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BENDIX SYSTEMS DIVISION ANN ARBOR, MICH.

Band Mechanism for Cask Retention

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This ATM outlines the structural considerations pertinent to the band mechanism which supports the graphite cask.

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I. Introduction

The purpose of this memo is to discuss the structural aspects of the mechanism which has been designed to constrain the smooth graphite cask, and to discuss its interaction with the cask itself. Because the graphite used for cask construction is a brittle material whose strength is far less than that of common structural metals, the design of the cask constraint mechanism has been predicated on minimizing the stresses induced in the cask. These stresses arise from vibrational loading which occurs during boost; from thermal loading by radioactive heating, from clamping forces inherent in the support mechanism and from tolerance buildups which necessarily occur in machine assemblies.

The operational environment of the cask support mechanism includes a temperature range from earth ambient to 800° F. The band tightening operation will occur at a temperature below 550° F, while the maximum loading of 60 earth g's in any direction occurs during the first 2.5 minutes of flight, at which time the band temperature is 550° F or higher. It is assumed that the radiation levels are not high enough to cause appreciable hardening of material or other changes in mechanical properties and that all rubbing mechanisms are lubricated to prevent unduly high friction coefficients or cold welding.

II. Assumed Properties and Failure Mode of Cask

The graphite cask material is assumed to have properties which are independent of both temperature and direction. These properties are as follows:

Tensile strength	5.22 KSI
Compressive strength	12.00 KSI
Shear strength	1.26 KSI
Modulus of Elasticity	1.50×10^6 PSI
Coefficient of Thermal Expansion	1.775×10^{-6} in/in/° F

The assumption is made that the material is isotropic so that it can be characterized by a single modulus of elasticity (E). Accordingly, commonly derived formulas for the behavior of shells whose derivations involve this assumption are valid.

The shaded triangle of Fig. 1 represents any typical element within the tensile member. Equilibrium dictates that the shear stress on the 45° plane be equal to one half of the maximum tensile stress. Since the tensile strength of the material is given in Table 1 as 5.22 ksi, it necessarily follows that the 45° plane has a shear strength equal to or greater than 2.61 ksi. On planes parallel to the shell surface, i.e. parallel to the graphite yarn, the shear strength value of 1.26 ksi applies. Therefore, different criteria for shear failure are used, depending upon the surface involved.

