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STRESS ANALYSIS: NRX-A OUTER

REFLECTOR SUPPORT RINGS

(Title Unclassified)

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Astronuclear Laboratory
Westinghouse Electric Corporation

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WANL-TME-1005
October 1964

SUBMITTED BY

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STRESS ANALYSIS: OUTER REFLECTOR
SUPPORT RINGS

(Title Unclassified)

Classification cancelled for changed to
by authority of DOC
W.A.C. TIC, date SEP 10 1973

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SPECIAL REVIEW
FINAL
DETERMINATION
Class: U
Reviewer JRP Date 5-4-82

INFORMATION CATEGORY:

CRD

McB... 10/26/64
Authorized Classifier Date

Group

Excluded From Automatic Downgrading
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ABSTRACT

This analysis updates previous stress calculations on the dome end support ring plate, by using newly acquired data on pressure drops and temperatures. In addition, it analyzes the stresses in the nozzle end support ring.

This analysis and the pertinent component tests show that the dome and nozzle end support rings are satisfactory for their intended service.

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NOTATIONS

- δ - Radial deformation relative to reactor ℓ (inch)
 θ - Rotation (Radian)
 H - Force acting perpendicular to reactor ℓ (lb/in.)
 V - Force acting parallel to reactor ℓ (lb/in.)
 M - Moment (in.lb/in.)
 p - Pressure (psi)

SUBSCRIPTS

A, B, C, E and G - refer to the rotation or deflection of the correspondingly designated item on Figure 1, or to the item on which the subscripted load acts.

AB, AE, AG and BC - refer to the interface reaction between the two components identified by the two individual letters on Figure 1.

At, Bt, Ct, Et and Gt - refer to the free thermal deformation or rotation of the item identified by the first letter of the combination.

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I. INTRODUCTION

The plate region of the dome end support ring of the NRX-A reactor has been previously analyzed (Reference 6.) Since that time much new information concerning system pressure drops and temperatures has become available. Therefore, the present analysis has been undertaken to update the stress calculations on the dome end support ring plate as well as to analyze and report the stress in the nozzle end support ring.

Before the stress calculations can be performed, the interface reactions acting between the components of the outer reflector as well as the deformations of these components must be determined. These have been calculated for a full power operating condition in sub-section II.A. These results which reflect the latest data available, are used to compute the stress in the dome end support ring plate (sub-section II.B) and the stress in the nozzle end support ring (sub-section II.C).

The maximum equivalent stresses which occur in the dome end support ring plate are shown on Figure 5. The highest stress in the nozzle end support ring is -32,600 psi at the thin web adjacent to the control drum bearing location. All of the stresses shown are acceptable.

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II. DETAILED ANALYSIS

A. OUTER REFLECTOR DEFORMATIONS AND REACTIONS

In order to analyze the stresses in the dome end support ring plate and the nozzle end support ring, it is first necessary to determine the deformation of the outer reflector and the interface reactions which occur between the components. In the sections which follow this is called the "Gross Effect". Then local conditions, such as junction flexibility or concentration of the calculated distributed reactions are examined.

A.1. Gross Effects:

In order to obtain the gross deformations, the method employed in the previous analysis (Reference 1) was used. That is, the deformation of each component of the reflector assembly was defined in terms of influence coefficients and interface reactions. Solution of the resulting set of simultaneous equations, to satisfy the condition of compatible deformation of adjacent components, determines the interface reactions and the final deformations of the parts.

The equations which follow differ from those found in Reference 1 because of the following reasons:

- 1) The pressure distribution acting on the external surface of the sectors is considered to have a trapezoidal shape rather than the earlier triangular distribution.
- 2) Some of the equations have been modified to take into account the effect of reactions which were previously neglected. For example, the effect of the interface shear forces on rotation of the dome and nozzle end plates (items A and C, Figure 1) has been included.

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Figure 1. Outer Reflector Components and Loads

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- 3) All of the equations are written such that the reactions are on a "lb/in." of circumference basis rather than the "lb/radian" basis which was used in Reference 1.

