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STRESS ANALYSIS: PRELOAD SPRING
ASSEMBLY AND DOME END SEAL
(Title Unclassified)

Classification cancelled (or changed to)
by authority of DOC
by H. F. C. T. C. J. a SEP 10 1973

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Astronuclear
WANL-TME-1030
December 1964

SUBMITTED BY:

Westinghouse Electric Corporation
Astronuclear Laboratory
Pittsburgh, Pennsylvania 15236

MASTER

STRESS ANALYSIS: PRELOAD SPRING
ASSEMBLY AND DOME END SEAL
(Title Unclassified)

PREPARED BY:

E. A. Henne
E. A. Henne
Reactor Design

INFORMATION CATEGORY <i>CRD</i>	
<i>W.B. Russell</i> Authorized Classifier	<i>12/17/64</i> Date

GROUP 1 Excluded From Automatic Downgrading and Declassification
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SUMMARY

All components were analyzed for the full-power, steady-state operating condition (100 per cent power, 100 per cent load), and their stresses were well within the specified yield point. A summary of each individual component is as follows:

A. Spring Support Ring

The computed maximum stress (16,000 psi) is acceptable when compared to the material yield strength (40,000 psi).

Using a conservative method, the critical buckling load was computed to be 632,000 lbs. Since the total axial spring load is only 16,800 lbs, buckling is not a problem.

B. Spring Retainer

The spring retainer is under a small axial load (90 lbs seated). Since this load causes a stress less than 500 psi, there is no problem.

C. Preload Springs

The shear stress was computed for the assembly, as well as the full-power, steady-state operating condition. The maximum shear stresses for assembly condition (84,700 psi for the outer and 98,200 psi for the inner spring) are below the torsional yield strength of 89,700 and 103,500 psi respectively at room temperature. The maximum stresses for operating condition (95,000 psi for the outer and 110,000 psi for the inner spring) are well below the torsional yield strength of 143,500 and 165,500 respectively for a normal operating temperature of -190° F. Therefore, it can be concluded that the springs operate at a safe stress condition.

D. Spring Bearing Ring

The maximum stress (less than 500 psi) from the preload springs is negligible.

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E. Dome End Seal

The dome end seal is exposed to a maximum bending stress of 28,000 psi. Since this stress is less than the operating yield strength of the material (36,700 psi), the seal will function properly.

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I. INTRODUCTION

This report presents a stress analysis of the following parts of the NRX-A reactor at 100 per cent power, 100 per cent flow, steady-state operating condition:

- A. Spring Support Ring
- B. Spring Retainer
- C. Preload Springs
- D. Spring Bearing Ring
- E. Dome End Seal

These parts are located in the triangular annulus between the core support plate and the core support ring (Figure 1). The spring support ring, spring retainer, preload springs, and spring bearing ring are members of the axial support system of the inner reflector cylinder. The preload springs provide an axial force to the inner reflector cylinder holding it in contact with the nozzle (for sealing) during operation, yet permitting relative axial thermal expansion and contraction of the inner reflector cylinder, nozzle, pressure vessel, dome end support ring, core support ring, and spring support ring closed loop. There are 96 sets of coil springs, each set consisting of an outer and inner spring (Figure 1). These springs are kept in place at each end by a spring retainer. The spring retainers rest in a counterbore of the spring support ring and the spring bearing ring respectively, thereby providing radial and circumferential positioning of the spring sets.

The dome end seal bridges the gap between the support plate and the core inlet, preventing excessive coolant leakage from the reflector area into the core inlet plenum, yet permitting the core to move axially and radially.

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Figure 1. Axial Support Component Arrangement

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II. STRESS ANALYSIS

Each component tested, drawing number, and component material is listed in order to provide a readily available reference source.

<u>Component</u>	<u>Drawing No.</u>	<u>Material</u>
Spring Support Ring	936J482	AL 2219-T-852
Spring Retainer	945C838	AL 6061-T6
Preload Spring	964B150	CRES
Preload Spring	964B151	CRES
Spring Bearing Ring	909E446	AL 2219-T-852
Dome End Seal	909L668	AL 2219-T-62

A. SPRING SUPPORT RING

The spring support ring was considered to be a thin shell of revolution and a computer program (Ref. 1) utilized to compute the stresses and deflections.

There are 96 axial and 96 radial holes drilled through the support ring in addition to the 96 slots in the core end of the spring support ring. These apertures provide channels for the coolant flow from the inner reflector to the core support ring. This results in an equal pressure around the spring support ring.

The temperatures of the ring shown in Figure 2 have been included for information purposes (Ref. 2). It can be seen that the temperature gradients are very small; therefore, a uniform average temperature of 312°R was used in the analysis. Even though both the spring support ring and the core support plate are made of AL 2219, they are not subjected to the same temperatures. At 100 per cent power, 100 per cent flow, steady-state operating condition, the support plate contracts more than the support ring at their junction. Therefore, a radial force could be exerted at Point D of the spring support ring (Figure 3).

