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**CLADDING STRESSES INDUCED
DURING DISASSEMBLY OF EBR-II
EXPERIMENTAL-FUEL SUBASSEMBLIES**

by

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TABLE OF CONTENTS

	<u>Page</u>
ABSTRACT	9
I. INTRODUCTION	10
II. STRUCTURAL-ANALYSIS MODEL	13
III. SOLUTION	15
A. General Solution	15
B. Particular Solutions	20
1. Three Spring Supports	20
2. Two End Spring Supports	22
3. One Central Spring Support	23
4. Prescribed Deflection	24
C. Solutions for Specific Boundary Conditions	24
D. Bending Stresses in Cladding	27
IV. RESULTS	27
A. Subassembly X062	28
B. Subassembly X059A	40
C. Subassembly X069A.	59
V. BOUNDS ON THE SOLUTION	59
A. Lower Bounds for Maximum Cladding Bending Stress	59
B. Upper Bounds for Maximum Cladding Bending Stress	61
1. Subassembly X062	61
2. Subassembly X059A	62
3. Subassembly X069A	62
VI. DISCUSSION	64
VII. CONCLUSIONS	66
ACKNOWLEDGMENTS	67

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
1	Irradiation-Induced Subassembly Swelling	11
2	Interference between Unswollen Hexagonal-can Wall and Swollen Outer-row Fuel Elements during Disassembly	12
3	Interference Loads Imposed on an Outer-row Fuel Element during Disassembly	14
4	Longitudinal Section of a Typical Subassembly	16
5	Cross Sections of a Typical Subassembly at Three Locations along a Single Spacer-Wire-Pitch Length	17
6	Structural Idealization of the Outer-row Fuel Element	18
7	Influence of End-support Stiffness on Fuel-element Bending Moment for Fuel Elements in Subassembly X062 (Case I)	32
8	Influence of End-support Stiffness on Support Reaction for Fuel Elements in Subassembly X062 (Case I)	33
9	Influence of End-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassembly X062 (Case I)	34
10	Influence of End-support Stiffness on Support Deflection for Fuel Elements in Subassembly X062 (Case I)	35
11	Influence of Central-support Stiffness on Bending Moment of Fuel Elements in Subassembly X062 (Case II)	36
12	Influence of End-support Stiffness on Support Reaction for Fuel Elements in Subassembly X062 (Case II)	37
13	Influence of Central-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassembly X062 (Case II)	38
14	Influence of Central-support Stiffness on Support Deflection for Fuel Elements in Subassembly X062 (Case II)	39
15	Influence of End-support Stiffness on Bending Moment of Fuel Elements in Subassembly X062 (Case V)	43
16	Influence of End-support Stiffness on Support Reactions for Fuel Elements in Subassembly X062 (Case V)	44
17	Influence of End-support Stiffness on Bending Stress of Cladding Fuel Elements in Subassembly X062 (Case V)	45
18	Influence of End-support Stiffness on Support Deflection of Fuel Elements in Subassembly X062 (Case V)	46
19	Influence of End-support Stiffness on Bending Moment of Fuel Elements in Subassemblies X059A and X069A (Case I)	47

LIST OF FIGURES (contd)

<u>No.</u>	<u>Title</u>	<u>Page</u>
20	Influence of End-support Stiffness on Support Reaction for Fuel Elements in Subassemblies X059A and X069A (Case I)	48
21	Influence of End-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassemblies X059A and X069A (Case I)	49
22	Influence of End-support Stiffness on Support Deflection for Fuel Elements in Subassemblies X059A and X069A (Case I)	50
23	Influence of Central-support Stiffness on Bending Moment for Fuel Elements in Subassemblies X059A and X069A (Case II)	51
24	Influence of Central-support Stiffness on Support Reaction for Fuel Elements in Subassemblies X059A and X069A (Case II)	52
25	Influence of Central-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassemblies X059A and X069A (Case II)	53
26	Influence of Central-support Stiffness on Support Deflection for Fuel Elements in Subassemblies X059A and X069A (Case II)	54
27	Influence of End-support Stiffness on Bending Moment of Fuel Elements in Subassemblies X059A and X069A (Case V)	55
28	Influence of End-support Stiffness on Support Reaction for Fuel Elements in Subassemblies X059A and X069A (Case V)	56
29	Influence of End-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassemblies X059A and X069A (Case V)	57
30	Influence of End-support Stiffness on Support Deflection for Fuel Elements in Subassemblies X059A and X069A (Case V)	58

LIST OF TABLES

<u>No.</u>	<u>Title</u>	<u>Page</u>
I	Descriptions of Subassemblies X062, X059A, and X069A	29
II	Maximum Cladding Stresses and Support Deflections for Subassembly X062	30
III	Maximum Cladding Stresses and Support Deflections for Subassembly X059A	41
IV	Maximum Cladding Stresses and Support Deflections for Subassembly X069A	60
V	Results for Cases I, II, and IV for a Relative Deflection of 6 Mills on an Outer-row Fuel Element for Subassembly X059A	63

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ABSTRACT

During disassembly of irradiated fuel subassemblies after irradiation in EBR-II, resistance to removal of the hexagonal can from the swollen fuel-element cluster has been observed. The stresses on the fuel-element cladding during disassembly have been evaluated. The primary parameters that influence cladding interference stresses are: the pull force on the hexagonal can, the coefficient of friction between the can wall and the cladding; the stiffness of the fuel-element cluster; and the spacer-wire pitch. A simple structural-analysis model has been selected and certain assumptions have been made, with resulting estimates of maximum cladding bending stresses. Current estimates can be improved if the rotational and lateral stiffnesses of the fuel-element cluster are known. A simple test to determine the lateral stiffness of the cluster is possible.

I. INTRODUCTION

Subassemblies of mixed-oxide fuel elements irradiated in EBR-II have been observed to bulge preferentially near the core midplane. This bulging is due at least in part to radiation-induced swelling of the fuel-element cladding. Swelling also has been observed in hexagonal-can material, again nearer the core midplane. The resulting geometry of the irradiated assembly is shown in Fig. 1.

Fueled experiments normally are disassembled by cutting the hexagonal can laterally below the fuel-element grid plate, as shown in Fig. 1, and pulling the can vertically over the fuel-element cluster. During this operation the cluster is cooled by forced air flowing through the subassembly. Figure 2 shows a subassembly with the hexagonal can partly removed and the possible interaction between the lower portion of the can and the swollen central region of the fuel-element cluster. If the can jams in this position, the fuel elements could overheat in subassemblies of some designs because of inadequate air flow. Therefore, removing the can without interruption is desirable.

