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36

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UNSEATING OF INNER REFLECTOR AND CLUSTERS

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(REVISION A)

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ABSTRACT

This report, Revision A of WANL-TME-422, presents an evaluation of the possibility of inner reflector lift off using the most recent pressure distributions from WANL-TNR-128. It shows that for the most conservative loading at full power, steady-state operation lift off cannot occur.

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CONTENTS

<u>SECTION</u>		<u>PAGE NO.</u>
	Abstract	ii
	List of Illustrations	iv
	List of Symbols	v
I.	Introduction	1
II.	Detailed Analysis	2
	Part A: Inner Reflector Lift Off	2
	Part B: Fuel Elements and Clusters	21

LIST OF ILLUSTRATIONS

<u>FIGURE NO.</u>		<u>PAGE</u>
Figure 1	Inner Reflector System	3
Figure 2	Plunger Pin Loads	5
Figure 3	Seal Block Forces	11
Figure 4	Filler Strip Model	16
Figure 5	Axial Forces Acting on Inner Reflector System	18
Figure 6	Forces Acting on Typical Fuel Element	21
Figure 7	Fuel Element Cross Sections	23
Figure 8	Axial Forces Acting on Outer Row of Fuel Element Clusters and Filler Strips	25
Figure 9	Forces on Filler Blocks	26
Figure 10	Forces Acting on Fuel Element Clusters	29

LIST OF SYMBOLS

For the reader's convenience the various symbols used throughout this report are listed below:

A_c	Contact surface area between filler strips (or filler blocks) and core per inch of circumference,	in. ² /in.
A_{fe}	Cross sectional area of fuel element (s),	in. ²
A_{fs}	Filler strip (or filler block) cross sectional area per inch of circumference,	in. ² /in.
B_1, B_2, B_3, B_4	Internal reactions used in plunger analysis,	lb.
C	Perimeter of fuel element (s),	in.
F_d	Dome end seal force,	lb/in.
F_h	Inner reflector hold down spring force,	lb.
F_i	Force acting on inner surface,	lb.
F_n	Axial force between n th seal row and inner reflector ledge,	lb/in.
F_N	Reaction at nozzle interface (includes seal force)	lb.
F_s	Axial spring force	lb/spring
K	Lateral support spring rate	lb/in.
L	Length of fuel element (s)	in.
P_{in}	Normal force at contact surface of plunger and n th seal row,	lb/in.
P'_{in}	Normal force acting on inner end of plungers at n th seal row,	lb/plunger

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P'_{on}	Normal lateral support spring force acting on outer end of plunger at n^{th} seal row,	lb/plunger
Q_n	Normal force between seal at n^{th} row and filler strips	lb/in.
R_{i1}	Inner radius of inner reflector cylinder at room temperature	in.
R_n	Radius to point of action of force at n^{th} seal row at full power, steady-state temperature,	in.
R_{o1}	Outer radius of inner reflector assembly at room temperature including assembly barrel thickness,	in.
R_{o2}	Outer radius of filler strips at room temperature,	in.
R_s	Radius to seal point on nozzle interface seal based on room temperature dimensions,	in.
V	Axial force between filler strip ledge and 18^{th} row of seals,	lb/in.
W	Weight of fuel element (s) and/or support block (s),	lb.
W_1	Total weight of inner reflector assembly,	lb.
W_2	Weight of filler strips (or filler blocks) per inch of circumference,	lb/in.
W_3	Weight of one row of seals per inch of circumference,	lb/in.
a	Length,	in.
b_1, b_2, b_3	Lengths between reactions in plunger analysis,	in.
d_n	Axial dimension locating force Q_n ,	in.
h	Axial height of seal block,	in.
k_n	Radial ledge width on n^{th} seal,	in.

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l_n	Axial dimension locating force P_{in} ,	in.
p_1	Core inlet plenum pressure,	psia
p_2	Inner reflector outlet plenum pressure,	psia
p_3	Core outlet plenum pressure,	psia
p_4	Inner reflector inlet plenum pressure,	psia
p_b	Bundling pressure,	psi
p_{ib}	Equivalent inlet band bundling pressure,	psi
p_{xn}	Pressure existing at dome surface of n^{th} seal,	psia
p_{xn}'	Pressure existing between filler strip and core at axial location corresponding to n^{th} seal row,	psia
p_{yn}	Pressure external to plunger pin at n^{th} seal row,	psia
s	Radial distance from plunger centerline to point of action of force P'_{in}	in.
t_1	Plunger pin diameter,	in.
t_2	Radial thickness of seal,	in.
α	Angle of sloped surface at plunger contact point on seal,	degrees
β	Angle of filler strip inner surface,	degrees
δ	Deflection of lateral support spring,	in.
ξ	Radial dimension locating force P_{in} on seal,	in.
μ	Coefficient of friction	
τ	Fluid shear force acting on outside of inner reflector assembly barrel,	lb.

