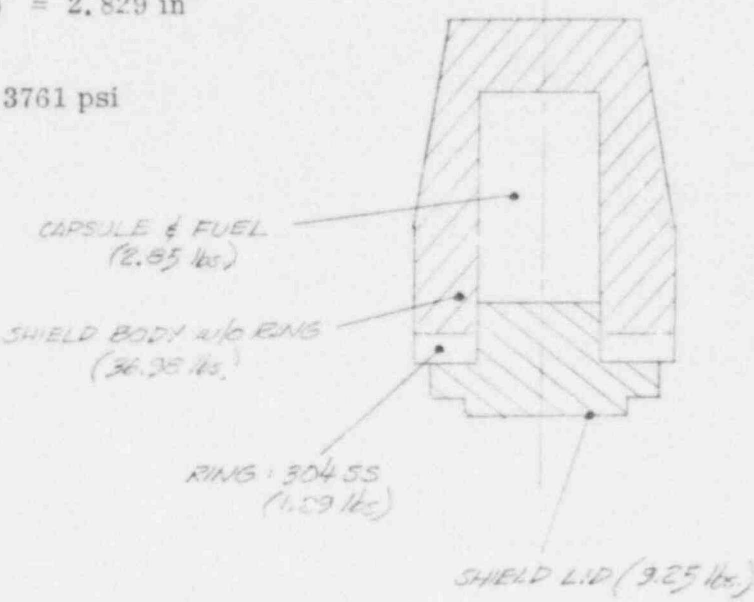
For the side-on impact the potential failure of the lid is in shear. The effective weight is,

$$W = .672 \left[\frac{\pi}{4} (2.705)^2 (.295) + \frac{\pi}{4} (3.955)^2 (1.10 - .295) \right] = 7.79$$

$$P = 1366 (7.79) = 10641 \text{ lbs}$$

$$A_s = \frac{\pi}{4} (1.898)^2 = 2.829 \text{ in}^2$$

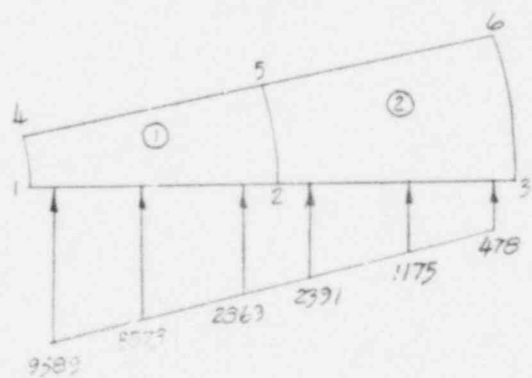
$$\Rightarrow \tau = \frac{10641}{2.829} = 3761 \text{ psi}$$



The inertia forces in the shield body are approximated by examining a section of unit length. The ANSYS finite element program was utilized where the thick walled circular section was modeled by 72 elements as shown in Figure 2.7-5. Each element is an isoparametric solid defined by four nodal points having two degrees of freedom at each node point, translations in the x and y directions. The tractions are based upon 1366 g's acting on the mass of each element in the -y direction. The total force is reacted as a line load at node 84. The resulting stresses are conservative since the constraints provided by the lid and lower end closure are neglected. The material properties of interest for the U-3/4% Ti include $E = 26.0 \times 10^6$ psi, $\nu = 0.3$ and $\rho = .672 \text{ lbs/in}^3$.

The stress distributions of interest are located at $\Theta = 0$, 90 and 270°.

a. At 0° (elements 1 and 2) the stress distribution across the wall is the superposition of bending and compression as shown below.