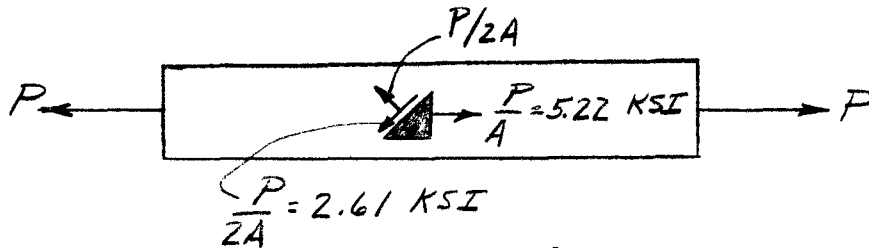


Figure 1

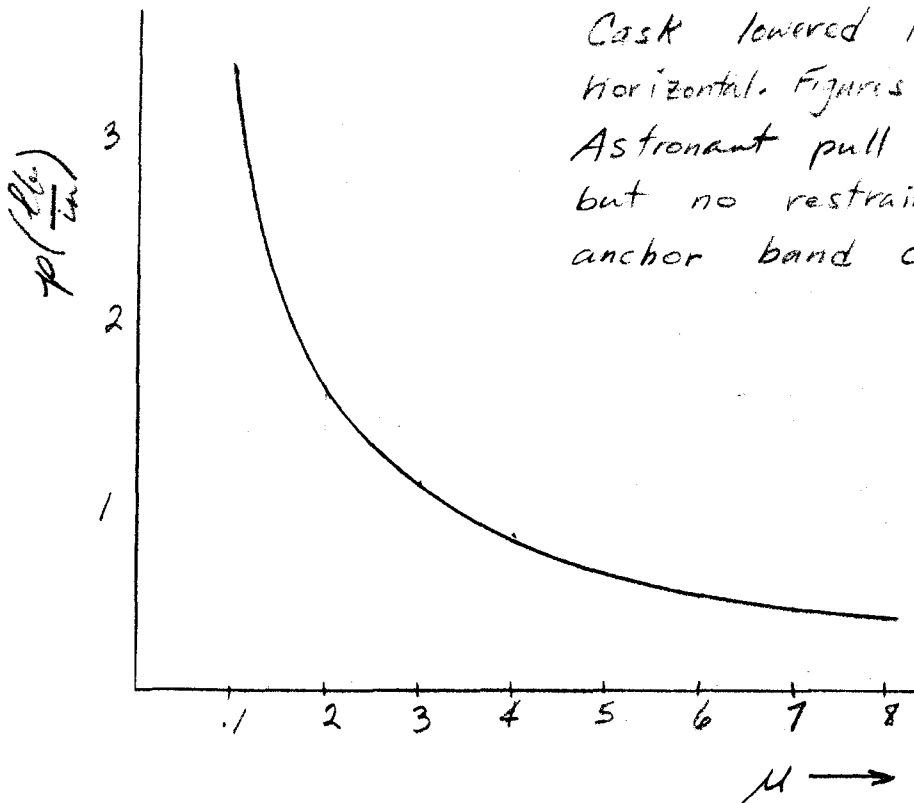


Fig 2 Required Clamping Load For Cask Retention, vs. Friction Coefficient



III Selected Band Configuration

One requirement of the cask band mechanism is that it operate over a temperature range from 70° F (earth ambient) to 800° F. Another requirement is that it retain some residual clamping pressure at 800° F so that after the astronaut tilts the cask for the fuel element removal, and after the axial band is released, the clamping pressure of the circumferential bands can prevent the cask from falling to the ground. This cask retention system is in addition to (i. e. redundant with) three lugs which fasten the lower anchor band to the lower cap. See Figure 2 for a plot of circumferential line load required of each band versus friction coefficient between cask and bands in lunar environment. In order for a residual clamping pressure of some magnitude to exist at 800° F, it is necessary to pretension the bands around the cask. During the temperature rise from the time the band is tightened to lunar operation, the bands expand more rapidly than the cask, relieving some or all of the initial clamping forces. In order to calculate the clamping forces required at earth ambient, the ring-cask interaction is approximated by two thin hoops as shown in Figure 3.

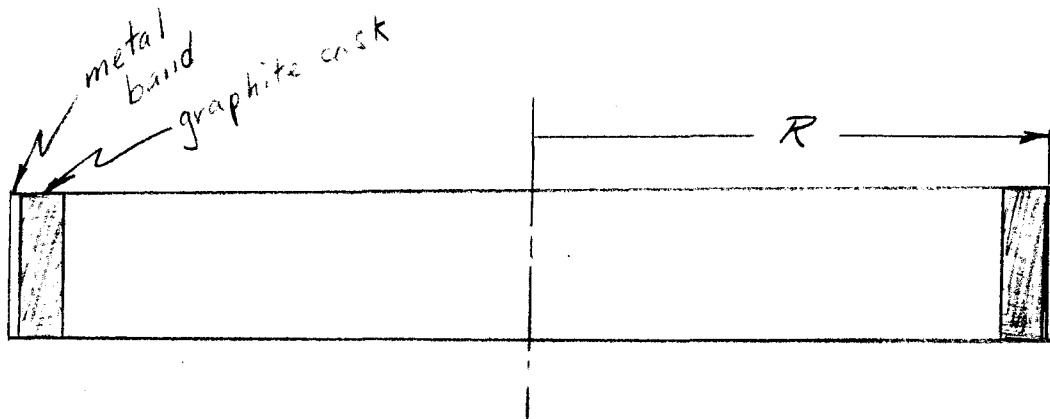


Figure 3 - Band-Cask Cross Section



The circumferential elongation of each material due to temperature change alone is:

$$e''_M = 2\pi R \alpha_M \Delta T \quad (\text{Eq. 1})$$

$$e''_G = 2\pi R \alpha_G \Delta T \quad (\text{Eq. 2})$$

where the subscript M is for metal, G for graphite. The circumferential elongation due to circumferential load P is:

$$e''_M = \frac{P_M (2\pi R)}{A_M E_M} \quad (\text{Eq. 3})$$

$$\text{and } e''_G = \frac{P_G (2\pi R)}{A_G E_G} \quad (\text{Eq. 4})$$

Using the free body diagram of Figure 4, and summing forces in the Y direction;
 $-P_M = P_G$.

Adding strains

$$e_M = e'_M + e''_M \quad (\text{Eq. 5})$$

$$\text{and } e_G = e'_G + e''_G \quad (\text{Eq. 6})$$

As long as the band remains in contact with the graphite,

$$e_M = e_G \quad (\text{Eq. 7})$$

Eqs 1 thru 7 yield:

$$\alpha_M \Delta T + \frac{P_M}{A_M E_M} = \alpha_G \Delta T + \frac{P_G}{A_G E_G} \quad (\text{Eq. 8})$$

$$\alpha_M \Delta T + \frac{P_M}{A_M E_M} = \alpha_G \Delta T - \frac{P_M}{A_G E_G} \quad (\text{Eq. 9})$$

$$\therefore P_M \left(\frac{1}{A_M E_M} + \frac{1}{A_G E_G} \right) = (\alpha_G - \alpha_M) \Delta T \quad (\text{eq. 10})$$

For a perfectly rigid metal ring, let $A_M E_M \rightarrow \infty$ so that $\frac{1}{A_M E_M} \rightarrow 0$ and:

$$\therefore P_M = A_G E_G (\alpha_G - \alpha_M) \Delta T \quad (\text{Eq. 11})$$