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I. INTRODUCTION

This report determines the minimum force acting at the inner reflector-nozzle interface for the reactor operating at its steady-state, full power condition. The analysis shows that even for the "worst possible" combination of system parameters a net compressive force exists at the nozzle interface, indicating that separation cannot occur.

The analysis used in substantiating this conclusion is conservative in that compensating strains produced in the components have been neglected. For example, the assumption of core motion forward or nozzle motion aft on which this analysis is predicated results in increased tie rod load which, in turn, tends to reduce the amount of separation that might occur at the interface. Therefore, if it had been found that a static force analysis predicted separation, it would then have been necessary to compute the strain magnitudes in order to determine if separation would actually occur.

While this report calculates only the smallest interface force, WANL-TME-518 presents the minimum, nominal, and maximum forces. However, in comparing these values, it must be remembered that the calculations in WANL-TME-518 were developed in a somewhat different fashion and were based on a slightly more conservative geometry than the minimum value quoted in this report. Therefore, the results of the two analyses are not directly comparable. Nevertheless, the range between the minimum and maximum interface forces presented in WANL-TME-518 would closely approximate the range that would have been obtained for this report if the assumptions used in both reports had been chosen in the same manner.

In addition to the lift off problem, this report also determines the amount of bundling pressure which can be exerted before friction causes the fuel elements to hang up on adjacent elements and a circumferential row of clusters to hang up on the filler strips.

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II. DETAILED ANALYSIS

A. INNER REFLECTOR LIFT OFF

The existence or absence of a force at the inner reflector-nozzle interface will determine whether or not there is an inner reflector lift off problem. As long as the force at the interface remains compressive, there is no tendency for the inner reflector to lift off.

It is possible to calculate the magnitude of this force in either of two ways, namely,

- 1) perform a static balance on the inner reflector assembly;
- 2) perform a static balance on the combined system of the inner reflector assembly, seals, and filler strips.

Since there are interactions between the forces on the various components of the system, the work involved is basically the same by either method. But the second approach relates the bundling pressure to the force at the nozzle interface which will prove to be of interest, therefore, that approach will be used.

Before proceeding with the force balance on the system (Figure 1) it is necessary to determine what forces are acting on the individual components. These individual static balances will be performed in the order they are used, i.e., the plunger pin analysis, the seal block analysis, and the filler strip analysis. The assumptions needed for the various components will be listed in the section on the corresponding component.

General assumptions affecting all parts of the analysis are as follows:

- 1) The choice of the direction of core motion. Three possibilities exist, viz., motion toward the dome end, no motion, or motion toward the nozzle end. It is obvious from Figure 1 that only motion toward the dome will result in friction forces acting to reduce the force at the nozzle interface. Therefore, that is the only logical choice.

Figure 1. Inner Reflector System

- 2) The choice of the coefficient of friction, μ . Recent test data has indicated that the average μ between graphite parts lies between 0.15 and 0.20. Once again referring to Figure 1, it can be seen that the largest friction force will result in the smallest reaction at the nozzle interface. Therefore, μ for this report is conservatively chosen as 0.25 and in all cases the maximum value of the friction force is used.
- 3) The pressure drops and external forces acting on the system will be chosen to give the smallest net force at the nozzle interface.

Plunger Pin Analysis

The geometry of the plunger pin and the forces acting on it are shown in Figure 2. Reference drawings on the plunger pins are:

Conoid Tip (rows 2-16, 18) 963B984 H03-
Flat Tip (rows 1 and 17) 963B727 H03A

The various dimensions are as follows:

- $r = 0.1875$ in. from drawing
- $\alpha = 0$ degrees on rows 1 and 17
- $\alpha = 30$ degrees on rows 2-16, and 18
- $s = 0$ in. on rows 1 and 17
- $s = 0.094$ in. on rows 2-16, and 18 from TME-518
- $b_1 = 0.341$ in. for rows 1 and 17 from TME-518
- $b_1 = 0.395$ in. for rows 2-16, and 18 from TME-518
- $b_2 = 0.434$ in. from TME-518
- $b_3 = 0.445$ in. from TME-518

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NOTE:

The plungers in Rows 1 and 17 are different from the one shown here; i.e., they are flat tipped and a and $s = 0$.