The equations which relate the reactions and deformations of the outer reflector assembly are:

$$\delta_A = -4.3995 \times 10^{-5} H_{AE} - 4.0056 \times 10^{-5} H_{AB} + 3.6163 \times 10^{-5} H_{AG} + \delta_{At} \quad (1)$$

$$\begin{aligned} \theta_A = & 3.7962 \times 10^{-3} V_{AE} + 4.3138 \times 10^{-4} H_{AE} - 3.9277 \times 10^{-4} H_{AB} \\ & - 3.5459 \times 10^{-4} H_{AG} - 1.7255 \times 10^{-3} M_{AE} - 1.5711 \times 10^{-3} M_{AB} \\ & - 1.4183 \times 10^{-3} M_{AG} + \theta_{At} + 3.0849 \times 10^{-3} V_{AG} \end{aligned} \quad (2)$$

$$\begin{aligned} \delta_B = & -1.4768 \times 10^{-4} H_{BC} - 4.2601 \times 10^{-6} M_{BC} - 2.9537 \times 10^{-5} H_{B1} \\ & - 5.5382 \times 10^{-5} H_{B2} + \delta_{Bt} \end{aligned} \quad (3)*$$

$$\begin{aligned} \theta_B = & -4.2601 \times 10^{-6} H_{BC} - 1.6385 \times 10^{-7} M_{BC} - 7.1002 \times 10^{-7} H_{B1} \\ & - 1.4200 \times 10^{-6} H_{B2} + \theta_{Bt} \end{aligned} \quad (4)*$$

$$\delta_C = 2.7923 \times 10^{-5} H_{BC} + \delta_{Ct} \quad (5)$$

$$\theta_C = -2.3208 \times 10^{-4} H_{BC} + 9.939 \times 10^{-4} M_{BC} + \theta_{Ct} \quad (6)$$

$$\begin{aligned} \delta_E = & 8.696 \times 10^{-7} V_{AE} + 1.1153 \times 10^{-4} H_{AE} + 5.2563 \times 10^{-5} M_{AE} \\ & - 1.0150 \times 10^{-4} p_E + \delta_{Et} \end{aligned} \quad (7)$$

$$\begin{aligned} \theta_E = & 1.4079 \times 10^{-6} V_{AE} + 5.2706 \times 10^{-5} H_{AE} + 4.6787 \times 10^{-5} M_{AE} \\ & + 1.3943 \times 10^{-5} p_E + \theta_{Et} \end{aligned} \quad (8)$$

*Note: These equations define the rotation and deflection of the aft end of the sectors relative to the forward end.

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$$\delta_G = -4.7975 \times 10^{-4} H_{AG} + 5.215 \times 10^{-4} M_{AG} + \delta_{Gt} \\ + 3.3567 \times 10^{-4} V_{AG} + 1.6713 \times 10^{-4} p_G + \delta_G (M_o, H_o) \quad (9)$$

$$\theta_G = -5.215 \times 10^{-4} H_{AG} + 8.515 \times 10^{-4} M_{AG} + \theta_{Gt} \\ + 4.6056 \times 10^{-4} V_{AG} + \theta_G (M_o, H_o) \quad (10)$$

These equations in turn are related by the following compatibility statements:

$$\delta_A = \delta_E \quad (11)$$

$$\theta_A = \theta_E \quad (12)$$

$$\delta_A = \delta_G \quad (13)$$

$$\theta_A = \theta_G \quad (14)$$

$$\delta_C = \delta_A + \delta_B + 52 \theta_A \quad (15)$$

$$\theta_C = \theta_A + \theta_B \quad (16)$$

The remaining equations needed to solve for the system unknowns are obtained from equilibrium considerations as follows:

$$-52 H_{BC} + M_{AB} - M_{BC} - 17.333 H_{B1} - 26 H_{B2} = 0 \quad (17)$$

$$H_{AB} - H_{BC} - H_{B1} - H_{B2} = 0 \quad (18)$$

$$24.575 V_{AE} - 20.2 V_{AG} + 22.375 (V_w + V_{\Delta p}) = 0 \quad (19)$$

Note: H_{B1} = Load from the triangular component of the pressure distribution p_B (See Figure 1).

H_{B2} = Load from the uniform component of the pressure distribution p_B (See Figure 1).

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To better understand the effects of each loading condition, the interface reactions and deformations of the outer reflector were obtained by solving the equations for the following three independent cases of loading:

- 1) Axial loads
- 2) Radial pressure loads
- 3) Thermal deformation of the parts due to their average temperature.

Note: The case of tie bolt preload was also examined but the results were negligible compared to the above cases, and therefore, not included.

The input for solution of equations (1) through (19) was calculated from information derived from analyses performed at WANL (References 2, 3, and 4). These inputs are listed in Table I.

