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EVALUATION OF COMPLIANT GRID-TYPE

FUEL ROD SPACERS

FOR

FTR FUEL ASSEMBLY

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CHAPTER 1

SUMMARY

The objective of this study was to develop a conceptual design for a compliant grid type fuel pin spacer for use in the FFTF fuel assembly. Three basic grid designs were considered:

1. Tri planar tubular grid
2. Honeycomb grid
3. Clustered rod grid

All three grid designs incorporated a new type of spring finger design using a torsion bar as the principal spring element.

The honeycomb grid was selected as having the best mechanical and thermal hydraulic characteristics.

This report presents the mechanical and thermal fluidic analysis leading to the selection of the honeycomb grid. The performance of a fuel assembly incorporating these grids is assessed. Development programs are given detailing the principal design and development activities required to produce a successful fuel assembly, incorporating compliant grid pin spacers.

No consideration of the effect of irradiation induced material swelling has been included in the design analysis of the actual grid structure. The data required to investigate this effect are not currently available.

CHAPTER 2

INTRODUCTION

The objective of this task is to produce a conceptual design for a compliant grid fuel rod spacer, as an alternate to the reference wire wrap design in the FFTF fuel subassembly. In practice it has been found that the design of the grid and the design of the fuel assembly cannot be separated. The interaction between the two is such that a meaningful design is obtained only when the grid is designed as a component of a specific fuel subassembly. The scope of this task has therefore, been broadened to include certain necessary considerations of the fuel subassembly design.

Pressure drop, flow distributions and coolant temperature distributions within the fuel assembly were evaluated. In addition, localized temperature variations around a rod in the vicinity of a grid were considered in the thermal design. Effects and interactions between thermal bow of fuel rods and coolant were analyzed.

Service conditions for the FFTF fuel subassembly are such that the design analysis must include the effects of gross plastic straining and creep of the fuel rods. Time dependent deformation of the subassembly support structure must be considered. The design analysis presented includes a conventional elastic analysis of the loading and deformation of the grid and fuel assembly structure. In addition a number of nonlinear phenomena with the greatest potential for limiting the design have been selected for additional study/analysis. These include:

1. Duct stress relaxation
2. Clad strain concentrations
3. Geometric effects of fuel rod swelling
4. Fuel rod creep buckling
5. Flow induced rod vibration

2.1 DUCT STRESS RELAXATION

Constraint stresses will exist in the flow duct under normal operating conditions. At FFTF operating temperatures these stresses will relax; with both the temperature gradients and the constraints removed a previously straight duct will be deformed at the end of one residence period. Repositioning of this duct within the reactor now presents a problem, since both the subject duct and the cavity defined by the surrounding ducts will be deformed. This deformation could well prevent duct removal and replacement on other than a complete core basis. The removal and replacement of ducts is necessary only if the fuel assembly is made integral with the duct. This is the case with the current reference wire wrap fuel assembly design. Two fuel subassembly designs incorporating compliant grid type pin spacers have been prepared. Of these two designs one is directly comparable with the reference design in that the grids attach directly to the duct. The second design incorporates the grids in a separate canned fuel subassembly. This fuel subassembly is housed within the duct and is separable from it. Due to its low length and minimal constraint stresses, deformation of this fuel subassembly due to stress relaxation will be negligible. It is felt that refueling problems will be eased considerably by the use of minimum length fuel subassemblies. No analysis of the duct deformation has been included since the design of the core constraint structure has yet to be finalized.

2.2 CLAD STRAIN CONCENTRATIONS

Fuel swelling induced strains of up to three percent have been assumed for the FFTF fuel rod cladding. These strains are applied at a very low rate and at temperatures well within the creep range for the material. The resulting stress-strain curve for a material strained under these conditions exhibits very little positive slope once an equilibrium plateau stress level has been reached. This result is produced by an integration of the effects of conventional straining and stress relaxation over the test specimen lifetime. The significance of this type of curve lies in its potential for producing large strain concentrations due to relatively minor changes in clad geometry/loading. Available irradiation data for austenitic stainless steels indicate that the

fracture elongation is low; values similar to the reference swelling strains have been reported. Any departure from uniform straining can thus directly limit the fuel rod service life. Shear webs in a clustered rod design, and hot spots beneath the wire in a wire wrap design provide the discontinuities necessary to produce strain concentrations. This problem is analyzed in Section 4.6.

2.3 GEOMETRIC EFFECTS OF FUEL ROD SWELLING

Fuel rods are supported at grid locations by four rigid dimples and a compliant spring. With this system any increase in rod diameter produces a movement of the rod center line. Rod swelling thus produces both a loss of flow channel area and a distortion of the channel cross section. This problem is common to all rod support systems. The compliant grid rod spacer, however, has an advantage in that the flow channel distortion is uniform over the subassembly cross section with the exception of the outermost rows of rods. Coolant flow axial distributions are thus minimized. The swelling induced geometric changes are summarized in Sections 3.2 to 3.4.

Swelling induced geometric changes influence the design of the grid spring finger. At three percent, swelling strain will produce a diametral change of 0.0075 inch, and a motion of the spring contact point of up to 0.011 inch. The spring must be designed such that the design stresses are not exceeded in the overtravel condition.