The can may be removed without interruption by either pulling it free with a large vertical force or slitting one edge of the can to minimize the interference and then pulling with a moderate force. The latter procedure is expected to avoid interference stresses due to can removal but is operationally undesirable because of increased handling and a risk of damaging the fuel elements during the slitting. Moreover, cooling of the subassembly during this procedure may be reduced by bypass of air through the slit. The simpler procedure of removing the can by applying the necessary vertical force may result in large interference stresses in the fuel-element cladding. In addition, the vertical pull force on the can cannot exceed the machine capacity, which is currently limited to a force of 2000 lb by a load-cell-controlled interlock. Although a possible solution for this jamming problem is to increase the machine capacity (i.e. override the interlock), the relationship between the pull force on the hexagonal can and the maximum interference stress in the fuel-element cladding during this operation must be determined.

Fig. 1. Irradiation-induced Subassembly Swelling

Fig. 2. Interference between Unswollen Hexagonal-can Wall and Swollen Outer-row Fuel Elements during Disassembly

This report presents estimates of maximum cladding stresses for several postulated disassembly conditions during removal of the integral hexagonal can. Although the structural model is somewhat general, it has been necessary to make certain assumptions related to: (1) the transfer of a vertical pull load on the can to a series of lateral concentrated and distributed loads on the fuel elements; the rotational and lateral stiffnesses of the fuel-element cluster; and the nature of contact between the can and the outer row of fuel elements and between the outer row of elements and their inner neighbors.

The structural model selected yields expressions for cladding stresses, bending moments, and deflections as functions of several variables. The variables are: the pull force exerted on the can by the disassembly machine; the coefficient of friction between the fuel-element cladding and the can wall; the number of fuel elements in contact with the can wall; the thickness and pitch of the spacer wire; the effective lateral stiffness of the inner fuel-element cluster; and the manner of load distribution and contact between the outer fuel elements and the can wall.

II. STRUCTURAL-ANALYSIS MODEL

The basic load-transfer path through the hexagonal can and the fuel-element cluster can be traced by examining the geometry of the irradiated fuel subassembly during disassembly, shown in Fig. 2. The axial load on the can is transferred to the outer row of fuel elements through frictional shear at locations of interference between the can wall and the swollen fuel-element cluster, as shown in Fig. 3. The can will slide over the fuel-element cluster when the product of the axial pull force, F , and the coefficient of friction, μ , between the fuel-element cladding and the can wall just exceeds the sum of all lateral interference forces, P_1 and w_1 . Figure 3 shows that the total axial load is eventually transferred to the fuel-element cluster and is reacted at the subassembly grid. The fuel elements in direct contact with the can during disassembly, therefore, can be represented as continuous beam-columns on an elastic foundation, loaded axially in shear, because of the pull force on the hexagonal can, and in lateral bending because of the interference between the can and the fuel elements.

Fig. 3. Interference Loads Imposed on an Outer-row Fuel Element during Disassembly

The elastic foundation for the outer row of fuel elements is provided by the cluster of inner fuel elements because of their ability to deflect under load. This deflection could result from rigid-body displacements or bending and torsional deflections of the inner fuel elements, or both. Figures 4 and 5 show longitudinal and lateral sections across typical subassemblies. From these sections the inner fuel elements apparently will deflect when subjected to a lateral force orthogonal to the can wall, but the net deflection of the elements may not be orthogonal to the wall. Furthermore the ability of the fuel-element cluster to accommodate lateral inward deflections will decrease as its swelling increases.

A simplified structural-analysis model is obtained by assuming that the cladding of the outer fuel elements is in contact with the wall of the hexagonal can at three locations corresponding to the spacer-wire pitch, as shown in Fig. 6. The elastic foundation provided by the cluster of inner fuel elements is represented by three elastic springs assumed to be located at three locations of contact between the outer fuel elements and their immediate inner neighbors. The influence of the axial load on the bending stresses in the fuel element is neglected in this analysis, and the lateral loading on the fuel element is assumed to be symmetrical about the central-contact location, C. Also assumed is that linear contact, along a length $2l$, exists between the hexagonal can and the fuel element at the central location, C, and point contact exists at the adjacent locations, A and B.

III. SOLUTION

A. General Solution

A general solution for the problem described above is obtained by using the area-moment theorems of structural analysis*. By using standard notations and Fig. 6, this solution is as follows:

*L. C. Maugh, Statically Indeterminate Structures, 2nd ed. Wiley, New York (1964).

Fig. 4. Longitudinal Section of a Typical Subassembly

Fig. 5. Cross Sections of a Typical Subassembly at Three Locations along a Single Spacer-wire-pitch Length

Fig. 6. Structural Idealization of the Outer-row Fuel Element

From equilibrium,

$$R_B + R_C = P + wL \text{ and} \quad (1)$$

$$M_B + M_C = Ps + \frac{wL^2}{2} - R_B s. \quad (2)$$

From the area-moment theorems:

$\theta_{B/C}$ = slope between points B and C;

$$\text{or } \theta_{B/C} = \frac{R_B s^2}{2EI} + \frac{M_B s}{EI} - \frac{Ps^2}{2EI} - \frac{wL^3}{6EI}; \text{ and} \quad (3)$$

$d_{B/C}$ = deflection between points B and C;

$$\text{or } d_{B/C} = \frac{R_B s^3}{3EI} + \frac{M_B s^2}{EI} - \frac{Ps^3}{3EI} - \frac{wL^3}{6EI} \frac{(s-L)}{4}. \quad (4)$$

From boundary conditions, if the slopes at A,B, and C are assumed to be zero,

$$\theta_{B/C} = 0. \quad (5)$$

Solution of the simultaneous equations (1) through (5) yields:

$$R_B = P + \frac{wL^3}{s^2} - \frac{wL^4}{2s^3} + \frac{12EI}{s^3} d_{B/C}; \quad (6)$$

$$R_C = wL - \frac{wL^3}{s^2} + \frac{wL^4}{2s^3} - \frac{12EI}{s^3} d_{B/C}; \quad (7)$$

$$M_B = \frac{wL^4}{4s^2} - \frac{wL^3}{3s} - \frac{6EI}{s^2} d_{B/C}; \text{ and} \quad (8)$$

$$M_C = \frac{wL^4}{4s^2} - \frac{wL^3}{3s} + \frac{wL^2}{2} - \frac{6EI}{s} d_{B/C}. \quad (9)$$

Equations (6) - (9) yield the beam end moments and reactions in terms of known geometric and loading variables and an unknown deflection function, $d_{B/C}$. If this function is specified, the bending moments and reactions, and consequently the bending stresses, can be determined.