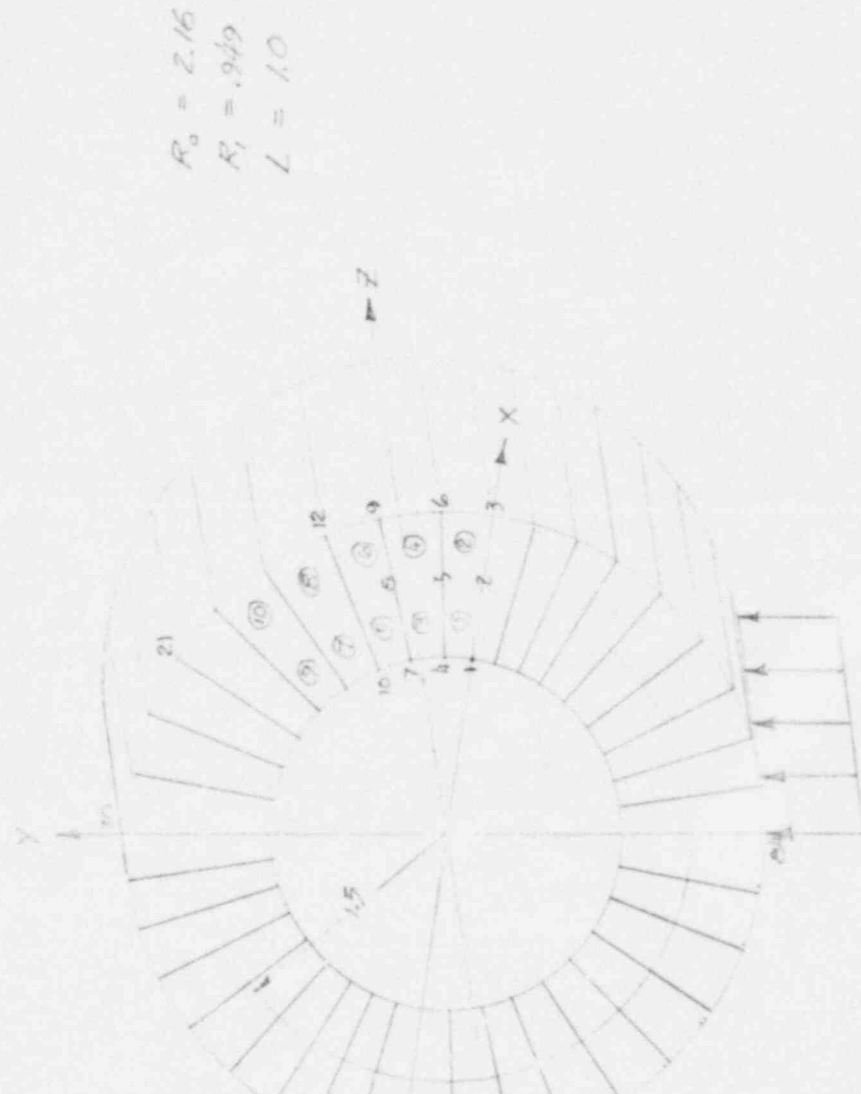


FIGURE 2.7-5
FINITE ELEMENT MODEL
- CENTRAL SECTION OF SHIELD -

b. The maximum tensile stress occurs at R = .949 inch and $\Theta = 90°$. The value is C079 psi.

c. The maximum stress occurs at the line of contact with the impacting surface (node 84) of element 54. At this location there is a compressive stress (-13,091 psi) due to bending, a compressive stress in the y-direction (-19,735 psi) and a shear component. The resulting equivalent stress is 26,831 psi.

The stresses derived, however, are a result of an equivalent static loading. The loads are not static but realistically are a result of a time dependent impulse. The actual response is a function of the shock period and the fundamental frequency of the body receiving the impulse. The shock pulse can be represented by a half-sine wave. However, although the shock rise time and the amplitude are known, the total duration of the pulse is not known. For this reason, the maximum potential amplification factor of 1.78 is applied. Table 2.7-1 presents the "static" stresses derived, the "dynamic" stresses and the latter is compared to the yield stress of the material of interest.

The shield in a structural sense is essentially a solid block. That is, the volume of the cavity receiving the fuel capsule and fuel is very small relative to the volume of actual material. The effective volume is 86%. In every way, the shield must be considered "thick walled." Without analysis the only conceivable plastic deformation envisioned would be a slight coining of an exterior edge.

2.7.2 Puncture

The second hypothetical accident condition in the sequence pertains to a free drop of 40 inches onto a stationary and vertical mild steel bar of 6 inches diameter with its top edge rounded to a radius of not more than 1/4 inch. As in the previous analysis, for the 30-foot drop, the applicable configuration addressed includes the shield and the encapsulated fuel. The structural capability of the cask is ignored. Since the mild steel bar diameter is nearly equal to any dimension (i.e., length, diameter, etc.) of the shield, potential penetration is impossible. The response would be identical to the impact from a 30-foot drop with the exception that the available energy would be considerably lower.

TABLE 2.7-1

COMPARISON OF CALCULATED STRESSES WITH YIELD AND ULTIMATE STRENGTHS

Impact Orientation	Stress Location	Static Stress (psi)	Maximum Dynamic Stress (psi)	Yield Strength (psi)	Ultimate Strength (psi)
Top	Upper surface of lid	-11972	21310	50000	128000
Top	Stainless steel ring	- 5528	9840	30000	75000
Side	Shear of lid	3761	6695	--	76800*
Side	Side	26831	47759	50000	128000

* The ultimate shear stress is approximated as 60% of the ultimate tensile stress.