TABLE I
KNOWN INPUT FOR SIMULTANEOUS EQUATIONS

$V_{AG} = 840 \text{ lb/in}$	$\delta_{At} = - .01096 \text{ in.}$
$V_{\Delta p} = 169.7 \text{ lb/in}$	$\delta_{Bt} = + .01704 \text{ in.}$
$V_w = 19.3 \text{ lb/in}$	$\delta_{Ct} = - .02779 \text{ in.}$
$H_{B1} = 445 \text{ lb/in}$	$\delta_{Et} = - .01619 \text{ in.}$
$H_{B2} = 327 \text{ lb/in}$	$\delta_{Gt} = - .01842 \text{ in.}$
$p_E = 34 \text{ psi}$	$\theta_{At} = 0 \text{ rad}$
$p_G = 6 \text{ psi}$	$\theta_{Bt} = + .0006560 \text{ rad}$
$\delta_G (M_o, H_o) = - .08330 \text{ in.}$	$\theta_{Ct} = 0 \text{ rad}$
$\theta_G (M_o, H_o) = - .19798 \text{ rad}$	$\theta_{Et} = + .006254 \text{ rad}$
	$\theta_{Gt} = 0 \text{ rad}$

The solution obtained for the three cases mentioned above and the total are summarized in Table II.

TABLE II
SOLUTIONS OF SIMULTANEOUS EQUATIONS

LOADING CASE				
<u>Unknown</u>	<u>Axial</u>	<u>Radial</u>	<u>Thermal</u>	<u>Total</u>
δ_A	.0078	-.0105	-.0172	-.0199 in.
δ_C	.0016	-.0088	-.0276	-.0348 in.
θ_A	0.429×10^{-4}	-2.73×10^{-4}	-5.09×10^{-4}	-7.39×10^{-4} rad
θ_C	-2.02×10^{-4}	3.08×10^{-4}	1.18×10^{-4}	2.24×10^{-4} rad
V_{AE}	518.4	0	0	518.4 lb/in
H_{AE}	155.7	-118.3	126.7	164.1 lb/in.
H_{AB}	56.9	455.2	6.8	518.9 lb/in
H_{AG}	468.8	70.6	-9.8	529.6 lb/in
H_{BC}	56.9	-316.8	6.8	-253.1 lb/in
M_{AE}	-190.1	117.3	-287.2	-360.0 in.lb/in.
M_{AB}	2974.4	-329.6	357.0	3001.8 in.lb/in
M_{AG}	65.3	42.9	-6.6	101.6 in.lb/in
M_{BC}	13.1	-73.7	1.7	-58.9 in.lb/in

A.2. Local Effect at Cylinder - Plate Junction:

However, the reactions given above for H_{AE} and M_{AE} will be affected by the flexibility of the support ring plate between the tie bolt and the outer radius, and by the thermal gradient in the plate in this region (neither of these factors was accounted for in the first part of the analysis). Assuming uniform distribution of these reactions in the circumferential direction and that this portion

of the plate can be represented as a beam fixed at the tie bolt centerline (See Figure 2), an expression for the rotation at the end of the beam can be developed from standard equations (Reference 5) as follows:

Figure 2. Local Plate Flexibility

$$\theta'_A = \theta_A + \frac{ML}{EI} + \frac{1}{2} \frac{V_{AE}L^2}{EI} + \theta'_{At},$$

and

$$\begin{aligned} \theta'_A = & (-7.3911 \times 10^{-4}) - (8.0176 \times 10^{-6}) M_{AE} + (1.5835 \times 10^{-6}) H_{AE} \\ & + (2.8062 \times 10^{-6}) V_{AE} + (-.001637). \end{aligned} \quad (20)$$

Where,

θ_A = Rotation of the plate from the gross deflection analysis
 (see Table II)

θ'_{At} = Rotation of the end of the plate due to its thermal gradient.

Assuming the radial deflection of the cylinder, as calculated in sub-section A.1, is not affected by the local flexibility of the plate, a better approximation of the reactions at the cylinder-plate junction can be obtained by equating equations 8 and 20 and solving simultaneously with equation 7.