Specific assumptions can now be made for stiffness values for the elastic springs at A, B and C, representing the elastic support provided to the outer row of fuel elements by the inner fuel-element cluster. These assumptions lead to specified values of the deflection function, $d_{B/C}$, which, through the use of equations (6) - (9), yield particular solutions for the model.

B. Particular Solutions

The deflection function can be obtained for three basic combinations of elastic-spring supports if the assumed symmetry of the structural model in Fig. 6 is to be maintained. These combinations are: 1. The two end supports have a spring constant, k_B , and the central support has a spring constant, k_C . 2. The two end supports have a spring constant, k_B , and the central support is rigid. 3. The two end supports are rigid and the central support has a spring constant, k_C . Particular solutions for these three cases are obtained below:

1. Three Spring Supports

For the general case of spring supports at all three reaction locations on the fuel element, the deflections at the spring locations are given by:

$$\text{deflection at A, B} = \frac{R_B}{k_B} \quad \text{and} \quad (10)$$

$$\text{deflection at C} = \frac{R_C}{k_C}. \quad (11)$$

To simplify the algebra, a substitution is made for the spring stiffnesses as follows:

$$k_B = \frac{EI}{s^3} \text{ and} \quad (12)$$

$$k_C = \frac{mEI}{s^3}. \quad (13)$$

It can be shown readily that a value of p or m equal to 6 corresponds to the deflection under a unit, centrally applied, load on a simply supported fuel element of span $2s$. Thus, a value of p or m equal to 12, for example, implies that the lateral stiffness of the fuel-element cluster is twice that of a single simply supported fuel element of length $2s$.

By use of equations (10) - (13), the deflection function $d_{B/C}$ becomes:

$$d_{B/C} = (\text{deflection at C}) - (\text{deflection at B}),$$

$$\text{or } d_{B/C} = \frac{2R_C s^3}{mEI} - \frac{R_B s^3}{pEI}. \quad (14)$$

Substitution of equation (14) into equations (6) - (9) yields:

$$R_B = \frac{P + \frac{wl^3}{s^2} - \frac{wl^4}{2s^3} + \frac{24}{m} \left[\frac{wl - \frac{wl^3}{s^2} + \frac{wl^4}{2s^3}}{(1 + \frac{24}{m})} \right]}{\left[1 + \frac{12}{p} \left(1 - \frac{24}{1 + \frac{24}{m}} \right) \right]}, \quad (15)$$

$$R_C = \frac{wl - \frac{wl^3}{s^2} + \frac{wl^4}{2s^3} + \frac{12}{p} \left[\frac{P + \frac{wl^3}{s^2} - \frac{wl^4}{2s^3}}{(1 + \frac{12}{p})} \right]}{\left[1 + \frac{24}{m} \left(1 - \frac{12}{1 + \frac{12}{p}} \right) \right]}; \quad (16)$$

$$M_B = \frac{Ps}{2} + \frac{wl^3}{6s} - \frac{\frac{Ps}{2} + \frac{wl^3}{2s} - \frac{wl^4}{4s^2} + \frac{12s}{m} \left[\frac{wl - \frac{wl^3}{s^2} + \frac{wl^4}{2s^3}}{(1 + \frac{24}{m})} \right]}{\left[1 + \frac{12}{p} \left(1 - \frac{24}{1 + \frac{24}{m}} \right) \right]}, \text{ and } (17)$$

$$M_C = \frac{Ps}{2} - \frac{wl^3}{6s} + \frac{wl^2}{2} - \frac{\frac{Ps}{2} + \frac{wl^3}{2s} - \frac{wl^4}{4s^2} + \frac{12s}{m} \left[\frac{wl - \frac{wl^3}{s^2} + \frac{wl^4}{2s^3}}{1 + \frac{24}{m}} \right]}{\left[1 + \frac{12}{p} \left(1 - \frac{24}{1 + \frac{24}{m}} \right) \right]}. (18)$$

2. Two End Spring Supports

The solution for the case in which the middle support is rigid and the two end supports have a spring constant $k_B = \frac{PEI}{s^3}$ is obtained by setting the deflection at the middle support, C, equal to zero in equation (14), whence,

$$d_{B/C} = \frac{R_B s^3}{pEI}. (19)$$

Substitution of equation (19) into equations (6) - (9) yields:

$$R_B = \frac{P + \frac{wl^3}{s^2} - \frac{wl^4}{2s^3}}{1 + \frac{12}{p}}; (20)$$

$$R_C = wL - \frac{wL^3}{s^2} + \frac{wL^4}{2s^3} + 12 \left[\frac{P + wL^3 - \frac{wL^4}{2s^3}}{(p + 12)} \right]; \quad (21)$$

$$M_B = \frac{wL^4}{4s^2} - \frac{wL^3}{3s} + 6s \left[\frac{P + \frac{wL^3}{s^2} - \frac{wL^4}{2s^3}}{(p + 12)} \right]; \text{ and} \quad (22)$$

$$M_C = \frac{wL^4}{4s^2} - \frac{2wL^3}{3s} + \frac{wL^2}{2} + 6s \left[\frac{P + \frac{wL^3}{s^2} - \frac{wL^4}{2s^3}}{(p + 12)} \right]. \quad (23)$$

3. One Central Spring Support

If the two end supports are assumed to be rigid and the central support has a spring constant, $k_C = \frac{mEI}{s^3}$, the solution is obtained by setting the deflections at A and B equal to zero in equation (14), whence,

$$d_{B/C} = \frac{2R_C s^3}{mEI}. \quad (24)$$

Substitution of equation (24) into equations (6) - (9) yields:

$$R_B = P + \frac{wL^3}{s^2} - \frac{wL^4}{2s^3} + 24 \left[\frac{wL - \frac{wL^3}{s^2} + \frac{wL^4}{2s^3}}{(m + 24)} \right]; \quad (25)$$

$$R_C = \frac{wL - \frac{wL^3}{s^2} + \frac{wL^4}{2s^3}}{(1 + \frac{24}{m})}; \quad (26)$$

$$M_B = \frac{wL^4}{4s^2} - \frac{wL^3}{3s} - 12s \left[\frac{wL - \frac{wL^3}{s^2} + \frac{wL^4}{2s^3}}{(m + 24)} \right]; \text{ and} \quad (27)$$

$$M_C = \frac{wL^4}{4s^2} - \frac{2wL^3}{3s} + \frac{wL^2}{2} - 12s \left[\frac{wL - \frac{wL^3}{s^2} + \frac{wL^4}{2s^3}}{(m+24)} \right]. \quad (28)$$