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Figure 2. Inner Reflector Spring Support
Ring Temperatures (100% Power,
100% Flow)

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Figure 3. Spring Support Ring Cross Section,
Stresses and Deflections

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Two cases were calculated.

- (1) Axially Loaded Only (i.e., coefficient of friction equals zero at the junction of the support ring and the support plate)
- (2) Axially Loaded With Forced Deflection at Point D.

Case 1. Axially Loaded Only: In the computation of the stresses and deflections due to axial load (F_D), the assumption is made that there is no radial constraint of the spring support ring at Point D (Figure 3).

The nominal total axial load exerted by the springs on the spring support ring is $F_T = 16,800$ lbs (Ref. 6), which corresponds to a load of

$$F_B = \frac{F_T}{2R_B\pi} = \frac{16,800}{2(18.72)\pi} = 142.7 \text{ lbs/in. of circumference.}$$

This spring load is reacted at Point C of the shell. The section between Points C and D was treated as a cantilever beam since the shell is slotted over that length. The radial offset between Points C and D causes a moment

$$M_C = F_B(R_D - R_C) = 142.7(18.277 - 18.370) = -13.27 \text{ in. lbs/in. of circumference.}$$

The spring support ring was divided into five regions (Figure 3), and the axial load (F_B) was applied at Point B. The effect of the cooling holes on the elastic behavior of the shell was neglected since the hole diameter is small compared to the pitch.

The computer inputs for this case were based on the above forces and moments and on the geometry of the ring (Figure 3). These inputs are shown on Table 1. The modulus of elasticity, $11.4(10^6)$, was obtained (Ref. 5) using a temperature of 312° R.

The following results were obtained from the computer printout:

- (1) The radial deflections due to mechanical load only (in.)
- (2) The rotation β (radians)

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*NE-8000 CORE ENGRG ANAL HENNE 601EAH059000 02
*SPRING SUPPORT RING
DATE 9.11 CASE 1. NREGIONS 5
ENDC(1,1) -1 2 0. ENDC(2,1) -1 4 0.
ENDC(3,1) +5 4 -13.27 ENDC(4,1) +5 2 0.
JUNC(1,1) +1 -2 0 0. 0. JUNC(2,1) +2 -3 0 0. 0.
JUNC(3,1) +3 -4 0 0. 0. JUNC(4,1) +4 -5 0 0. 0.
NSTATIONS 3 DELTAXI .0014 NU .3 PRINTOUT 20
RVCONT 0 NORV 0 RVI 0.0 NLAYERS 1
GEOM 1 RO 19.320 ANGLE 97.2 XI 0.0(.070).140
P(1-3) 0. PCOMP 0.0
T(1-3) 0. T(26-28) 0.
A(1-3) 10.8E-6 Y(1-3) 11.4E6 H(1) 1.19 1.15 1.11
1
NSTATIONS 3 DELTAXI .0021 NU .3 PRINTOUT 20
RVCONT 0 NORV 0 RVI -2670. NLAYERS 1
GEOM 1 RO 19.265 ANGLE 85.0 XI 0.0(.105).21
P(1-3) 0. PCOMP 0.0
T(1-3) 0. T(26-28) 0.
A(1-3) 10.8E-6 Y(1-3) 11.4E6 H(1) 1.11 1.01 .91
1
NSTATIONS 3 DELTAXI .0041 NU .3 PRINTOUT 20
RVCONT 1 NORV 0 RVI 0.0 NLAYERS 1
GEOM 1 RO 19.180 ANGLE 104.2 XI 0.0(.205).41
P(1-3) 0. PCOMP 0.0
T(1-3) 0. T(26-28) 0.
A(1-3) 10.8E-6 Y(1-3) 11.4E6 H(1) .91 .598 .286
1
NSTATIONS 3 DELTAXI .0028 NU .3 PRINTOUT 20
RVCONT 1 NORV 0 RVI 0.0 NLAYERS 1
GEOM 1 RO 18.765 ANGLE 102.0 XI 0.0(.14).28
P(1-3) 0. PCOMP 0.0
T(1-3) 0. T(26-28) 0.
A(1-3) 10.8E-6 Y(1-3) 11.4E6 H(1) .286 .210 .134
1
NSTATIONS 3 DELTAXI .0075 NU .3 PRINTOUT 20
RVCONT 1 NORV 0 RVI 0.0 NLAYERS 1
GEOM 1 RO 18.630 ANGLE 105.0 XI 0.0(.375).75
P(1-3) 0. PCOMP 0.0
T(1-3) 0. T(26-28) 0.
A(1-3) 10.8E-6 Y(1-3) 11.4E6 H(1) .134 .125 .125
1

Table 1. Computer Input

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- (3) The axial stress (psi)
- (4) The circumferential stress (psi)

The radial deflections u and rotations β are listed in Table 2 for Points A, B, and C. The radial deflection, axial stress, and circumferential stress are plotted on Figure 3. The axial stress consists of the bending stress superimposed to the direct stress.