These values, which replace those calculated in sub-section A.1 are:

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$$H_{AE} = 118.6 \text{ lb/in.}$$

$$M_{AE} = -263.5 \text{ in.lb/in.}$$

3. Concentration of Reactions:

Up to this point the reactions have been assumed to be uniformly distributed around the circumference. However, since they are transmitted between the dome end support ring and the remainder of the outer reflector by a discrete number of tie bolts (36), there will be some concentration at these points. Dr. A. M. Wahl in an earlier report (Reference 6) estimated the concentration would be based on the average spacing of the tie bolts. Effectively he concentrated 4.27 inches of uniformly distributed load on a 1.25 inch strip of the plate at the tie bolt, i.e. an effective concentration factor of 3.42. A calculation based on the compatible deformation of the cylinder and plate loaded by thirty-six concentrated forces results in a concentration factor of 3.26 (based on a 1.25 inch strip) at the tie bolt adjacent to the control drum hole and 1.89 at the tie bolt between sectors. Using the 3.26 value and assuming it applies equally as well to the moment and shear reactions, the concentrated reactions (designated by primes) become:

$$V'_{AE} = (3.26) (518.4) = 1690 \text{ lb/in.}$$

$$H'_{AE} = (3.26) (118.6) = 386.5 \text{ lb/in.}$$

$$M'_{AE} = (3.26) (-263.5) = -859 \text{ in.lb/in.}$$

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B. DOME END SUPPORT RING PLATE STRESS

The total stresses existing in the support rings during operation have been obtained by superposition of two cases, namely;

- 1) The gross effect of deformation caused by all external loads and the average temperatures of the various outer reflector components.
- 2) The local effects of reaction concentration, local flexibilities and thermal gradients in the support rings.

At the end of the section those stresses which act in the same region will be added either directly or by the Mises criterion to determine the total stress.

B.1. Hoop Stress Caused by Deflection:

Table II gives the total radial deflection of the plate as - 0.0199 inch. Of this quantity, free deflection due to thermal considerations amounts to - 0.0110 inch. The amount of forced deflection which results in hoop stress is then - 0.0089 inch radially.

To calculate the stress in the plate all holes except the control drum holes were neglected. Because of its repetitive geometrical pattern only one 15° section (of the 24 similar sections) need be examined. This element is shown in Figure 3.

Figure 3. Symmetry Element of Dome Plate

Considering equilibrium of a small segment, $d\theta$, of the plate:

$$HR_o d\theta = 2 \sigma_h A_n \sin \frac{d\theta}{2},$$

where σ_h = the hoop stress,

A_n = the cross sectional area at the n^{th} cross section.

For small angles this reduces to

$$\sigma_h = \frac{HR_o}{A_n}. \quad (21)$$

Since A_n varies along the circumference and if H is assumed constant at all circumferential positions, the change in circumferential length of the plate based on the symmetry element shown in Figure 3 is;

$$2\pi(\delta_A) = 24 \frac{HR_o}{E} \sum \frac{l_n}{A_n}$$

Rearranging the equation gives,

$$H = \frac{E (\delta_A) 2 \pi}{24 R_o \sum \frac{l_n}{A_n}} \quad (22)$$

Evaluating equation 22 for the dome plate gives $H = 208.5$ lb/in, and substitution of this value into equation 21 results in a (compressive) hoop stress of $-27,900$ psi at the thin web of the plate. At sections away from the control drum hole, the stress is -2790 psi (compression).

B.2. Hoop Stress (Bending) Caused by Rotation:

Assuming the plate rotates uniformly (i.e. no radial distortion of sections) the stress is;

$$\sigma = \frac{\theta E y}{R} \quad (23)$$

However, for the rotation given in Table II the stress is insignificant. Therefore, it is neglected.

B.3. Stress at Junction with Cylinder and at Tie Bolt Centerline:

The adjusted reactions existing at the junction of the plate and the cylinder, taking into account both the local flexibility of the plate, (sub-section A.2.) and the effect of reaction concentration at the tie bolt location (sub-section A.3.) are as indicated on Figure 4.

Figure 4. Reactions at Junction of Plate and Cylinder

The stresses to which these reactions give rise (conservatively assuming the plate acts as a simple beam) are given by:

$$\sigma_a = - \frac{6(859)}{(.35)^2} + \frac{1690}{.35} = - 37,170 \text{ psi}$$

$$\sigma_b = + \frac{6(859)}{(.35)^2} + \frac{1690}{.35} = 46,830 \text{ psi}$$

$$\tau_{c(\max)} = \frac{3}{2} \frac{(386.5)}{.35} = 1660 \text{ psi}$$

$$\sigma_d = - \frac{386.5}{.447} + \frac{(6)(386.5)(.2235)}{(.447)^2} + \frac{(6)(859.)}{(.447)^2} + \frac{(6)(1690)(.7)}{(.447)^2} = 62,980 \text{ psi}$$

$$\sigma_e = - \frac{386.5}{.447} - \frac{(6)(386.5)(.2235)}{(.447)^2} - \frac{(6)(859)}{(.447)^2} - \frac{(6)(1690)(.7)}{(.447)^2} = - 64,710 \text{ psi}$$

$$\tau_{f(\max)} = \left(\frac{3}{2}\right) \frac{(1690)}{(.447)} = 6420 \text{ psi}$$