4. Prescribed Deflection

If the relative deflection between the central support at C and the two end supports at A and B is prescribed to be $\pm\delta$, the solution is obtained by setting $d_{B/C} = \pm\delta$ in equation (14). On substitution of this value of $d_{B/C}$ into equations (6) - (9), we obtain:

$$R_B = P + \frac{wL^3}{s^2} - \frac{wL^4}{2s^3} + \frac{12EI(\pm\delta)}{s}; \quad (29)$$

$$R_C = wL - \frac{wL^3}{s^2} + \frac{wL^4}{2s^3} - \frac{12EI(\pm\delta)}{s^3}; \quad (30)$$

$$M_B = \frac{wL^4}{4s^2} - \frac{wL^3}{3s} - \frac{6EI(\pm\delta)}{s^2}; \text{ and} \quad (31)$$

$$M_C = \frac{wL^4}{4s^2} - \frac{2wL^3}{3s} + \frac{wL^2}{2} - \frac{6EI(\pm\delta)}{s^2}. \quad (32)$$

C. Solutions for Specific Boundary Conditions

The expressions for end reactions and bending moments obtained thus far can be simplified further by specifying additional geometric and loading boundary conditions. From Fig. 6, equilibrium requires:

$$\frac{F}{\mu n} = 2P + 2wL, \quad (33)$$

where μ is the coefficient of friction between the hexagonal-can wall and the fuel-element cladding; n is equal to the number of fuel elements in direct contact with the can wall; and F is the total pull force applied on the can.

The lateral loads on the fuel element are assumed to be equal at all three locations, A, B, and C. Thus:

$$P = 2w\ell = L. \quad (34)$$

From equations (33) and (34) we have:

$$L = \frac{F}{3\mu n}. \quad (35)$$

The length, 2ℓ , along which the central portion of the fuel element is assumed to be in direct contact with the wall of the hexagonal can, is equal to one sixth of the spacer-wire pitch. Thus:

$$\ell = \frac{s}{6}. \quad (36)$$

By substituting equations (33) - (36) into the solutions obtained earlier for the four particular cases, final solutions for reactions and bending moments can be obtained in terms of the pull force, F ; the coefficient of friction μ ; the subassembly design expressed through the number of elements, n , in contact with the hexagonal can; the spacer-wire pitch s ; and the stiffness of the inner fuel-element cluster expressed by the spring constants, m and p .

Final solutions are given below for the cases of two end spring supports, one central spring support and a specified deflection.

For the case where the middle support is rigid and the end supports have a spring constant, $k_B = \frac{PEI}{s^3}$, we obtain:

$$R_B = \frac{875}{2592} \frac{F}{\mu n \left(1 + \frac{12}{p}\right)}; \quad (37)$$

$$R_C = \frac{F}{2592\mu n} \left[421 + \frac{10,500}{(p + 12)} \right]; \quad (38)$$

$$M_B = \frac{Fs}{5184\mu n} \left[7 - \frac{10,500}{(p + 12)} \right]; \text{ and} \quad (39)$$

$$M_C = \frac{Fs}{5184\mu n} \left[57 + \frac{10,500}{(p + 12)} \right]. \quad (40)$$

For the case where the two end supports are rigid and the central support has a spring constant, $k_C = \frac{mEI}{s^3}$, we obtain:

$$R_B = \frac{F}{2592\mu n} \left[875 + \frac{10,104}{(m + 24)} \right]; \quad (41)$$

$$R_C = \frac{421}{2592} \frac{F}{\mu n \left(1 + \frac{24}{m} \right)}; \quad (42)$$

$$M_B = \frac{Fs}{5184\mu n} \left[7 + \frac{10,104}{(m + 24)} \right]; \text{ and} \quad (43)$$

$$M_C = \frac{Fs}{5184\mu n} \left[57 - \frac{10,104}{(m + 24)} \right]. \quad (44)$$

When the relative deflection between the central support, C, and the two end supports, A and B, is specified to be $\pm\delta$, we obtain:

$$R_B = \frac{875F}{2592\mu n} + \frac{12EI}{s} (\pm\delta); \quad (45)$$

$$R_C = \frac{421F}{2592\mu n} - \frac{12EI}{s} (\pm\delta); \quad (46)$$

$$M_B = \frac{-7Fs}{5184\mu n} - \frac{6EI}{s} (\pm\delta); \text{ and} \quad (47)$$

$$M_C = \frac{57Fs}{5184\mu n} - \frac{6EI}{s^2} (\pm\delta) . \quad (48)$$

The solution for the case where all three supports are rigid and there is no relative deflection between the supports is obtained simply by setting $\delta = 0$ in equations (45) - (48).

D. Bending Stresses in Cladding

Cladding stresses can be obtained for each of the cases considered above by using the relationship:

$$\sigma = \frac{M}{I} y \quad (49)$$

where: σ is the bending stress in the cladding at a reference axial section; M is the bending moment in the fuel element at the reference section; I is the moment of inertia of the fuel element; and y is the distance between the longitudinal axis of the fuel element and the location in the cladding at which the stress is desired, measured in the plane of bending.

The maximum stress is obtained for the extreme outer fibers of the fuel element, where y is equal to the outer radius of the fuel element, r_o , at the section where the maximum bending moment, $M_{\max}$, occurs. Thus

$$\sigma_{\max} = \frac{M_{\max}}{I} r_o . \quad (50)$$

The deformed shape of the fuel element and its lateral deflections can be obtained from the solution provided and equations (3) and (4).

IV. RESULTS

The solutions presented above yield cladding bending stresses, bending moments, reactions, slopes, and deflections in the outer row of fuel

elements during removal of the hexagonal can as a function of: the pull force, F ; the coefficient of friction μ ; the spacer-wire pitch, $2s$; the number of fuel elements in contact with the can wall, n ; and the elastic support provided by the inner fuel-element cluster, expressed by the spring constants m and p . These solutions are illustrated by application to the experimental subassemblies X062, X059A, and X069A, described in Table I, for a pull force, F , = 2000 lb. These three subassemblies show the effect of variations in spacer-wire pitch and diameter.

The coefficient of friction between bare steel surfaces, obtained from standard references, is 0.45 - 0.75. Because in EBR-II subassemblies sodium deposits are expected to retard sliding between the hexagonal can and the fuel-element cladding, a value of $\mu = 0.75$ is assumed. Furthermore, the fuel is assumed to crack on bending; thus it has no bending stiffness. This assumption leads to the conservative assumption that the fuel-element cross section is represented structurally by the cladding alone. Finally, in the present examples the fuel-element cladding in all three subassemblies is assumed to have an inner diameter of 0.22 in. and an outer diameter of 0.25 in. The influence on the cladding bending stresses of deviations in the cladding dimensions from these values may be obtained from the relationships presented in the previous section.