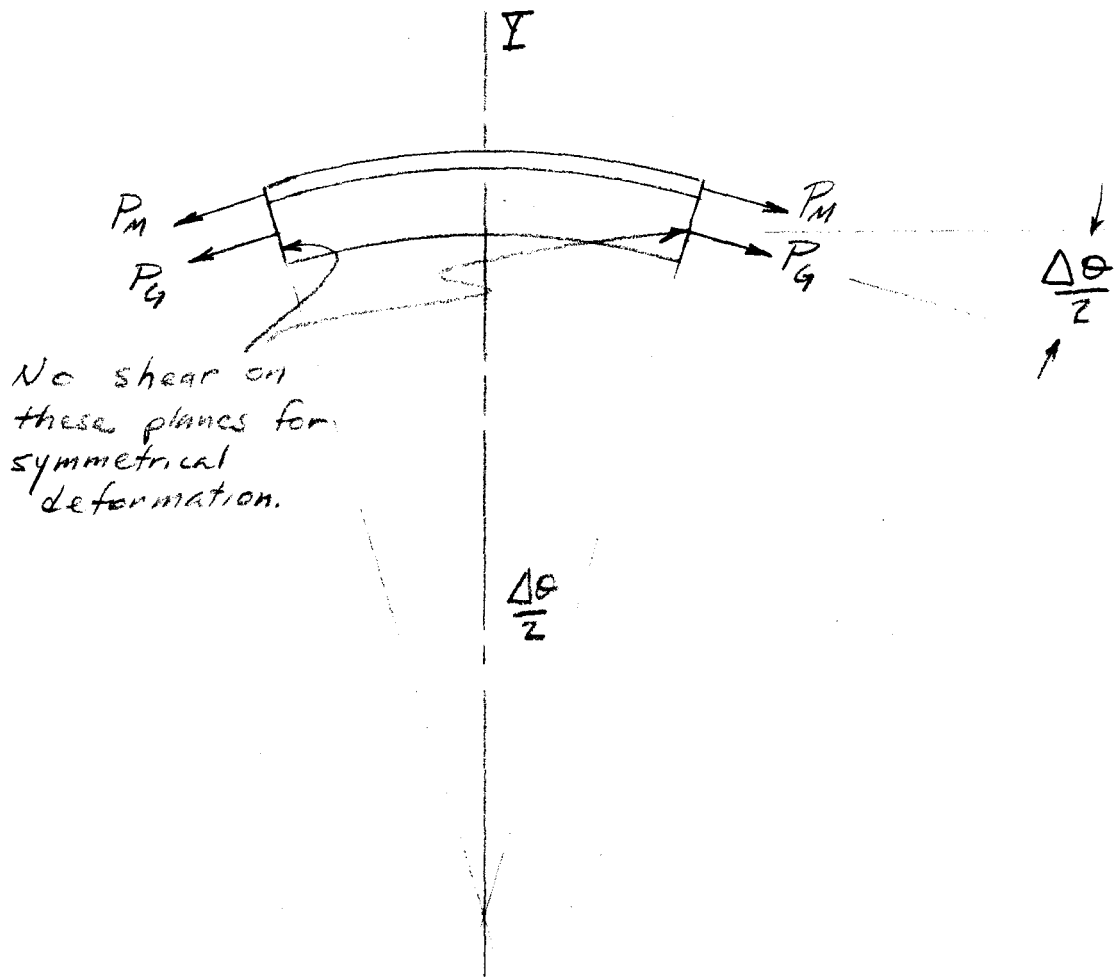


Fig 4 FBD of Cask-Band Element



using $\alpha_M = 6 \times 10^{-6}$ and $\Delta T = 730^\circ\text{F}$:

$$P_M = (.87) (1.5 \times 10^6) (1.775 - 6.0) (10^{-6}) (730) \quad (\text{Eq. 12})$$
$$= -4020 \text{ lbs}$$

The negative sign means that the tensile load P_M increases as the temperature decreases. The absolute value of the load is used in succeeding calculations.

To determine the line load that is produced on the cask, consider the free body diagram, Figure 5. See Page 7.