Point	A	B	C
$u(10^3)$	2.056	1.583	-8.694
$\beta(10^3)$	-3.443	-3.445	-13.64

Table 2. Radial Deflections and Rotations

Since the radial holes reduce the cross-sectional area, they will increase the axial stress and cause stress concentrations. There are 96 holes 0.375 inch in diameter per circumference, and the radius of the neutral surface at the center of the holes is 18.475 inch. Therefore, the stress correction factor K for the material removed by the holes is

$$K = \frac{(2)(18.475)\pi}{(2)(18.475)\pi - (96)(0.375)} = 1.45 .$$

This correction is shown as the dotted line in the axial stress curve of Figure 3.

The stress concentration factor was estimated to be approximately 2. Its effect is not shown in the stress diagrams and is only of concern if there were fatigue loads.

Deflection at Point D

The rotation at Point C was calculated to be $\beta_C = -0.01364$, and the deflection $u_C = -0.008694$ (Table 2).

Figure 4. Tab Deflection

Considering the distance CD as a cantilever beam, the total deflection of Point D can be calculated from:

$$\begin{aligned}
 u_D &= u_C + u'_6 + u''_6 = u_C + \beta_C l_6 + \frac{F_D}{EI_6} \left[\sin(-15^\circ + \beta_C) \right] \frac{l_6^3}{3} \\
 &= -0.008694 - (0.01364)(0.350) + \frac{207}{(11.4)(10^6)(1.627)(10^{-4})} \left[-0.272 \frac{0.350^3}{3} \right] \\
 &= -0.01389,
 \end{aligned}$$

where

$$F_D = \frac{F_T}{2R_D \pi - (96)(0.350)} = \frac{16,800}{(2)(18.277)\pi - 33.6} = 207 \text{ lbs/in. of tab width}$$

$$E = (11.4)(10^6) = \text{modulus of elasticity at } -218^\circ \text{ F}$$

$$I_6 = \frac{(1)(0.125)^3}{12} = 1.627(10^{-4}) = \frac{\text{in.}^4}{\text{in.}} = \text{moment of inertia of tab}$$

$$l_6 = 0.350.$$

Axial Stress in Tabs

$$\text{Point D: } \sigma_D = -\frac{F_D \cos(15^\circ + \beta)}{A} = -\frac{(207)(0.962)}{(1)(0.125)} = -1,592 \text{ psi}$$

$$\begin{aligned}\text{Point C: } \sigma_C &= -\frac{F_D \cos (15^\circ + \beta)}{A} \pm \frac{F_D \sin (15^\circ + \beta) \ell_6 C_6}{I_6} \\ &= -1,592 \pm \frac{(207)(0.272)(0.350)(0.0625)}{(1.627)(10^{-4})} = -1,592 \pm 7,560 \\ &= \begin{matrix} +5,968 \\ -9,152 \end{matrix} \text{ psi .}\end{aligned}$$

Case 2. Axially Loaded With Forced Deflection at Point D: This is a computation of the stresses and deflections due to axial load F_D and for a given radial deflection at Point D due to differential thermal contractions.

The radial thermal contraction of the spring support ring under the temperature condition shown in Figure 2 is

$$(\Delta R)_R = -\alpha R_D \Delta T = -(10.8)(10^{-6})(18.277)(530 - 312) = -0.0430 \text{ in.}$$

where

α = mean linear coefficient of thermal expansion based on an average temperature of 312° R (Ref. 5).

The total radial contraction at the periphery of the core support plate calculated for 100 per cent power, 100 per cent flow, steady-state operating condition is

$$(\Delta R)_P = -0.0519 \text{ in. (Ref. 4) .}$$

Assuming sufficient friction to prevent sliding between the core support plate and spring support ring, a radial deflection of $u = (\Delta R)_P - (\Delta R)_R = -0.0089$ inch is imposed at Point D of the spring support ring. This corresponds to -0.00502 inch at Point C (based on a similar calculation as in Case 1) and causes the stresses and deflections shown in Figure 5. (Note that the deflections and, therefore, the stresses are less than Case 1, Figure 3.)

Figure 5. Spring Support Ring Cross Section,
 Stresses and Deflections

The radial load H_D at Point D is

$$H_D = Q_C \frac{R_C}{R_D} K \cos (15^\circ + \beta) = (19.97) \left(\frac{18.370}{18.277} \right) (1.45) (0.963) = 28.0 \text{ lb/in. of tab}$$

where

Q_C = shear force at Point C, perpendicular to the reference surface (meridian) of the shell, obtained from computer print-out.