B.4. Thermal Stresses:

In the calculation to determine the gross conditions in the reflector assembly, the assumed average temperature of the plate was 430°R. With reference to this average temperature, thermal stresses are developed at the hottest and coldest regions of the plate. These values will be superimposed on those previously determined. In addition, radial and tangential stresses in the plate caused by the difference in expansion of the plate and beryllium sectors must be determined. All temperature values used were obtained from Reference 2 in which a reflector inlet temperature of 162°R was used in analysis reflecting the AGC aluminum cast nozzle tube exit temperature.

The maximum stresses occurring at the hot and cold regions are obtained by using the relationship $\sigma = \frac{E \alpha \Delta T}{1 - \nu}$ (or $E \alpha \Delta T$ where applicable) where the base temperature is 430°R (the average temperature used in part A). The area reported to be hottest is the forward surface of the plate at the tie bolt location. The expected temperature in that region is approximately 640°R. For this condition the stress is:

$$\sigma = (17 \times 10^6)(5 \times 10^{-6})(430 - 640)(1.43) = -25,500 \text{ psi.}$$

The coldest region (230°R) is seen to be the aft surface of the plate over the plenum (not at point "d" of Figure 4). The corresponding stress is:

$$\sigma = (17 \times 10^6)(4.5 \times 10^{-6})(430 - 230)(1.43) = 21,900 \text{ psi.}$$

Both of these stresses act in the tangential and radial directions.

At the narrow section of the plate in line with the control drum there is poor heat removal because of the stagnant pool of hydrogen surrounding the plate. The temperature in this region is estimated to be about 700°R. Based on the previously assumed average of 430°R, this causes a maximum circumferential compressive stress of:

$$\sigma = (17 \times 10^6)(5 \times 10^{-6})(430 - 700) = -23,000 \text{ psi.}$$

Despite the fact that the coefficients of expansion are similar for the titanium plate and the beryllium sectors the two operate at such different temperatures that a different amount of thermal expansion occurs in each part. It has been found that the forces which could potentially be generated by the thermal condition exceed the maximum friction force which is expected to act at the interface of the parts. Therefore, radial and tangential slipping between the plate and sector is expected. Considering the temperature to be

uniform at all points in the plate and similarly in the sector, the stress developed in the plate by the maximum friction force is - 10,100 psi (circumferential compressive stress). Whether or not this stress actually exists will be determined by the strain history of the plates. That is, during the initial cooldown the plates cool faster than the adjacent portions of the sectors causing tensile stress in the plate. Subsequent additional cooling of the sector and/or heating of the plates may then simply relieve the initial tensile stresses in the plate rather than cause the relatively high compressive stress given above. Since it is conservative to use the full value of the compressive stress when combining stresses at the tie bolt, this will be done.

B.5. Summary of Dome End Plate Stresses:

For the most part, the stresses calculated above occur independently. They add at only the following three locations:

a. At the Thin Section in Line with the Control Drum:

The total of the thermal and mechanical stresses in the circumferential direction is:

$$\sigma = - 23,000 - 10,100 - 27,900 = -61,000 \text{ psi}$$

b. At Point "d" of Figure 4:

The total stress conditions results from the bending stress and the hoop stress. In this case, the total stress is an equivalent stress obtained from the Mises criterion. The total hoop stress consists of the sum of the mechanical and thermal components, i.e.

- 2790 psi (mechanical) plus - 10,100 psi (thermal) for a total of - 12,890 psi.