Figure 2. Plunger Pin Loads

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Assumptions

- 1) The location of action of the forces B_m ($1 \leq m \leq 4$) was established by assuming that contact between the plunger and the inner reflector occurs along the full length of the hole and results in a triangularly distributed pressure load of which the forces B_m are the resultants acting at the third point of the triangle, i.e., at $1/6$ the distance from either end of the inner reflector hole. The actual forces should act closer to the ends of the hole. Therefore, the forces B_m used herein result in friction forces slightly higher than the true values, thus making this a conservative assumption.
- 2) The true force system would have only one force at an angle to the coordinate axes in place of the forces B_1 and B_3 and similarly for the reactions B_2 and B_4 . But the system used is slightly easier to work with and gives conservative results because the friction forces are a little higher than those which would be obtained from the true system.
- 3) Ignore the small fluid shear force on the sides of the plunger.
- 4) The positive direction of forces is in the positive direction of the coordinate axes shown in Figure 2. The positive direction of moments is such that their vector using the right hand rule is in the direction of the positive coordinate axes.

The equations of equilibrium are now obtained below:

$$\sum F_x = 0$$

$$\begin{aligned} & -P'_{in} \cos \alpha_n + \mu P'_{in} \sin \alpha_n + \mu B_1 + \mu B_2 + \mu B_3 \\ & + \mu B_4 = -P'_{on} - \pi r^2 (p_{yn} - p_{xn}) \end{aligned} \quad (1)$$

$$\Sigma F_y = 0$$

$$-B_1 + B_2 = \mu P'_{on} \quad (2)$$

$$\Sigma F_z = 0$$

$$-P'_{in} \sin \alpha_n - \mu P'_{in} \cos \alpha_n + B_3 - B_4 = 0 \quad (3)$$

$$\Sigma M_y = 0$$

$$P'_{in} \cos \alpha_n (s) - \mu P'_{in} \sin \alpha_n (s) + (\mu r - b_1) B_3 \\ + (b_1 + b_2 - \mu r) B_4 = 0 \quad (4)$$

$$\Sigma M_z = 0$$

$$(\mu r - b_1) B_1 + (b_1 + b_2 - \mu r) B_2 = \mu (b_1 + b_2 + b_3) P'_{on} \quad (5)$$

By the simultaneous solution of Equations (1) through (5), the general solution for P'_{in} is obtained in the following form:

$$P'_{in} = \frac{P'_{on} \left[(b_2 + 2r\mu^3 + b_2\mu^2) + 2b_3\mu^2 \right] + b_2 \pi r^2 (p_{yn} - p_{xn})}{D} \quad (6)$$

where,

$$D = \left[(b_2 - \mu^2 b_2 + 2r\mu^3) \cos \alpha_n + (2\mu^2 r - 2b_2\mu) \sin \alpha_n \right. \\ \left. + 2\mu s (\cos \alpha_n - \mu \sin \alpha_n) \right] - b_1 (2\mu^2 \cos \alpha_n + 2\mu \sin \alpha_n) \quad (7)$$

$$P'_{on} = K \delta = K \left[\delta(\text{preload}) + \Delta R (\text{core}) - \Delta R (\text{inner reflector}) \right] \quad (8)$$

After obtaining the values of P'_{in} by Equation (6) at each of the seal row locations, those values are converted to values of P_{in} using the following relationship.

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$$P_{in} = \frac{36 P'_{in}}{2\pi R_n} \quad (9)$$

where,

$$R_n = R_{o2} + t_2 - \delta + \Delta R_n (\text{core}) \quad (10)$$

The calculations based on Equations (6) through (10) are located in Table 1.