A. Subassembly X062

Maximum bending stresses and moments of cladding, and reactions and deflections at the support, are given in Table II for subassembly X062 for a pull force, F , of 2000 lb. This force corresponds to a value of $P = w_1 \approx 50$ lb. Results are provided for five combinations of support stiffnesses and fuel-element loading.

Case I in Table II assumes that the middle support is rigid and the end supports have stiffnesses characterized by the constant p . For a spring constant corresponding to $p = 24$, a maximum cladding stress of 49.6 ksi is obtained and the deflection of the fuel element at the end supports is 16.3 mils. For $p = 48$, the maximum cladding stress decreases to 32.9 ksi and the support deflection is 9.8 mils. This result

TABLE I. Descriptions of Subassemblies X062, X059A, and X069A

	Subassembly		
	X062	X059A	X069A
Subassembly Type	F37A	F37A	H37A
Wire Pitch, in.	6	12	12
Wire Dia., in.	0.0625	0.0625	0.040
Wire Material -- Stainless Steel Type	304	304	304
Average Preirradiation Cladding Dia., in.	0.2502	0.2505	0.2508
Cladding Material - Stainless Steel Type	304, 316, 321	304	316
Stainless Steel Swelling, % $\Delta D/D$			
WADCO "Beta" Correlation for 304 SS	1.0	2.4	-
EBR-II Correlation for 304 SS	1.1	2.5	-
WADCO Correlation for 316 SS	0.6	-	1.4
Growth in Fuel-element Dia., % $\Delta D/D$			
Composite Data for 304 SS	1.5	2.5	-
Composite Data for 316 SS	1.0	-	1.5
Estimated Current Midplane Diameter			
Average Fuel Element, in.	0.2534	0.2568	0.2546
Average Spacer Wire, in.	0.0634	0.0641	0.0406
Current Pin-bundle Size across Flats ^a , in.	1.9629	1.9883	1.8291
Inner Hex Can: Size across Flats -- Preirradiation, in. ^a	2.006	2.006	1.846
Can Swelling, in.	0.015	0.006	0.011
Gap Opposite Spacer-wire Contact			
During Disassembly ^b	0.043	0.017	0.017
Total Interference ^c	0.020	0.047	0.024

^a(pin + wire diameter)(6 cos 60° + 1)

^bCurrent pin-bundle size

^cGap opposite spacer wire-contact

TABLE II. Maximum Cladding Stresses and Support Deflections for Subassembly X062

Case No.	Description	Additional Conditions	$k_{A,B}$, lb	$2k_C$, lb	$M_{A,B}$, in.-lb	M_C , in.-lb	σ_{max} , ksi	$\delta_{A,B}$, mils	δ_C , mils
I	$k_A = k_B = \frac{PEI}{s^3}$ $P = wL = 50 \text{ lb}$ $s = 3 \text{ in.}$	(a) $p = 24$ $k_{A,B} = 2.04 \text{ lb/mil}$	33.2	83.4	24.7	30.3	49.6	16.3	0
		(b) $p = 48$ $k_{A,B} = 4.08 \text{ lb/mil}$	40.0	70.0	14.6	20.1	32.9	9.8	0
II	$k_C = \frac{mEI}{s^3}$ $P = wL = 50 \text{ lb}$ $s = 3 \text{ in.}$	(a) $m = 24$ $k_C = 2.04 \text{ lb/mil}$	62.9	24.2	-18.9	-13.3	30.7	0	11.9
		(b) $m = 48$ $k_C = 4.08 \text{ lb/mil}$	58.8	32.4	-12.8	-7.2	20.8	0	7.9
III	$P = wL = 50 \text{ lb}$ $s = 3 \text{ in.}$	(a) $\delta_{A,B} = 15 \text{ mils}$	33.4	80.8	22.2	27.8	45.5	15	0
		(b) $\delta_C = 15 \text{ mils}$	66.6	16.8	-23.4	-17.8	38.0	0	15
		(c) $\delta_{A,B} = \delta_C = 0$	50.6	48.8	-0.6	5.0	8.1	0	0
IV	$P = wL = 0$ $s = 3 \text{ in.}$	(a) $\delta_{A,B} = 15 \text{ mils}$	-16	32	22.8	22.8	37.4	15	0
		(b) $\delta_C = 15 \text{ mils}$	16	-32	-22.8	-22.8	37.4	0	15
V	$k_A = k_B = \frac{PEI}{s^3}$ $P = 75 \text{ lb}, w = 0$ $s = 3 \text{ in.}$	(a) $p = 24$ $k_{A,B} = 2.04 \text{ lb/mil}$	50	50	37.5	37.5	61.2	24.5	0
		(b) $p = 48$ $k_{A,B} = 4.08 \text{ lb/mil}$	60	30	22.5	22.5	36.7	14.6	0

indicates that maximum cladding bending stresses and the corresponding deflections at the elastic supports decrease as the support stiffness increases. This trend is substantiated by Figs. 7 - 10, which show bending moments, reactions, cladding bending stresses, and deflections respectively, as functions of support stiffness. As expected, all these variables monotonically approach the values corresponding to the case of fixed supports [Case III(c) in Table II], as the support stiffness increases.

Case II in Table II provides results for fixed end supports and a central spring support. For a value of spring constant corresponding to $m = 24$, a maximum cladding stress of 30.7 ksi and a support deflection of 11.9 mils are obtained. For this case, as in Case I, the stresses and deflections decrease with increasing support stiffness. For a value of $m = 48$, the maximum cladding stress decreases to 20.8 ksi and support deflection becomes 7.9 mils. Figures 11 - 14 illustrate the monotonic convergence of support bending moments, reactions, cladding bending stresses, and deflections to the three fixed-supports case [Case III(c)], with increasing stiffness of the central support.

Results for rigid end and central supports are provided by Cases III and IV in Table II. In Case III, loads and deflections are specified; in Case IV, the loads are set equal to zero and only deflections are specified.

If the end supports alone are permitted to deflect by a distance, $\delta = 15$ mils (see Table I), corresponding to the difference between the swollen and unswollen portions of the hexagonal can, the resulting maximum bending stress in the cladding is 45.5 ksi. If the central support at C is permitted to deflect by a distance, $\delta = 15$ mils, while the two end supports remain undeflected, the resulting maximum stress is 38.0 ksi. The maximum stresses obtained for Cases III (a) and (b) differ because of the influence of the loads P and w . Case III(c) represents the condition in which all three supports deflect by an equal amount with a rigid-body motion resulting from removal of the hexagonal can. This corresponds to a condition in which $\delta = 0$ and the maximum cladding bending stresses are 8.1 ksi. Values of $\delta = 0$ and $\delta = 15$ mils provide bounds for the solution as a function of deflections; they are discussed in detail in the next section.