Since F_D equals 207 (lb/in. of tab width) and the coefficient of friction between the core support plate and spring support ring in hydrogen is as high as 0.38, a friction force of 78 (lb/in. of tab width) could be transmitted. Therefore, the stresses and deflections of Figure 5 will occur under normal operating conditions. Table 3 lists the minimum tensile strength (σ_u) and the minimum yield strength (σ_y) of the material at R.T. (based on drawing specifications) and at operating temperature of 312°R.

		R. T.	312° R
Axial Direction	$(\sigma_u)_{\min}$	59,000	61,000
	$(\sigma_y)_{\min}$	39,000	40,000
Radial and Circumferential	$(\sigma_u)_{\min}$	62,000	64,000
	$(\sigma_y)_{\min}$	45,000	46,000

Table 3. Material Properties of AL 2219 T-852

From the computer results (Figure 3), the maximum stress in the spring support ring is less than 16,000 psi in the axial direction. This is a very safe stress level compared to the yield strength of 40,000 psi.

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Buckling: This is a complicated conical shell with variable wall thickness for which no theoretical solution is known. Reference 8 considers the buckling of a thin-walled conical shell with a uniform wall thickness and lists the following critical axial compression load for a simply supported conical shell:

$$P_{cr} = \frac{2\pi}{\sqrt{3(1 - \nu^2)}} E(t \cos \alpha)^2.$$

This equation should be relatively good for cones with clamped edges as well (Ref. 8). In order to be able to calculate a critical buckling load for the spring support ring, the actual cross section was simulated by a thin conical shell 0.125 inch thick and the length AD (Figure 6). This results in a very conservative critical load of

$$P_{cr} = \frac{2\pi}{\sqrt{3(1 - 0.3^2)}} (11.4)(10^6) [(0.125)(0.966)]^2 = 632,000 \text{ lbs.}$$

Since the total axial spring load is only 16,800 lbs, buckling does not present any problem.

Figure 6. Simulated Cross Section

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B. SPRING RETAINER

The function of the spring retainer is to transmit the axial force of the inner spring and positively locate the springs and keep them properly spaced.

The maximum load of the inner spring is 90 lbs (seated) which causes a negligible stress in the spring retainer. Since the spring retainers are at a uniform temperature of 312°R , no thermal stresses occur.

C. PRELOAD SPRINGS

The springs were designed to have the required characteristic and to fit into the available space. They were considered to be statically loaded (constant load or one repeated less than about 100 to 1000 times, Ref. 3) under reactor assembly and operating condition. Table 4 lists the spring temperatures for 100 per cent power, 100 per cent flow, steady-state operating condition (Ref. 2).

	Preload Spring	
	Outer	Inner
Surface Temperature ($^{\circ}\text{R}$)	263.0	262.4
Centerline Temperature ($^{\circ}\text{R}$)	278.8	270.4

Table 4. Spring Temperatures

Table 5 lists basic mechanical spring data based on present drawing specifications.

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	Preload Spring	
	Outer	Inner
Drawing No.	964B150	964B151
Material*	AISI 302	AISI 302
Minimum Tensile Strength at R. T.	195,000	225,000
Spring O. D.	1.168	0.768
Spring I. D.	0.840	0.534
Wire Diameter	0.164	0.117
No. of Active Coils	7.2	12.0
Free Length	2.686	2.856
Solid Height	1.43	1.58
Spring Rate (lb/in.)	124.55	70.9
*QQ-W-423a, Form 1, Composition 302, Condition B		

Table 5. Basic Spring Data

Shear Stress: Based on Ref. 3, the corrected torsional shear stress versus the spring compression was calculated utilizing the following equations:

$$\tau' = \frac{8KPD}{\pi d^3}$$

$$\delta = \frac{8PD^3n}{Gd^4}$$

where,

τ' = corrected shear stress (psi)

K = curvature correction factor (-)

P = spring load (lbs)

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D = mean coil diameter (in.)

d = wire diameter (in.)

δ = axial deflection (in.)

n = number of active coils

G = shear modulus (psi)

From the above two equations we obtain

$$\tau' = \frac{K\delta d G}{\pi D^2 n}$$

But K is a function of $\frac{D}{d}$. Denoting the outer spring by o and the inner spring by i we obtain

$$\left(\frac{D}{d}\right)_o = \frac{1.004}{0.164} = 6.12 \longrightarrow K_o = 1.24 \quad (\text{Ref. 3}),$$

$$\left(\frac{D}{d}\right)_i = \frac{0.651}{0.117} = 5.56 \longrightarrow K_i = 1.27 \quad (\text{Ref. 3}).$$

For a modulus of elasticity (E) of $(29.1)(10^6)$ at 270°R (Ref. 5), the shear modulus becomes

$$G = \frac{E}{2(1 + \lambda')} = (11.2)(10^6)$$

where

$$\lambda' = 0.3 = \text{Poisson's ratio.}$$

Then

$$\tau'_o = \frac{(1.24)(\delta)(0.164)(11.2)(10^6)}{\pi(1.004)^2(7.2)} = 99,700 \delta \text{ (psi),}$$

$$\tau'_i = \frac{(1.27)(\delta)(0.117)(11.2)(10^6)}{\pi(0.651)^2(12.0)} = 114,200 \delta \text{ (psi).}$$

These two equations will be used to calculate the spring stresses for certain conditions after defining the spring compressions δ .