The equivalent stress then becomes:

$$\sigma_e = \sqrt{(62,980)^2 - (62,980)(-12,890) + (-12,890)^2}$$

$$\sigma_e = 70,200 \text{ psi.}$$

c. At Point "e" of Figure 4:

In the radial direction, the combined stress is:

$$\sigma_r = -64,710 - 25,500 = -90,210 \text{ psi.}$$

In the tangential direction, it is:

$$\sigma_t = -2,790 - 10,100 - 25,500 = -38,390 \text{ psi.}$$

As above, the maximum equivalent stress is obtained by the

Mises criterion as follows:

$$\sigma_e = \sqrt{(-90,210)^2 - (90,210)(-38,390) + (-38,390)^2}$$

$$\sigma_e = 78,500 \text{ psi}$$

The critical stress is the one at point "e". At that point, the operating temperature is approximately 640°R. The minimum yield strength at this temperature is about 80,000 psi and the minimum ultimate strength is 87,000 psi (Reference 7). The calculated value exceeds the allowable of 69,500 psi (based on 80 percent of the ultimate strength) if the load is as severe as that which has been used in the analysis. However, the analysis disregards the restraining effect of the nearly unloaded material on each side of the 1.25 inch strip which was analyzed and this is expected to be very helpful in reducing the stress and deflection.

At point "d", the temperature is about 300°R for which the minimum yield strength is 127,000 psi and the minimum ultimate strength is 133,000 psi. Therefore, the stress at "d" (based on $0.8 S_u = 106,000$ psi) is acceptable.

Figure 5. Location of Maximum Equivalent Stresses

6. Axial Gap at Dome End Plate - Sector Interface:

Because of the flexibility of the plate at the junction with the cylinder of the dome end support ring, local plate deflection causes a gap to develop between the plate and the sectors. Including the effects of both thermal and mechanical loads, the deflection of the strip is given by:

$$\delta = \delta_{at} + \frac{1}{EI} \left\{ \frac{L^2}{2} \left[-M'_{AE} + H'_{AE} \frac{t}{2} \right] + \frac{V'_{AE} L^3}{3} \right\} + \frac{3.9 L V'_{AE}}{AE}, \quad (24)$$

$$\delta = -.0006 + \frac{12}{(17 \times 10^6)(1)(.447)^3} \left\{ \frac{(0.7)^2}{2} \left[-(-859) + (386.5)(.2235) \right] + \frac{(1690)(0.7)^3}{3} \right\} + \frac{(3.9)(.7)(1690)}{(1)(.447)(17 \times 10^6)} = 0.0034 \text{ inch.}$$

However, the conservativeness of this calculation is demonstrated by the fact that, during the course of a recent axial load test on the outer reflector assembly, insertion of a 0.001 inch feeler into the gap was not possible.

C. NOZZLE END SUPPORT RING STRESS

C.1. Hoop Stress Caused by Deflection:

The total nozzle end ring radial deformation is given in Table II as - 0.0348 inch. Of this amount, the stress free thermal deflection is - 0.0278 inch. Therefore, the stress producing forced radial deflection is - 0.007 inch. Evaluating equation 22 gives $H = 216 \text{ lb/inch}$. Substituting this value into equation 21, a stress of - 24,800 psi is determined at the thin webs and - 3300 psi at sections away from the thin webs.

C.2. Hoop Stress Caused by Rotation:

The nozzle end ring hoop stress caused by rotation is insignificant because the rotation is small (See Table II). This is similar to the condition in the dome end support ring plate.

C.3. Thermal Stress:

Because of the axial temperature gradient in the nozzle ring (hotter on the aft face), it will try to bow radially. However, the tie bolt preload

will tend to restrain the bow. Assuming that the severest gradient acts over the entire ring and that the resulting bow is completely restrained by the tie bolt load, the calculated radial stress is 9350 psi (tension on the forward side of the ring and compression on the aft surface).

The average temperature of the ring was taken to be 260°R for the gross deflection calculation. The thermal stresses at the hottest and coldest points in the ring will be determined using $\sigma = \frac{E\alpha\Delta T}{1-\nu}$.

The hottest point in the ring is seen to occur near the tie bolt. The temperature is given as 395°R. For this value a compressive stress of -15,900 psi is developed.

The coldest regions at 160°R are in the cooling holes and on the forward side of the ring. For this temperature difference, a tensile stress of 10,200 psi is calculated.

Again, as in the case of the dome end ring, the operating temperatures of the nozzle ring and the sectors are sufficiently different to cause different amounts of thermal expansion despite their close coefficients of expansion. Because of the frictional shear developed at the interface, a circumferential compressive stress of -7800 psi develops in the nozzle ring.

C.4. Summary of Nozzle End Support Ring Stresses

The stress condition in the nozzle end ring is not as severe as that in the dome end plate. The highest stress is the combined mechanical and thermal circumferential compression of -32,600 psi at the thin web. Conservatively assuming the material temperature in that region to be 300°R, the minimum yield strength is found to be 127,000 psi. Therefore, the nozzle ring stresses will obviously present no problem.

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