Fig. 7. Influence of End-support Stiffness on Fuel-element Bending Moment for Fuel Elements in Subassembly X062 (Case I)

Fig. 8. Influence of End-support Stiffness on Support Reactions for Fuel Elements in Subassembly X062 (Case I)

Fig. 9. Influence of End-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassembly X062 (Case I)

Fig. 10. Influence of End-support Stiffness on Support Deflection for Fuel Elements in Subassembly X062 (Case I)

Fig. 11. Influence of Central-support Stiffness on Bending Moment of Fuel Elements
in Subassembly X062 (Case II)

Fig. 12. Influence of End-support Stiffness on Support Reaction for Fuel Elements in Subassembly X062 (Case II)

Fig. 13. Influence of Central-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassembly X062 (Case II)

Fig. 14. Influence of Central-support Stiffness on Support Deflection for Fuel Elements in Subassembly X062 (Case II)

If the loads are set equal to zero, as in Case IV, and a deflection equal to 15 mils is specified either at the ends or at the central support, a maximum stress of 37.4 ksi is obtained. In this case the maximum stress is the same, regardless of whether the deflection occurs at the ends or at the center; only the directions of the reactions and bending moments are altered. The magnitudes of the reactions, moments, and cladding bending stresses predicted are related linearly to the specified relative deflection, δ .

Case V in Table II provides a solution for the situation in which interference loads exist between the hexagonal can and the spacer wire at only two points separated by the spacer-wire pitch, as shown in Table II. In this case, the central load, w , is equal to zero and P is 75 lb. For a value of spring constant, p , of 24, the maximum cladding bending stress is 61.2 ksi and the support deflection is 24.5 mils. Again, as in the previous cases, the maximum cladding bending stress and the spring-support deflections decrease to 36.7 ksi and 14.6 mils, respectively, with an increase in support stiffness to a value characterized by $p = 48$. The influence of increasing support stiffness on the computed bending moments, reactions, cladding bending stresses, and deflections is shown in Figs. 15 - 18.

Superposition of the two-point loading, considered in Case V, on the fuel-element geometry represented in Case II with a central spring support will result in no bending stresses in the fuel element.

B. Subassembly X059A

Maximum cladding bending stresses, support reactions, bending moments, and deflections for subassembly X059A are given in Table III for a pull force, F , of 2000 lb. As with subassembly X062, these results are provided for five combinations of support stiffness and fuel-element loading. The results in Table III again indicate that the solutions for the cases with spring supports approach the solution for the fixed-supports case as the spring stiffnesses increase.

TABLE III. Maximum Cladding Stresses and Support Deflections for Subassembly X059A

Case No.	Description	Additional Conditions	$k_{A,B}$ lb	$2k_C$ lb	$M_{A,B}$ in.-lb	M_C in.-lb	$\sigma_{\max}$ ksi	$\delta_{A,B}$ mils	δ_C mils
I	$k_A = k_B = \frac{pEI}{s^3}$ $P = wL = 50 \text{ lb}$ $s = 6 \text{ in.}$	(a) $p = 240$ $k_{A,B} = 2.55 \text{ lb/mil}$	48	54	6	17	29	19	0
		(b) $p = 480$ $k_{A,B} = 5.1 \text{ lb/mil}$	49	52	2	13	22	9.6	0
II	$k_C = \frac{mEI}{s^3}$ $P = wL = 50 \text{ lb}$ $s = 6 \text{ in.}$	(a) $m = 240$ $k_C = 2.55 \text{ lb/mil}$	53	44	-7	3	11.5	0	17.3
		(b) $m = 480$ $k_C = 5.1 \text{ lb/mil}$	52	46	-4	6	9.8	0	9
III	$P = wL = 0$ $s = 6 \text{ in.}$	(a) $\delta_{A,B} = 6 \text{ mils}$	49.8	50.4	1.0	12.2	19.8	6	0
		(b) $\delta_C = 6 \text{ mils}$	51.4	47.2	-3.4	7.8	12.6	0	6
		(c) $\delta_{A,B} = \delta_C = 0$	50.6	48.8	-1.2	10.0	16.2	0	0
IV	$P = wL = 0$ $s = 6 \text{ in.}$	(a) $\delta_{A,B} = 6 \text{ mils}$	-0.8	1.6	2.2	2.2	3.6	6.0	0
		(b) $\delta_C = 6 \text{ mils}$	-0.8	1.6	-2.2	-2.2	3.6	0	6.0
V	$k_A = k_B = \frac{pEI}{s^3}$ $P = 75 \text{ lb,}$ $w = 0,$ $s = 6 \text{ in.}$	(a) $p = 240$ $k_{A,B} = 2.55 \text{ lb/mil}$	71	8	12	12	18	27.8	0
		(b) $p = 480$ $k_{A,B} = 5.1 \text{ lb/mil}$	73	4	6	6	9	14.3	0

Where the loading is specified -- Cases I, II, and III(c), and V -- the maximum cladding stresses for subassembly X059A are twice as high as those obtained for subassembly X062 for the same values of p and m . This difference results because bending moments and stresses are related linearly to the spacer-wire pitch, $2s$, which is 12 in. for subassembly X059A; it was 6 in. for subassembly X062. The support reactions are the same in both subassemblies, because the interference loads P and $w\bar{z}$ are assumed to be the same for the corresponding cases. However, the support deflections, for the same values of p and m , will be greater for subassembly X059A by a factor of 8 because the stiffness of the support varies inversely with s^3 . The influence of increasing support stiffness on the bending moments, reactions, cladding bending stress, and support deflections for Class I, II and V is shown in Figs. 19 - 30.

For the cases where deflections and loads or deflections alone are specified -- Cases III(a), III(b) and IV -- the support reactions, bending moments, and cladding bending stresses predicted for the subassembly X059A are lower than those predicted for subassembly X062A for the same specified deflections and loads. Furthermore, this decrease is not related linearly to the spacer-wire pitch as in the cases where loads alone were specified. Examination of equations (45) - (48) reveals that the support reactions are inversely proportional to the cube of half of the spacer-wire pitch (i.e. $\propto \frac{1}{s^3}$). However, the bending moments and the resulting cladding stresses are obtained from the sum of two terms, each of which is a function of the spacer-wire pitch. In the first term, the bending moments and stresses increase linearly with the spacer-wire pitch; in the second, they decrease inversely with the square of half of the spacer-wire pitch. The net result of changing the pitch is a function of the weighting provided by the specified relative deflection, δ , and the loading function, F or P and $w\bar{z}$. The maximum relative deflection, δ , specified for subassembly X059A is 6.0 mils (Table I). Under this specified deflection and loads $P = w\bar{z} = 50$ lb, the maximum bending stress for cladding is 19.8 ksi. If the relative deflection, $\delta = 6.0$ mils, alone is specified, the maximum bending stress is reduced to 3.6 ksi.