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Thermal Stress: The thermal stresses are caused by the radial temperature gradient in the spring wire and can be calculated by assuming the spring to be a long circular cylinder (Ref. 9).

$$\text{Radial Stress} \quad \sigma_r = \frac{\alpha E}{1-\nu} \left[\frac{1}{b^2} \int_0^b T r dr - \frac{1}{r^2} \int_0^r T r dr \right]$$

$$\text{Circumferential Stress} \quad \sigma_\theta = \frac{\alpha E}{1-\nu} \left[\frac{1}{b^2} \int_0^b T r dr + \frac{1}{r^2} \int_0^r T r dr - T \right]$$

$$\text{Axial Stress} \quad \sigma_z = \frac{\alpha E}{1-\nu} \left[\frac{2r}{b^2} \int_0^b T r dr - T \right]$$

where

b = wire outside radius

r = wire radius

T = temperature at r (Figure 7)

Figure 7. Temperature and Stress Distribution

Assuming a linear temperature gradient (Figure 7), these equations become:

$$\sigma_r = \frac{\alpha E T_c}{1-\nu} \left[\frac{1}{b^2} \left(\frac{b^2}{6} \right) - \frac{1}{r^2} \left(\frac{r^2}{2} - \frac{r^3}{3b} \right) \right] = \frac{\alpha E T_c}{1-\nu} \left[\frac{1}{6} - \frac{1}{2} + \frac{r}{3b} \right] = \frac{\alpha E T_c}{1-\nu} \left[-\frac{1}{3} + \frac{r}{3b} \right]$$

$$\begin{aligned}\sigma_{\theta} &= \frac{\alpha_{ET_c}}{1-\nu} \left[\frac{1}{b^2} \left(\frac{b^2}{6} \right) + \frac{1}{r^2} \left(\frac{r^2}{2} - \frac{r^3}{3b} \right) - \left(1 - \frac{r}{b} \right) \right] = \frac{\alpha_{ET_c}}{1-\nu} \left[\frac{1}{6} + \frac{1}{2} - \frac{r}{3b} - 1 + \frac{r}{b} \right] \\ &= \frac{\alpha_{ET_c}}{1-\nu} \left[-\frac{1}{3} + \frac{2r}{3b} \right] \\ \sigma_z &= \frac{\alpha_{ET_c}}{1-\nu} \left[\left(\frac{2\nu}{b^2} \right) \frac{b^2}{6} - \left(1 - \frac{r}{b} \right) \right] = \frac{\alpha_{ET_c}}{1-\nu} \left[\frac{2\nu}{6} - 1 + \frac{r}{b} \right]\end{aligned}$$

Numerical analysis:

$$\alpha = (8.53)(10^{-6}) \text{ from R. T. to } 270^{\circ} \text{ R (Ref. 5)}$$

$$E = (29.1)(10^6) \text{ at } 270^{\circ} \text{ R (Ref. 5)}$$

T_c = temperature difference between centerline and surface

$$= 15.8^{\circ} \text{ R for the outer and } 8.0^{\circ} \text{ R for the inner spring (Ref. 2)}$$

With:

$$\frac{\alpha_{ET_c}}{1-\nu} = \frac{(8.53)(10^{-6})(29.1)(10^6)(15.8)}{1 - 0.3} = 5,600 \text{ (psi)}$$

the thermal stresses for the outer spring are:

	$r = 0$	$r = \frac{b}{2}$	$r = b$
σ_r	-1866	- 933	0
σ_{θ}	-1866	0	+1866
σ_z	-5040	-2240	+ 560

Table 6. Thermal Stresses

Table 6 shows that the thermal stresses at the surface of the spring (that is for $r = b$) are small. Therefore, they were neglected and not

superimposed on the spring shear stress (τ').

Table 7 lists the spring length, preload compressions (δ_1 , based on Ref. 10), and the corresponding loads and corrected stresses (τ').