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TABLE I

Calculation of Plunger Pin Forces

Row No.	Axial Position (in.)	Core ΔR (in.)	Reflector ΔR (in.)	b_1 (in.)	b_3 (in.)	P'_{on} ($\frac{lb}{plunger}$)	P_{yn} (psi)	P_{xn} (psi)	P'_{in} ($\frac{lb.}{plunger}$)	P_{in} (lb/in.)
1	2.938	0.011	-0.030	0.300	0.486	39.87	699.5	695.0	56.65	17.64
2	5.75	0.021	-0.029	0.345	0.495	41.10	703.2	693.5	129.09	40.27
3	8.562	0.032	-0.028	0.335	0.505	42.47	706.5	690.5	132.65	41.35
4	11.374	0.045	-0.027	0.323	0.517	44.11	709.8	686.4	136.66	42.57
5	14.186	0.055	-0.026	0.314	0.526	45.35	713.0	681.0	140.34	43.69
6	16.998	0.069	-0.026	0.300	0.540	47.27	716.0	674.8	144.82	45.06
7	19.810	0.086	-0.026	0.283	0.557	49.59	719.0	667.2	149.81	46.56
8	22.622	0.102	-0.027	0.266	0.574	51.92	722.0	658.7	154.71	48.04
9	25.434	0.116	-0.028	0.251	0.589	53.98	724.7	649.4	159.46	49.48
10	28.246	0.131	-0.030	0.234	0.606	56.31	727.4	639.6	164.26	50.93
11	31.058	0.143	-0.032	0.220	0.620	58.23	730.0	629.1	168.55	52.23
12	33.870	0.155	-0.035	0.205	0.635	60.28	732.5	618.3	172.95	53.55
13	36.682	0.166	-0.038	0.191	0.649	62.20	734.8	607.6	176.94	54.76
14	39.494	0.174	-0.042	0.179	0.661	63.84	737.0	596.7	180.61	55.87
15	42.306	0.179	-0.045	0.171	0.669	64.94	739.3	583.6	184.43	57.04
16	45.118	0.183	-0.050	0.162	0.678	66.17	741.4	571.0	188.10	58.16
17	47.930	0.185	-0.054	0.101	0.684	66.99	743.2	565.0	113.83	35.11
18	50.742	0.188	-0.059	0.148	0.692	68.09	745.0	562.5 560.0	191.49	59.19

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Seal Block Analysis

In Figure 3 which follows, the seal block which has the angled notch at the plunger pin contact surface is shown. Seal blocks of this type are used in rows 2 to 16 and 18. In rows 1 and 17, the contact surface is flat. The equations of static equilibrium which are developed will be applicable to all of the seal rows. However, it must be remembered when making the final calculations that the angle α for rows 2 to 16 and 18 is 30° , but for rows 1 and 17 it is zero. Also, the force V shown acting axially on the seal is the force transmitted by the filler strip ledge to seal row 18. Therefore, it enters into the force balance on that row only. Finally, from Figure 3 it is seen that there are only the three unknowns, F_n , Q_n , and d_n associated with seal rows 1 to 17 and one additional unknown force, V , acting on row 18. When the equations of static equilibrium are developed, it is found that the unknown distance d_n appears only in the summation of moments equation; and, the forces F_n and Q_n are completely determinable from the remaining equations, i.e., the summation of forces axially and radially. On seal row 18 which includes the additional unknown V , the forces F_n and Q_n are obtained in terms of V which is evaluated in the filler strip calculation which follows this section. Since the location of action of Q_n is not needed, only the two equations summation of forces axially and radially will be developed below. The development will be for a portion of the seal block having a circumferential length of one inch.

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Figure 3. Seal Block Forces

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Typical reference drawings on the seal blocks are:

Rows 2 to 16, and 18 977D843H09

Rows 1 and 17 977D980H02

Assumptions:

- 1) The force P_{in} is evenly distributed circumferentially along the seal block.
- 2) The hydraulic pressure drop along surfaces of the seal block which contact adjacent components is linear.
- 3) The choice of μP_{in} is completely arbitrary. It depends on the angular position of the pin. If the axis of the plunger is parallel to the bottom (sliding) surface of the seal block, it will even be zero. It has been chosen in the manner shown on Figure 3 to result in a conservative value of Q_n which later gives conservative values for p_b . In TME-518, this effect was not included.