$$\sum F_Y = 0:$$

$$2P \sin \frac{\Delta\theta}{2} = pR \Delta\theta \quad (\text{Eq. 13})$$

Since $\Delta\theta$ can be taken arbitrarily small

$$2P \frac{\Delta\theta}{2} = pR \Delta\theta \quad (\text{Eq. 14})$$

$$p = \frac{P}{R} \quad (\text{Eq. 15})$$

The line load resulting from the P_M of Eq. 12 is therefore

$$p = \frac{4020}{4} = 1005 \frac{\text{lb}}{\text{in}} \quad (\text{Eq. 16})$$

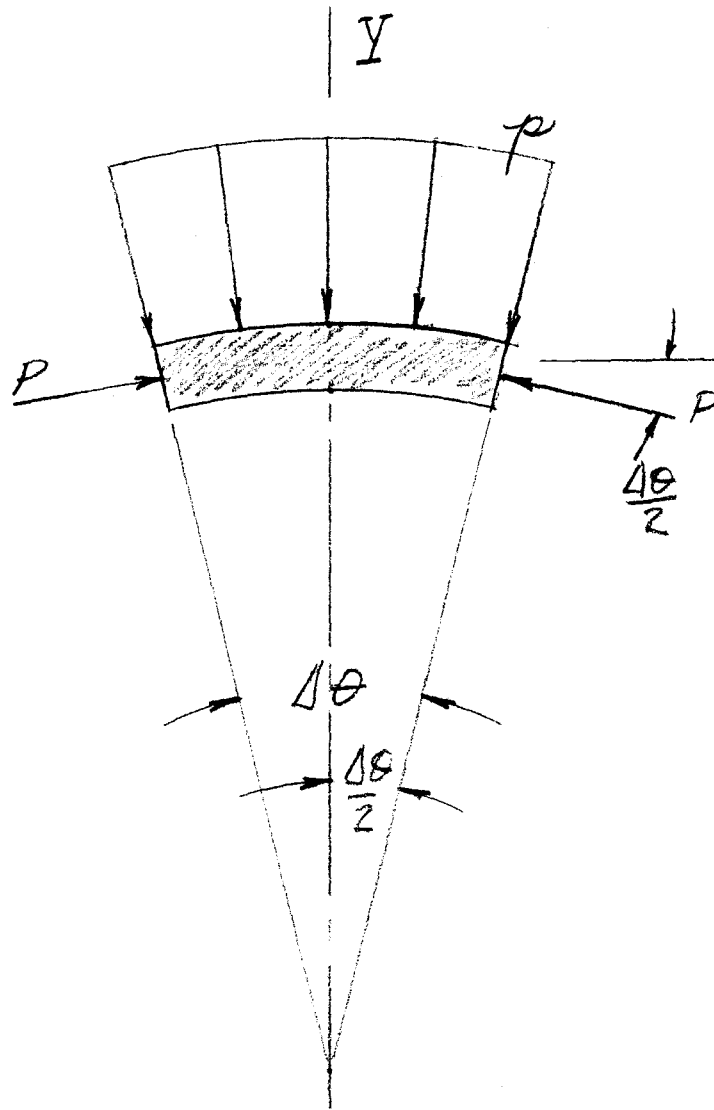


Fig. 5. FBD of cask element.



The p of Eq. 16 is the line load which would have to be applied to the cask by a rigid metal structure at 70° F in order to maintain zero clearance at 800° F. This value of p is an order of magnitude greater than the desired line load. Reduction of p can be accomplished by using a band material whose coefficient of expansion is closer to that of graphite and whose product $A_M E_M$ is low (See Eq. 10). In order to achieve a low value of A_M it is necessary to choose a material whose tensile strength is high, so that a small area can resist a high "g" load. Table 2 lists the pertinent properties of a number of materials which have been considered.

Material	$E \times 10^{-6}$	$\alpha \times 10^6$	Ductility	F_{tu} KSI
Graphite	1.5	1.775	0	5.22
Tantalum	27	3.6	95%	60
Tungsten	59	2.5	0	220
TZM	46	2.7	55%	125
S glass	10.5	1.6	0	665
Titanium 6A14V	16	5.3	10%	160
Stainless Steel 17-7PH (H1050)	29	6.3	9%	200

Table 2

MATERIAL PROPERTIES

Three of the materials of Table 1 - tungsten, S glass and graphite have α values equal to or nearly equal to the graphite of the cask. Thermal expansion stresses associated with their use are therefore zero or quite small. However their complete lack of ductility makes them extremely sensitive to scratches, and to stress concentrations at their attachment points. Therefore it was decided to avoid these brittle materials on a reliability basis. Tantalum and TZM (a molybdenum alloy) are all attractive from the expansion coefficient standpoint. Both however would have a high product of Young's modulus and cross sectional area (EA) which is undesirable. In addition both metals have high density, high cost, and a high melting point. A low melting point material is desirable per specification IC 314121.

The above considerations narrow down the band material choice to titanium and to stainless steel. Titanium is superior to stainless steel in terms of both its expansion coefficient and product EA. In addition it has a lower weight density. Its only disadvantage is its melting point which is approximately 300° F higher than the stainless.