	Preload Spring	
	Outer	Inner
Free Length	2.686	2.856
Nominal Length	1.881	2.051
Maximum Length	1.935	2.115
Minimum Length	1.837	1.997
Nominal Compression, δ_1	0.805	0.805
Minimum Compression, δ_1	0.751	0.741
Maximum Compression, δ_1	0.849	0.859
Nominal Load	100.2	57.1
Minimum Load	93.5	52.6
Maximum Load	105.8	60.9
Nominal τ'	80,300	92,000
Minimum τ'	74,900	84,600
Maximum τ'	84,700	98,200

Table 7. Spring Compressions and Stresses at Assembly

Based on the individual forces (Table 7) and 96 sets of springs, the total spring force at assembly is:

Nominal	15,100 lbs
Minimum	14,020 lbs
Maximum	16,000 lbs

At 100% power, 100% flow, steady state operating condition additional spring deflections (δ_2) occur due to pressure drop loads and thermal contractions.

These deflections (δ_2) were obtained from Ref. 10 and are listed in Table 8. This table also lists the total spring deflection ($\delta_3 = \delta_1 + \delta_2$), the corresponding loads and stresses τ'_3 .

	Preload Spring	
	Outer	Inner
Nominal Compression δ_2	0.0951	0.0951
Minimum Compression δ_2	0.0858	0.0858
Maximum Compression δ_2	0.1044	0.1044
Nominal Compression δ_3	0.9001	0.9001
Minimum Compression δ_3	0.8368	0.8268
Maximum Compression δ_3	0.9534	0.9634
Nominal Load	112.2	63.8
Minimum Load	104.2	58.7
Maximum Load	118.8	68.3
Nominal Stress τ'_3	89,700	102,900
Minimum Stress τ'_3	83,500	94,500
Maximum Stress τ'_3	95,000	110,000

Table 8. Spring Compressions and Stresses at Operating Condition

The total spring force due to the 96 spring sets at 100 per cent power, 100 per cent flow, steady-state operating conditions then becomes:

Nominal	16,890 lbs
Minimum	15,620 lbs
Maximum	17,970 lbs

Since the reactor could be subject to lower temperatures than those occurring at 100 per cent power, 100 per cent flow operating conditions, (for instance low power mapping condition when the total reactor cools to 55° R), the springs could be compressed even more, resulting in a higher stress.

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The highest obtainable spring stress occurs for solid spring compression ($\delta_{\max}$) and is listed in Table 9.

	Spring	
	Outer	Inner
$\delta_{\max}$	1.256	1.276
$\tau'_{\max}$	125,200	145,700

Table 9. Highest Obtainable Spring Stress

Allowable Stress: The torsional yield strength (τ_y) of the spring material was considered to be maximum allowable stress. The working stress should be less than τ_y .

The ratio $\frac{\text{torsional elastic limit } \tau_y}{\text{ultimate tensile strength } \sigma_u}$

for 302 stainless steel spring wire is 0.46 (Ref. 3). Table 10 lists the

$\frac{(\sigma_u)_T}{(\sigma_u)_{\text{R.T.}}}$ values (Ref. 6) for three temperatures and the corresponding

torsional elastic limit τ_y based on the above ratio. The minimum tensile strength at R.T. is taken from Table 5.

The maximum shear stresses for assembly condition (84,700 psi for the outer and 98,200 psi for the inner spring, Table 7) are still below the torsional yield strength of 89,700 and 103,500 psi respectively at room temperature (Table 10).

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Temperature T ($^{\circ}$ F)		-410	-190	R.T.
	$\frac{(\sigma_u)_T}{(\sigma_u)_{R.T.}}$ (psi)	2.27	1.60	1
Outer Spring	τ_y (psi)	204,000	143,500	89,700
Inner Spring	τ_y (psi)	235,000	165,500	103,500

Table 10. Torsional Yield Strength

The maximum stresses for operating condition (95,000 psi for the outer and 110,000 psi for the inner spring, Table 8) are well below the torsional yield strength of 143,500 and 165,500 respectively for a normal operating temperature of -190° R. Even when solidly compressed (125,200 psi for the outer and 145,700 psi for the inner spring, Table 9), the stress is within safe limits for -190° F operating temperature as well as down to the lowest possible temperature of -410° F.

Since AISI 302 stainless steel has still more than 20 per cent elongation at 50° R (Ref. 6), embrittling due to low temperature does not present any problem. Hydrogen, being an embrittling agent, does not effect AISI 320 and did not show any adverse effect on the springs in the NRX-A1 and A2 test runs.

Buckling: The critical buckling was checked for both springs in an unnested condition (Ref. 3).

The ratio $\frac{l_o}{D}$ ($\frac{\text{Free Length}}{\text{Mean Coil Diameter}}$) was calculated, and the ratio

$\frac{\delta_{cr}}{l_o}$ ($\frac{\text{Critical Buckling Deflection}}{\text{Free Length}}$) was obtained for fixed spring ends (Ref. 3).