The equations of static equilibrium are as follows:

$$\Sigma F (\text{axial}) = 0$$

$$\begin{aligned} & V (1) (\text{Seal Row 18 only}) + p_{xn} (t_2) (1) - \mu Q_n (1) \\ & - \left[p_{x(n+1)} (t_2) (1) + \left(\frac{p_{xn} - p_{x(n+1)}}{2} \right) (k_n) (1) \right] - F_n (1) \\ & + P_{in} \sin \alpha_n (1) + \mu P_{in} \cos \alpha_n (1) - W_3 (1) = 0 \end{aligned} \quad (11)$$

$$\Sigma F (\text{radial}) = 0$$

$$\begin{aligned} & Q_n (1) + \left(\frac{p_{xn} + p_{x(n+1)}}{2} \right) (h) (1) - \mu F_n (1) - p_{xn} (h) (1) \\ & - P_{in} \cos \alpha_n (1) + \mu P_{in} \sin \alpha_n (1) = 0 \end{aligned} \quad (12)$$

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By simultaneous solution of Equations (11) and (12), the following values of F_n and Q_n are obtained:

For rows 1 through 17

$$F_n = \left[\frac{1}{1 + \mu^2} \right] \left[\frac{P_{xn} - P_x(n+1)}{2} \right] (2t_2 - \mu h - k_n) + P_{in} \sin \alpha_n - \left[\frac{1}{1 + \mu^2} \right] W_3 \quad (13)$$

$$Q_n = \left[\frac{P_{xn} - P_x(n+1)}{2} \right] \left[h + \left(\frac{\mu}{1 + \mu^2} \right) (2t_2 - \mu h - k_n) \right] + P_{in} \cos \alpha_n - \left[\frac{\mu}{1 + \mu^2} \right] W_3 \quad (14)$$

For row 18 Equations (13) and (14) become:

$$F_{18} = \left[\frac{1}{1 + \mu^2} \right] \left[\frac{P_{x18} - P_3}{2} \right] (2t_2 - \mu h - k_{18}) + P_{i18} \sin \alpha_{18} - \left[\frac{1}{1 + \mu^2} \right] W_3 + \left[\frac{1}{1 + \mu^2} \right] V \quad (15)$$

$$Q_{18} = \left[\frac{P_{x18} - P_3}{2} \right] \left[h + \left(\frac{\mu}{1 + \mu^2} \right) (2t_2 - \mu h - k_{18}) \right] + P_{i18} \cos \alpha_{18} - \left[\frac{\mu}{1 + \mu^2} \right] W_3 + \left[\frac{\mu}{1 + \mu^2} \right] V \quad (16)$$

The calculated values of F_n and Q_n are listed in Table 2.

TABLE 2
Forces Acting on Seal Blocks

Row	p_{xn} (psi)	k_n (in.)	F_n (lb/in.)	Q_n (lb/in.)
1	695.0	0.296	0.30	18.81
2	693.5	0.368	20.65	37.20
3	690.5	0.378	21.38	39.00
4	686.4	0.390	22.20	41.06
5	681.0	0.399	22.87	42.65
6	674.8	0.413	23.75	44.91
7	667.2	0.430	24.58	46.90
8	658.7	0.447	25.38	48.78
9	649.4	0.462	26.10	50.40
10	639.6	0.479	26.85	52.17
11	629.1	0.493	27.46	53.50
12	618.3	0.508	28.04	54.56
13	607.6	0.522	28.59	55.74
14	596.7	0.534	29.33	58.36
15	583.6	0.542	29.81	58.98
16	571.0	0.551	29.64	54.92
17	565.0	0.419	0.39	37.04
18	562.5 560.0	0.565	29.78 + 0.941 V	53.14 + 0.235 V

$$\sum_{n=2}^{18} Q_n = 829.3 + 0.235 V$$

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Filler Strip Analysis

The filler strips in the NRX-A reactor duplicate themselves every 60 degrees. Within the 60 degree sector, however, they have whatever shape is required to make the core periphery round. Since both the contact surface area along the inside of the filler strips and the cross sectional area of the filler strips enters into the force balance, the analysis is simplified by assuming that both areas are homogeneously distributed around the core periphery, i.e., the total area in each case is divided by the circumference at the outside diameter of the core. Then the filler strips are analyzed per inch of circumference. Also, since the net force in the tangential direction is zero, the shape of the filler strip is assumed to be symmetrical (Figure 4) thus eliminating tangential forces.

Similarly, the forces acting between the filler strips and the core are known to vary somewhat from one axial position to the next. But only the total effect is considered herein, i.e., it is assumed that an evenly distributed bundling pressure, p_b , acts between the filler strips and the core.