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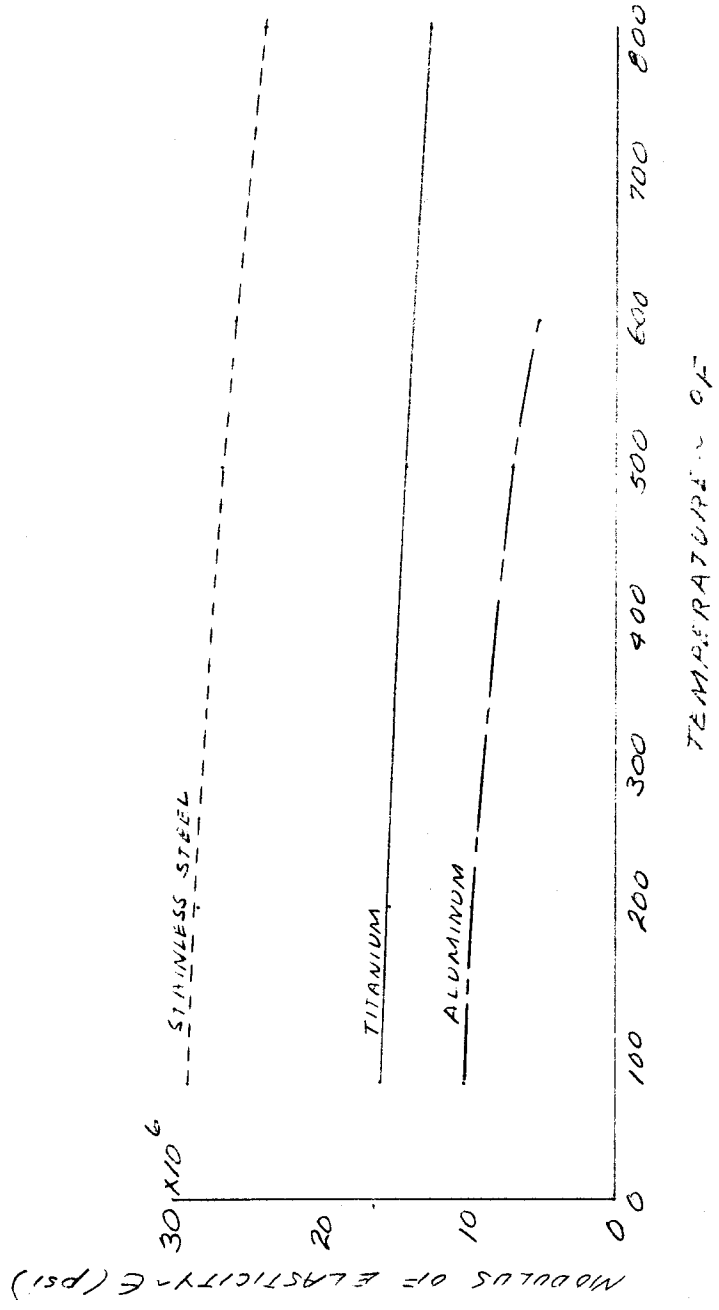


Figure b. E vs. Temperature



On the basis of the above considerations, titanium has been chosen as the most desirable band material. It has low enough values of EA and α to keep expansion stresses to a reasonable level; it has high strength and ductility; and in addition it has low weight density.

Two alloys of titanium could be used with almost equal efficiency. These are Ti 8Al 1Mo 1V and Ti 6Al 4V. The former alloy has superior creep properties but is slightly inferior from a strength standpoint. Reactive Metals, Inc. estimates creep of the bands to be less than .00001 in/in for our application. This deformation would cause a reduction in line load on the cask of less than .65 lb/in. These magnitudes are so low that creep is not considered an important consideration. Therefore the stronger alloy, Ti 6Al 4V, is selected for the bands.