Figure 8. Critical Buckling (From Ref. 3)

Then the critical buckling load (P_{cr}) was calculated using the equation

$$P_{cr} = \left(\frac{\delta_{cr}}{\ell_o} \right) \ell_o k$$

where k = spring rate. When the values ℓ_o/D are 2.67 and 4.38 for outer and inner spring respectively, the δ_{cr}/ℓ_o values (Ref. 3) become very large. Therefore, no buckling can occur even when the springs are solidly compressed.

D. SPRING BEARING RING

The spring bearing ring serves as a support and locating device for the nozzle end of the preload springs, transmits the axial spring load into the inner reflector cylinder, and, in addition, serves as a metering and flow directing device.

The only load acting on the ring is due to the preload springs and has a maximum of 247 lbs per spring set (seated). The resulting stresses are less than 500 psi and can be neglected. The bearing ring is at a uniform temperature of 306° F (Reference 11); therefore, no thermal stresses occur.

E. DOME END SEAL

The dome end seal is a cylindrical finger seal consisting of two concentric rings. It bridges the gap between the support plate and the filler blocks, limiting the coolant leakage from the inner reflector area into the core inlet plenum, yet permitting the core to move axially and radially. A pressure drop of $(716 - 696) = 20$ psi acts across the seal (Ref. 2).

The seal portion in contact with the core support plate was assumed to be at the same temperature as the core support plate at that region; that is, at 282° R (at Point A, Figure 1) and 295° R (at Point B, Ref. 2). The overhanging part (fingers) of the seal was assumed to be at coolant temperature of 250° R.

The core support plate has a seal diameter of 35.500 ± 0.005 . The inside diameter of the dome end seal is 35.485 ± 0.010 . Then the radial interference (ΔR) at room temperature is,

$$\begin{aligned}(\Delta R)_{R.T.} &= \frac{1}{2}(35.485 \pm 0.010 - 35.500 \pm 0.005) \\ &= 0 \text{ to } 0.015\end{aligned}$$

The tangential stress at R.T. due to that interference is,

$$(\sigma_t)_{R.T.} = \frac{E(\Delta R)_{R.T.}}{R} = \frac{(10.55)(10^6)(0.015)}{17.75} = 0 \text{ to } 8,920 \text{ psi.}$$

The values of (E) , (α) , $(\sigma_y)_{\min}$ and $(\sigma_u)_{\min}$ used in this and the following areas of analysis are listed in Table 11 and based on Ref. 5.

At 100 per cent power, 100 per cent flow, steady state operating condition, the core support plate seal radius contracts by the amount

$$(\Delta R)_p = 0.0519 \text{ in. (Ref. 4)}$$

	α	E	$(\sigma_y)_{\min}$	$(\sigma_u)_{\min}$
R. T.	$(12.0)(10^{-6})$	$10.55(10^6)$	34,500	52,000
295 ($^{\circ}$ R)	$(11.15)(10^{-6})$	$11.25(10^6)$	36,000	54,300
282 ($^{\circ}$ R)	$(11.1)(10^{-6})$	$11.30(10^6)$	36,200	55,000
250 ($^{\circ}$ R)	$(10.9)(10^{-6})$	$11.40(10^6)$	36,700	56,200

Table 11. Material Properties of AL 2219 T-62

The dome end seal contracts the maximum amount at Point A, which is:

$$(\Delta R)_S = \alpha(\Delta T)R = (11.1)(10^{-6})(530 - 282)\left(\frac{35.485}{2}\right) = 0.0488 \text{ in.}$$

The total radial interference at 100 per cent power, 100 per cent flow, steady-state operating condition then becomes:

$$\begin{aligned} (\Delta R)_T &= (\Delta R)_{R.T.} - (\Delta R)_P + (\Delta R)_S = (0 \text{ to } 0.015) - 0.0519 + 0.0488 \\ &= 0 \text{ to } 0.0119 . \end{aligned}$$

The maximum tangential stress at operating condition then becomes:

$$(\sigma_t)_{282^{\circ} R} = \frac{E(\Delta R)_T}{R} = \frac{(11.30)(10^6)(0.0119)}{17.75} = 7,570 \text{ psi.}$$

Stress in Fingers Due to Mechanical Deformation: The maximum finger deflection occurs at 100 per cent power, 100 per cent flow, steady-state operating condition and consists of:

(a) The mechanical radial deflection at assembly

$$\begin{aligned} \delta_1 &= R_{\text{core}} + \Delta R_{\text{core}} - R_{\text{seal}} - (\Delta R)_{R.T.} = \frac{1}{2}(35.625 \pm 0.015 \\ &\quad + 0.061 - 35.525 \pm 0.010) - (0 \text{ to } 0.015) = 0.084 \text{ to } 0.123 \end{aligned}$$

- where R_{core} = Nominal Core Radius
 ΔR_{core} = Increase of R_{core} due to bundling based on NRX-A2 test data.
 R_{seal} = Radius of sealing point of free seal.
 $(\Delta R)_{\text{R.T.}}$ = Increase of R_{seal} when seal is assembled to core support plate.