The model which is analyzed is shown in Figure 4.

$$\Sigma F (\text{axial}) = 0$$

$$(p_{x2} - p_3) A_{fs} (1) - \mu p_b A_c (1) - V (1) + \sum_{n=2}^{18} \mu Q_n (1) - W_2 (1) = 0 \quad (17)$$

$$\Sigma F (\text{radial}) = 0$$

$$p_b A_c \cos \beta(1) - \sum_{n=2}^{18} Q_n (1) - (p_{xn} - p'_{xn}) A_c \cos \beta(1) = 0 \quad (18)$$

where,

$$A_{fs} = \frac{59.3}{2\pi (17.8125)} = 0.53 \text{ in.}^2/\text{in.}$$

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Figure 4. Filler Strip Model

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Also, the total contact area between the filler strips and core is 6425 in.^2 . But, as can be seen in Figure 1, the contact area for the purpose of this calculation is reduced by the amount covered by the filler blocks at the dome end of the core. Therefore,

$$A_c = \left(\frac{47.25}{52} \right) \frac{6425}{2\pi (17.8125)} = 52.1 \text{ in.}^2/\text{in.}$$

By reference to Figure 4, the angle β may be calculated as follows:

$$A_c = 52.1 = (47.25) (2) \left(\frac{0.5}{\cos \beta} \right)$$

$$\cos \beta = \frac{47.25}{52.1}$$

$$\beta = 25^\circ$$

And finally, the average value of the radial hydraulic pressure drop, $p_{xn} - p'_{xn}$, is given as 0.2 psi in WANL-TNR-128 (Reactor Analysis of NRX-A Volume III).

Substituting from the previous calculations, Equations (17) and (18) have been solved to give the following values of V and p_b .

$$V = 44.53 \text{ lb/in.}$$

$$p_b = 18.00 \text{ psi}$$

Nozzle Reaction:

Having obtained the above values it is now possible to calculate the reaction at the nozzle by performing a static balance on the inner reflector assembly, seal, filler strip system. The configuration of the system is given in Figure 5. The dimensions needed in this calculation are:

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Figure 5. Axial Forces Acting on Inner Reflector System

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$$R_{o2} = 17.8125 \text{ in.}$$

$$R_{i1} = 18.0625 \text{ in.}$$

$$R_{o1} = 19.866 \text{ in.}$$

$$R_1 = R_{i1} - \Delta R_1 (\text{inner reflector}) + 1/2 k_1$$

$$R_s = 18.832 \text{ in.}$$

Since all forces except the nozzle reaction have been calculated, only the equation of axial equilibrium need be written for this system. It is as follows:

$$\Sigma F (\text{axial}) = 0$$

$$\begin{aligned}
 & F_h + (p_2 - p_4) \pi (R_{o1}^2 - R_s^2) + (p_2 - p_3) \pi (R_s^2 - R_{i1}^2) \\
 & + (p_{x2} - p_3) \pi (R_{i1}^2 - R_{o2}^2) + (p_{x2} - p_3) \Sigma A_{fs} \\
 & - \mu p_b \Sigma A_c - \Sigma_c W_2 - \sum_{n=2}^{18} (\Sigma_c W_{3n}) - W_1 + 2\pi R_1 F_1 \\
 & - \Sigma \mu P'_{i1} - \tau = F_N
 \end{aligned} \tag{19}$$

The assumptions used in the calculation which follows are:

- 1) The nozzle interface seal radius, R_s , is chosen at its innermost point. This is a conservative assumption since it results in the largest upward acting pressure drop force.
- 2) The radii defining areas for pressure drop forces are taken as the room temperature values.

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- 3) The barrel shape of the core is neglected. This results in a slight nonconservatism estimated to be at most about 400 pounds.

Substituting values into Equation (19), the calculated value for the minimum nozzle reaction force F_N is:

$$F_N = 4590 \text{ lb.}$$

It is to be noted that this number was calculated based on all load factors being chosen in the conservative direction. It is practically impossible that this can occur. It is much more likely that some favorable as well as some adverse tolerances will occur. Considering this factor, the net reaction at the nozzle, when calculated for nominal conditions, i.e., $\mu = 0.2$, nominal pressure drops, nominal hold down spring force, and minimum sealing radius on nozzle interface seal, is:

$$F_N (\text{nominal}) = 21,500 \text{ lb.}$$

In addition, the results of TME-518 point out the large range between the minimum and maximum values of the nozzle interface reaction (5110 to 32,170 lb). However, the calculations were for an unbanded core design and included conservative assumptions (necessary at the time because complete information concerning pressure distributions was not available) which prevent direct comparison of the values in TME-518 and the values herein. But the range should be fairly close and the conclusion can be drawn that values even higher than the specified nominal value of 21,500 pounds may be expected.