The clamping stress required of a .017" thick by 1" wide band of Ti 6Al 4V for a temperature change can be obtained from equation 10:

$$P_M \left\{ \frac{1}{.017 (16 \times 10^{-6})^*} + \frac{1}{.87(1.5 \times 10^6)} \right\} = (1.775 - 5.3) (10^{-6}) (250) \quad (\text{Eq. 17})$$

$$P_M \left\{ \frac{1}{.272} + \frac{1}{1.305} \right\} = -3.525 (250)$$

$$P_M (3.68 + .0766) = -882$$

$$P_M = \frac{-882}{3.7566} = -235 \text{ lb.}$$

From Eq. 15:

$$p = \frac{235}{4} = 58.7 \text{ lb/in.} \quad (\text{Eq. 18})$$

This clamping load, Eq. 18, is small enough that special corrugated (i. e. spring loaded) bands which had been conceived are not necessary for cask retention. That type of band would have weight and cost disadvantages and could result in increased trunnion loading on the cask.

Another special consideration is the possibility of chemical interaction between the band material and the coating to be used on the graphite cask. At the present time, however, no coating has been specified.

IV Trunnion Design

The bands which were discussed in the previous section attach to four trunnion blocks which provide bearing areas for the cask release (top) and the cask rotation mechanism (bottom). These trunnions carry inertia and gravitational loads from the cask to the truss structure. A simplified sketch of a trunnion mount is shown in Figure 7.

* This is Young's Modulus at room temperature. A somewhat lower value could be justified. See Figure 6.

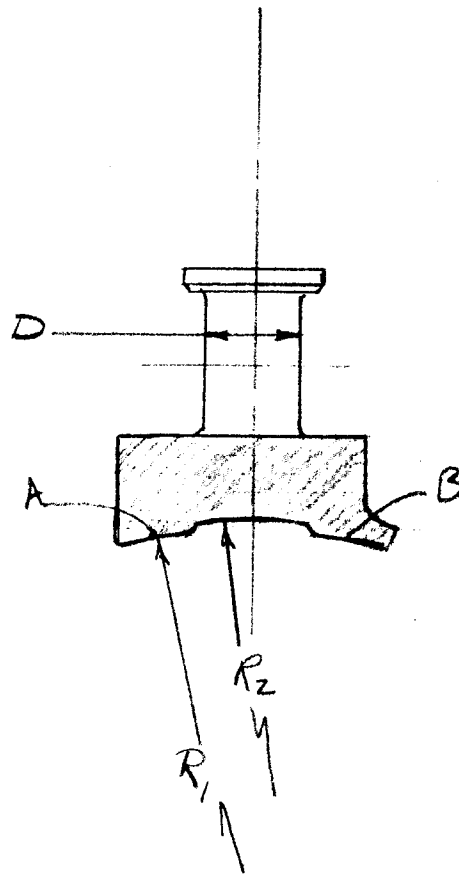


FIGURE 7: UPPER TRUNNION BLOCK



In order to minimize the stresses in the graphite cask, the following design considerations have been made. (Refer to Figure 7.)

1. Minimize angularity tolerance between trunnion block and U member support. A perpendicularity tolerance difference of .001 in/in is considered practical.
2. The trunnion is relieved by radius R_2 as shown. This precludes reaction of the cask against the trunnion at a single point. Experiments* have shown that local bending due to a concentrated load on a cylindrical shell completely decays through an angle 12° away from the point of application. Therefore, we desire that "A" and "B" be approximately 12° apart. Smaller angles result in superposition of bending stresses. Large angles allow the trunnion mounts to react to transverse loads that should be taken by band tension to result in minimum cask loads.
3. The radius R_1 should be nearly identical to the outside radius of the graphite cask in order to minimize contact stresses. Contact stresses are more important for the brittle graphite than for a ductile metal which can yield to adjust to local loading. Accordingly, the maximum difference between the cask radius and the trunnion radius R_2 is limited to .0025 inches.

*Reference - Roark, R. J.; The Strength and Stiffness of Cylindrical Shells Under Concentrated Loading; Am. Soc. Mech. Eng.; Jour. Appl. Mech; Vol. 2, No. 4, p. A-147; 1935.