Fig. 15. Influence of End-support Stiffness on Bending Moment of Fuel Elements in Subassembly X062 (Case V)

Fig. 16. Influence of End-support Stiffness on Support Reaction for Fuel Elements in Subassembly X062 (Case V)

Fig. 17. Influence of End-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassembly X062 (Case V)

Fig. 18. Influence of End-support Stiffness on Support Deflection of Fuel Elements in Subassembly X062 (Case V)

Fig. 19. Influence of End-support Stiffness on Bending Moment of Fuel Elements in Subassemblies X059A and X069A (Case I)

Fig. 20. Influence of End-support Stiffness on Support Reaction for Fuel Elements in Subassemblies X059A and X069A (Case I)

Fig. 21. Influence of End-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassemblies X059A and X069A (Case I)

Fig. 22. Influence of End-support Stiffness on Support Deflection for Fuel Elements
in Subassemblies X059A and X069A (Case I)

Fig. 23. Influence of Central-support Stiffness on Bending Moment for Fuel Elements in Subassemblies X059A and X069A (Case II)

Fig. 24. Influence of Central-support Stiffness on Support Reaction for Fuel Elements in Subassemblies X059A and X069A (Case II)

Fig. 25. Influence of Central-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassemblies X059A and X069A (Case II)

Fig. 26. Influence of Central-support Stiffness on Support Deflection for Fuel Elements in Subassemblies X059A and X069A (Case II)

Fig. 27. Influence of End-support Stiffness on Bending Moment of Fuel Elements in Subassemblies X059A and X069A (Case V)

Fig. 28. Influence of End-support Stiffness on Support Reaction for Fuel Elements in Subassemblies X059A and X069A (Case V)

Fig. 29. Influence of End-support Stiffness on Bending Stress of Cladding of Fuel Elements in Subassemblies X059A and X069A (Case V)

Fig. 30. Influence of End-support Stiffness on Support Deflection for Fuel Elements in Subassemblies X059A and X069A (Case V)

C. Subassembly X069A

The solutions for subassembly X069A are identical to those obtained for subassembly X059A in Cases I, II, III(c), and V, where only loads are specified. Solutions for Cases III(a), III(b), and IV, where a relative deflection of $\delta = 11$ mils (Table I) is specified, are presented in Table IV. Under this deflection, the maximum bending stress for cladding is 22.7 ksi when loads $P = wL = 50$ lb also are present. If the deflection alone is specified, the maximum stress reduces to 6.5 ksi.

V. BOUNDS ON THE SOLUTION

The solutions presented for subassemblies X062, X059A, and X069A can be bounded by examining the relationships among the pull force exerted on the hexagonal can, the interference force between the can and the outer-row fuel elements, and the relative support deflections and stiffnesses.

A pull force of $F = 2000$ lb on the can has been selected to obtain the results in the previous section. In general, F will be less than the total machine pull force required for disassembly because a part of the machine force may be reacted at contact points in addition to the three assumed in the present model. The boundaries obtained from the current solutions therefore are believed to be conservative.

A. Lower Bounds for Maximum Cladding Bending Stress

A lower bound for the maximum cladding bending stress during disassembly is provided by assuming that the support stiffnesses are very high and that no relative deflection occurs between the various supports. Under these conditions, the maximum cladding bending stress is estimated to be 8.1 ksi in subassembly X062 and 16.2 ksi in subassemblies X059A and X069A, for a pull force of 2000 lb on the hexagonal can.

TABLE IV. Maximum Cladding Stresses and Support Deflections for Subassembly X069A

Case No.	Description	Additional Conditions	$k_{A,B}$ lb	$2k_C$ lb	$M_{A,B}$ in.-lb	M_C in.-lb	σ_{max} ksi	$\delta_{A,B}$ mils	δ_C mils
I ^a									
II ^a
	(a) $\delta_{A,B} = 11$ mils	49.1	51.8	2.8	14.0	22.7	11	0
III	$P = wL = 50$ lb	(b) $\delta_C = 11$ mils	52.1	45.8	-5.2	6.0	9.7	0	11
	$s = 6$ in.	(c) $\delta_{A,B} = \delta_C = 0$	50.6	48.8	-1.2	10.0	16.2	0	0
	(a) $\delta_{A,B} = 11$ mils	-1.5	3.0	4.0	4.0	6.5	11	0
IV	$P = wL = 0$	(b) $\delta_C = 11$ mils	1.5	-3.0	-4.0	-4.0	6.5	0	11
	$s = 6$ in.								

V^a

^a These are the same as in Table III - "Maximum Cladding Stresses and Support Deflections for Subassembly X059A"

B. Upper Bounds for Maximum Cladding Bending Stress

The upper bound for the maximum bending stress for cladding during disassembly has more practical significance than the lower bound. This bound can be obtained by examining simultaneously the normal interference loads on the fuel element and the fuel-element deflection. An estimate of the maximum normal loads is provided by the machine pull force. Simultaneously, the deflections under these maximum loads can be computed and compared with the maximum fuel-element deflections that can occur within the limited space between the fuel-element-cladding and the inner wall of the hexagonal can. An "upper-bound" solution then can be obtained by adjusting the loads or the support stiffnesses, or both, to yield a deflection equal to the specified maximum deflection for the subassembly. Adjustments between the loads and the support stiffnesses are necessary because the latter are unknown. However, the support stiffness can be estimated from the correlations described above.

Determination of the upper bound for the current analytical solution therefore hinges upon the maximum possible deflection (relative between the supports considered in the model) that the fuel element can undergo during disassembly. Clearly, this maximum deflection cannot exceed the differential flat-to-flat dimensions between the swollen and unswollen portions of the hexagonal can, δ_H . If there is no gap between the outer-row fuel elements and the can, this deflection is expected to be distributed between the fuel elements on opposing flats. Furthermore, this specified deflection, δ_H , may not be entirely relative between adjacent supports if they both deflect by some amount. The upper bound obtained by using the differential swelling of the can, δ_H , as the maximum relative support deflection is, therefore, expected to be conservative.