It is finally concluded that there is no inner reflector lift off problem in the NRX-A reactor design at steady-state, full power operation.

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B. FUEL ELEMENTS AND CLUSTERS

In addition to the foregoing, it is desirable to know how bundling pressure affects the axial movement of the fuel elements. In the following, it will be determined what critical bundling pressure will prevent a fuel element or group of elements from seating on the fuel cluster support block when acted on by the nominal core pressure drop of 129 psi.

Individual Elements

In its upward firing position, the fuel element is acted on by the force system shown in Figure 6.

Figure 6. Forces Acting on Typical Fuel Element

Summing forces vertically, the following equations are obtained:

$$W + \mu p_b L C - F_s - (p_1 - p_3) A_{fe} = 0 \quad (21)$$

From which:

$$p_b = \frac{F_s + (p_1 - p_3) A_{fe} - W}{\mu L C} \quad (22)$$

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Before evaluating Equation (22), the assumptions which are made and the needed parameters will be listed.

Assumptions:

- 1) For combinations of elements of less than a full cluster, disregard the spring load, F_s
- 2) The value of bundling pressure, p_b , is uniform over the axial length of the element.
- 3) As in the previous section the coefficient of friction, μ , will be taken as 0.25.

Value and source of parameters:

F_s = 58.54 lb/spring in the operating position (Drawing 963B483-)

L = 51.25 in. (Drawing 909E263G)

W (tie rod element) = 1.034 lb (Drawing 909E263G)

W (fueled element) = 1.44 lb (Drawing 936J654E)

W (partial element) = .8 lb (Drawing 909E009F)

W (cluster plate) = .074 lb (Drawing 909E303D)

W (regular support block) = .31 lb (Drawing 936J209J)

W (J3 support block) = .80 lb (Drawing 936J584D)

W (J4 support block) = .43 lb (Drawing 936J585D)

W (J5 support block) = .43 lb (Drawing 936J586D)

W (J6 support block) = .32 lb (Drawing 936J587E)

W (J7 support block) = .32 lb (Drawing 936J588D)

W (J8 support block) = .29 lb (Drawing 936J589D)

W (J9 support block) = .80 lb (Drawing 936J590D)

The cross sectional geometry of typical fuel elements is shown in Figure 7. Complete dimensional data for the fuel elements can be obtained from Drawings 936J654E and 909E263G.

Figure 7. Fuel Element Cross Sections

The fluid shear on the sides of the cooling holes equals the pressure drop through the element. Therefore, the area acted on by the axial pressure drop through the fueled element is the same as that of a solid hexagon. In the tie rod element, however, the fluid shear in the large central hole is reacted by the tie rod. Therefore, the hole area should be subtracted from the hexagon. Using room temperature dimensions, the length "a" and these areas are as follows:

$$a = (.754) \tan 30^\circ = .4353 \text{ inches}$$

$$A_{fe} \text{ (fueled element)} = (.25) (6) (.754)^2 \tan 30^\circ = .492 \text{ in.}^2$$

$$A_{fe} \text{ (tie rod element)} = .492 - (.25) \pi (.361)^2 = .390 \text{ in.}^2$$

Values of p_b , for combinations of elements of less than a full cluster, may now be obtained by setting $F_s = 0$ in Equation (22). In this manner Equation (23) is obtained and is evaluated in Table 3.

$$P_b = \frac{(p_1 - p_3) A_{fe} - W}{\mu L C} \quad (23)$$

TABLE 3

Bundling Pressure Which Would Hang Up Fuel Elements

N_e	$(p_1 - p_3)$ (psi)	A_{fe} (in. ²)	W (lb.)	C (in.)	$\mu L C$ (in. ²)	P_b (psi)
1	129	.390	1.034	6a = 2.613	33.48	1.47
2	129	.882	2.474	10a = 4.353	55.77	2.00
3	129	1.374	3.914	12a = 5.224	66.93	2.59
4	129	1.866	5.354	14a = 6.094	78.08	3.01
5	129	2.358	6.794	16a = 6.965	89.24	3.33
6	129	2.850	8.234	18a = 7.836	100.40	3.58

N_e = number of elements being considered.