- (b) The radial contraction of the dome end seal (from R.T. to operating temperature) at core support plate location

$$\delta_2 = (\Delta R)_P = -0.0519 \quad (\text{Ref. 4}) .$$

- (c) The radial thermal expansion of the core at station 0 (based on NRX-A2 test data)

$$\delta_3 = 0.009 \text{ in.}$$

Then the maximum finger deflection becomes:

$$\begin{aligned} \delta_{\text{total}} &= \delta_1 - \delta_2 + \delta_3 = (0.084 \text{ to } 0.123) + 0.0519 + 0.009 \\ &= 0.145 \text{ to } 0.184 . \end{aligned}$$

Assuming the finger fixed and rotation $\theta = 0$ at Point C (Figure 10), the stress at that point of each leaf becomes:

$$\sigma = \frac{M}{I/C} = \frac{3}{2} \left(\frac{\delta_{\text{tot}} E t}{\ell^2} \right) = \frac{(3)}{(2)} \frac{8(11.25)(10^6)(10.032)}{(2.016)^2} = 1,133,0008$$

(positive at inner, negative at outer fiber of leaves)

where

ℓ = finger length; t = finger thickness, Figure 9.

Figure 9. Dome End Seal

Seal - finger stress due to the pressure difference ΔP :

Figure 10. Seal - Finger Under Pressure Load ΔP

Then:

- (a) Stress resulting from mechanical radial deflection (δ_1) at assembly

$$\sigma_1 = \pm 11,160 \text{ to } \pm 16,350 \text{ psi.}$$

- (b) Stress resulting from the radial contraction (δ_2) of the dome end seal (at operating condition)

$$\sigma_2 = \pm 6,910 \text{ psi.}$$

- (c) Stress resulting from the radial thermal expansion of the core (δ_3)

$$\sigma_3 = \pm 1,200 \text{ psi.}$$

Total Stress:

$$\sigma_{\text{total}} = \pm 19,270 \text{ to } \pm 24,460 \text{ psi.}$$

After assembly (i.e., shipping), the core could assume some eccentricity due to a shock loading. The maximum expected acceleration of 4 g would cause a radial core deflection of $\delta_4 = 0.1167$ in. (Ref. 12). In that case, the total deflection and stress amount to:

$$\delta_T = \delta_1 + \delta_4 = (0.084 \text{ to } 0.123) + 0.117 = 0.201 \text{ to } 0.240$$

$$\sigma_T = (\pm 11,160 \text{ to } \pm 16,350) + 15,510 = 26,670 \text{ to } 31,860 \text{ psi}$$

Seal Finger Stress Due to Pressure Difference: For a finger with one end fixed and one end simply supported under a uniform pressure load ΔP (Figure 10), the bending moment in one leaf is:

$$M = \frac{\Delta P}{2} \ell \left(\frac{3}{8}x - \frac{x^2}{2\ell} \right) \text{ in. lb/ in. of width}$$

The maximum moment occurs at the fixed end C and is:

$$M_C = -\frac{1}{16} \Delta P \ell^2$$

Then:

$$\sigma_c = \frac{1}{2} \frac{M}{I/C} = \frac{1}{2} \frac{(20)(2.032)^2}{(16)(1.647)(10^{-4})} = \frac{1}{2} 31,350 \text{ psi} .$$

Where

$$(I/C) = \left[\frac{(35.485)\pi - (120)(0.032)}{(35.485)\pi} \right] \frac{(0.032)^2}{(6)} = (1.647)(10^{-4}) \text{ in.}^3/\text{in.}$$

width of
one leaf

The bending stress σ_c is positive at the outer and negative at the inner fiber.

The stress from mechanical deformation is superimposed on the stresses from the pressure differences and the results plotted on Figure 11, which shows the bending stress in the outer fiber of each seal finger leaf versus the finger length for several conditions. The stress in the inner fiber of each seal finger leaf has the same magnitude but the opposite sign. Comparing the stresses of Figure 11 with the yield strength of Table 11 it can be seen that all stresses are less than σ_y . The seal, therefore, should not cause any stress problem.

The stresses calculated and shown in Figure 11 are conservative since the actual dome end seal will assume a small angle at Point C, therefore reducing the finger stress.

CURVE No.	1	2	3	4	5	6
ASS'Y COND.	X	-	X	-	-	-
OPER. TEMP.	-	-	-	X	-	X
SHOCK LOAD	0	4g	4g	0	0	0
ΔP	0	0	0	0	20	20

Figure 11. Seal Finger Bending Stress

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