1. Subassembly X062

For subassembly X062, the differential hexagonal-can swelling, δ_H , is 15 mils (Table I). If a relative deflection of 15 mils is imposed upon an outer-row fuel element, the following results are obtained:

a. Case I. If the interference conditions are represented by Case I, a 15-mil deflection is obtained at the end supports, A and B, for a value of $p = 28$, which corresponds to a spring stiffness, $k_{A,B} = 2.88$ lb/mil (Fig. 10). From Fig. 9, the corresponding maximum cladding bending stress is 45 ksi.

b. Case II. For this case, a 15-mil deflection is obtained at the central support, C, for a value of $m = 13$, which corresponds to a spring stiffness, $k_c = 1.1$ lb/mil (Fig. 14). The corresponding maximum cladding bending stress, from Fig. 13, is 39 ksi.

c. Case V. For the two-point loading considered in Case V, a 15-mil deflection is obtained at the two end supports, A and B, for a value of $p = 47$, which corresponds to a spring stiffness, $k_{A,B} = 4.07$ lb/mil (Fig. 18). This support stiffness yields a maximum cladding bending stress of 38 ksi from Fig. 17.

The upper boundary for the maximum cladding bending stress obtained from the three cases described above is 45 ksi, corresponding to Case I. The "true" upper boundary is obtainable only if the "real" loading distribution and the relative support stiffnesses are known. Because any one of the three cases could occur and the results obtained from these cases are within a range of about 10%, the question of determining the real case is not belabored.

2. Subassembly X059A

The differential swelling of hexagonal cans in subassembly X059A is 6 mils (Table I). If a relative deflection of 6 mils is imposed on an outer-row fuel element, the results shown in Table V are obtained in the manner described above.

3. Subassembly X069A

As discussed earlier, the solutions for subassemblies X059A and X069A are identical except for the differential swelling of the hexagonal

TABLE V. Results for Cases I, II and IV for a Relative Deflection of 6 mils on an Outer-row Fuel Element for Subassembly X059A

Case No.	Maximum Cladding Bending Stress, ksi	P	M	$k_{A,B}$, lb/mil	k_C , lb/mil	Figures
I	20	700		7.4		22 and 22
II	10		600		6.4	23 and 24
V	8	600		6.4		29 and 30

can, which, for subassembly X069A, is 11 mils (Table I). Thus, by using this value of relative deflection in conjunction with Figs. 19 - 30 as shown above, the upper boundary for the maximum cladding bending stress in subassembly X069A is 28 ksi, corresponding to Case I.

VI. DISCUSSION

The results that have been described provided estimates of fuel-element reactions, deflections, and bending moments; and maximum cladding bending stresses for several assumed conditions of disassembly interference with disassembly of subassemblies. The five cases considered are believed to cover the range of conditions expected for the outer-row fuel elements during disassembly. Other cases can be assumed and analyzed by using the relationships presented in Section III.

The present analytical model shows that the critical parameters that define the disassembly-loading conditions of the outer-row fuel elements are: (a) the normal reaction between the hexagonal can and the fuel element; and (b) the stiffnesses of the inner fuel-element cluster.

Figure 6 indicates that at least three contact locations exist between the hexagonal can and an outer-row fuel element during disassembly. Additional contact points may exist without significant interference. However, if a large gap exists between the outer-row fuel element and the can wall, initially, most of the interference loads might be taken up by the spacer wires at two locations separated by the spacer-wire pitch as in Case V.

The effective lateral stiffness of the inner fuel-element cluster is an unknown variable in the present analytical model. However, if a relative deflection is specified for an outer-row fuel element and the pull force necessary to remove the hexagonal can is known, a value for the spring-support stiffness, representing the inner fuel-element cluster, can be obtained. For the values of deflections and pull force specified in the analysis, support stiffnesses appear to be 1-8 lb/mil.

The lateral stiffness of the inner fuel-element cluster, along its length, can be estimated directly by a simple experiment. A prototypal

fuel subassembly can be loaded laterally at several locations along its length. The outer row of fuel elements can be loaded directly by a flat rectangular plate inserted into the hexagonal can through a slit. The deflections corresponding to known loads can be measured by a dial gauge and the resulting stiffness obtained. Swelling of the fuel bundle after irradiation and the resulting interference between the fuel elements and the hexagonal can can be simulated by shimming the prototypal fuel subassembly. Thus, varying degrees of interference can be simulated by adjusting the degree of shimming.

A very high rotational stiffness of the outer fuel elements at the two end-contact locations, A and B, in Fig. 6 was assumed. This assumption was used in setting the slope of the fuel element equal to zero at these two locations. Experimental determination of the rotational stiffness is expected to be very complex; the present assumption appears to be acceptable for the current investigation.

Some of the assumptions made in the present model must be kept in mind. In particular, the axial force in the fuel element was assumed to have no influence on the bending moments and stresses in the fuel-element cladding. If it is assumed that no axial load is transferred from the outer rows of fuel elements to the inner fuel-element cluster, then the axial load in each outer fuel element will be F/n lb at the lower end of the fuel element, because the load is transferred progressively at succeeding reaction locations as shown in Fig. 6. The axial stress at the lower ends of the outer row of fuel elements in subassembly X062 will be about 10 ksi. If some axial load is transferred to adjacent inner fuel elements, the axial stress will decrease. This stress will not damage the fuel-element cladding but may cause some local damage to the spade or the T-bar grid.

Contact stresses resulting from the application of a lateral load on the spacer wire of a fuel element have not been analyzed in the present study. These stresses are not expected to be significant. In the core midplane region, the swelling of fuel and cladding is expected to result in higher interference loads. The fuel, however, can support compressive loads, so that local buckling of the cladding will be precluded. In the plenum region, where the beneficial action of the fuel is absent,

interference between the fuel element and the hexagonal can, and consequently the lateral load on the cladding, is expected to be minimal, if not absent.

VII. CONCLUSIONS

The simple structural-analysis model selected shows that the primary parameters that influence interference-induced bending stresses in the fuel-element cladding during disassembly are: the pull force on the hexagonal can; the coefficient of friction between the can wall and the fuel-element cladding; the stiffness of the fuel-element cluster; and the spacer-wire pitch.

The fuel-element and spacer-wire diameters are secondary parameters in the present model insofar as they influence the stiffness of the fuel-element cluster.

An upper bound for the cladding bending stresses is determined in the model by the differential flat-to-flat dimensions between the swollen and unswollen portions of the hexagonal can. This dimension is a function of the subassembly design and irradiation history.

The results demonstrate that the cladding bending stresses decrease with increasing spacer-wire pitch and decreasing lateral stiffness of the fuel-element cluster.

Improved analytical predictions may be obtained if the lateral stiffness of the inner fuel-element cluster is known. This stiffness may be estimated by performing some simple tests and by recording machine pull forces during disassembly of irradiated subassemblies.

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