From the above calculation, it can be seen that the nominal preload bundling pressure of 3.8 psi can hang up six elements in any cluster.

Individual Clusters

The above analysis could be extended in a similar manner to full clusters.

Circumferential Row of Clusters

Consider now the hypothetical case in which the outermost circumferential row of clusters expands more than the remainder of the core. Then their hoop expansion will cause clearance to develop between this circumferential ring and the inner core making the radial reaction F_i , between the ring and inner core zero as shown in Figure 8.

Figure 8. Axial Forces Acting on Outer Row of Fuel Element Clusters and Filler Strips

For this case, a gap could develop between fuel element clusters and the support blocks if the inner reflector moves toward the dome end and the friction force between the fuel element clusters and the filler strips and blocks is sufficiently large to overcome the pressure drop force on the clusters. All of the forces needed to check this condition have been determined in previous sections of the report except for the frictional forces between the filler blocks and the fuel elements which will now be calculated. The filler block force distribution is shown in Figure 9.

Figure 9. Forces on Filler Blocks

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Using the same notation as previously, the equations of axial and radial static equilibrium for a one inch circumferential section are:

$$(p_{x2} - p_1) A_{fs} + \mu Q_1 - \mu F_d - k (\mu p_b A_c) + W_2 = 0 \quad (24)$$

$$p_b A_c \cos \beta + \int_0^l (p_i - p_o) dA - Q_1 - F_d - (p_{ib}) (l) = 0 \quad (25)$$

The parameters needed to evaluate Equation (24) are:

$$p_{x2} = 694.1 \text{ psi}$$

$$p_1 = 681 \text{ psi (minimum value, reference WANL-TME-518)}$$

$$Q_1 = 18.8 \text{ lb/in.}$$

$$F_d = 19.9 \text{ lb/in.}$$

$$W_2 = \frac{4.5}{49.25} (1.69) = 0.155 \text{ lb/in.}$$

The friction force acting on the clusters is then calculated to be:

$$k \mu p_b A_c = (694.1 - 681) (.53) + (.25) (18.8) - (.25) (19.9) + 0.155$$

$$k p_b A_c = 6.84 \text{ lb/in.}$$

Using Equation (25), the bundling pressure under the filler blocks was calculated to be between 22.2 and 31.2 psi depending upon whether the minimum or maximum value of p_{ib} is used. For either value it is seen that the maximum friction force based on $\mu = 0.25$ is in excess of the friction force required for equilibrium as calculated by Equation (24) i.e.,

$$f = \mu p_b A_c$$

$$f_{\max.} \text{ (based on minimum } p_b) = (.25) (22.2) (4.97) = 27.6 \text{ lb/in.}$$

$$f_{\max.} \text{ (based on maximum } p_b) = (.25) (31.2) (4.97) = 38.8 \text{ lb/in.}$$

Since these values exceed the 6.84 lb/in. calculated above, the full 6.84 lb/in. will act on the fuel element clusters.

The forces acting on the clusters are shown in Figure 10.

The basic question to be answered is whether the combination of the pressure drop and spring forces is sufficient to overcome the frictional resistance and the weight. In this outer row of clusters there are 90 springs whose total force contribution is then:

$$F_s = (90) (58.54) = 5270 \text{ lb.}$$

The pressure drop force will consist of the pressure drop times the total area of all the individual elements in the row which is 239.9 in.² Therefore, the pressure force is:

$$(p_1 - p_3) A = (681 - 560) (239.9) = 29,000 \text{ lb.}$$

The total weight of the fuel elements in the circumferential row is 688.3 lb. This when added to the friction forces determines the total resisting force as follows:

$$\begin{aligned} W + \left[\mu p_b A_c \text{ (at filler strips)} + k \mu p_b A_c \right] 2\pi (17.8125) \\ = 688.3 + \left[(.25) (18) (52.1) + 6.84 \right] 2\pi (17.8125) = 27,688 \text{ lb.} \end{aligned}$$

But this is less than the combination of 29,000 lb + 5270 lb = 34,270 lb. which acts on the clusters due to the pressure and spring forces. Therefore, the clusters will remain seated on their support blocks for steady-state, full power operating conditions.

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Figure 10. Forces Acting on Fuel Element Clusters

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