2.4 FUEL ROD CREEP BUCKLING

For the major portion of its life the fuel rod will operate with swelling induced hoop strains well into the plastic range. Under these conditions a major reduction occurs in the end load required to produce strut buckling. A further reduction occurs due to the effect of creep at core operating temperatures. Strut end loads are produced by the frictional resistance to fuel rod sliding at the grids. The loading is strain controlled in that buckling produces a drop in the applied loading; stable operation in the buckled condition is thus possible. The lateral deflections which buckling produces are the principal reason for the interest in this problem, they

contribute a major portion of the coolant flow channel closing. An analysis of the creep buckling problem is given in Section 4.4.

2.5 FLOW INDUCED ROD VIBRATION

A number of design tools are available for the prediction of flow induced vibration amplitudes in elastic rods. These tools must be modified to account for the effect on rod vibration response of plastic hoop strains and compressive strut loading. Both serve to decrease the rod natural frequency and increase the lateral vibration amplitude. The rod vibration analysis is given in Section 4.3.

CHAPTER 3
DESIGN DESCRIPTION

3.1 DESIGN CRITERIA

3.1.1 Design Stress Limits

The following types of stress have been considered:

1. Primary stresses
2. Secondary stresses
3. Peak stresses
4. Buckling stresses

3.1.1.1 Primary Stresses

The primary membrane and bending stresses in the grid shall not exceed the design stress which is the lesser of:

$$\sigma_u/3 = 1/3 \text{ ultimate stress at } 1100^{\circ}\text{F}$$

$$2/3 \sigma_y = 2/3 \text{ yield stress at } 1100^{\circ}\text{F}$$

$$\sigma_c = \text{creep stress for a creep rate of } 0.0001\%/ \text{hr at } 1100^{\circ}\text{F}$$

The use of the same design stress level for both primary membrane and primary bending stress leads to a conservative design. When test data on the grids becomes available it is anticipated that the allowable primary bending stress will be increased.

3.1.1.2 Secondary Stress

No increase of secondary stresses beyond the yield limit has been allowed in order to avoid permanent deformations of the grid. All secondary stresses which are produced by temperature differences have been treated as peak stresses.

3.1.1.3 Peak Stresses

Peak stresses resulting from stress concentrations and temperature differences have been subject to a low cycle fatigue analysis. The cumulative damage usage factor* during the lifetime of the grid in the reactor must not exceed 0.8. The grid has to withstand 30 zero-to-full-power cycles.

The fatigue design curves used must incorporate factors allowing for the effects of the reactor environment. In addition, safety factors of 1.5 on stress amplitude or 10 on cycles-to-failure are incorporated.

3.1.1.4 Buckling Stresses

Stresses/loads in the grid or any of its structural parts must not exceed the buckling stress/load.

The maximum stress in the springs which hold the fuel rods must not exceed the yield stress σ_y during the lifetime of the grid in the reactor. At the beginning of life the preload in the spring should produce a stress not higher than 2/3 of the yield stress.

3.1.2 Selection of Grid Materials

Based on a consideration (Section 6.1) of the following factors:

1. Mechanical properties
2. Corrosion/erosion in flowing sodium
3. Fabrication

Inconel-718 has been selected as the reference grid material. Inconel X-750 is designated as an alternate. The brazing alloy selected is Nicrobraz-50, with coast metals Nicrobraz-52, Nicrobraz-53, and Nicrobraz-NP as possible alternates.

* Cumulative damage usage factor = $\sum_{\text{life}} \frac{\text{number of cycles applied}}{\text{number of cycles to failure}}$

3.2 TRI PLANAR GRID

The grid consists of tubular elements arranged in three planes, such that one-third of the elements are at a lower plane, one-third are at an intermediate plane, and the remaining one-third are at an upper plane. The planes are spaced either side of the intermediate plane at approximately one half the element axial length (Figures 3-1 and 3-2). The elements are held together as a single rigid structure by brazing. Each element has its own spring finger and dimples formed out of the parent material.

The purpose of the staggered pitch of the elements is to reduce the pressure losses across the grid structure. As the coolant enters the grid, one third of the grid element material cross section restricts the flow passage; a further increase in flow passage restriction, amounting to two-thirds of the grid element material cross section, occurs over the center longitudinal section of the grid. This is then followed by a decrease in the restriction to coolant flow by having the final section of the flow passage restricted by one-third of the grid element material cross section.

The elements are cut from seamless tubular material having similar dimensional tolerance values as attained for fuel tubes. Dimples are formed by dies, and the spring finger may be formed by punching or an Electro-Discharge Machine. The elements, together with a hexagonal peripheral strap, are held in a fixture for brazing.

3.3 CLUSTERED ROD GRID

The grid consists of formed straps attached together to provide "cells" for restraining the clustered rod configuration (Figure 3-3). The strap form has a compound corrugated section, the top and bottom faces of the envelope being machined to approximately one half of the strap material thickness.

Spring fingers and dimples are formed out of the parent grid material. The straps are assembled and brazed with the machined surfaces in contact.

There is only a single wall thickness of strap material between each four-rod cluster.

The overall hexagonal shape of the grid is obtained by integrating three rhombic sections. At the boundary of each adjacent section the adjacent fuel rods are on a square pitch, with all other rods on a triangular pitch. An exception is the three center rods which form a triangular pitch of a greater dimension than the nominal value. Obviously this does not give a constant relationship between rod periphery and inter-rod coolant channel area across the overall grid. This problem is further aggravated by the shear webs which make up the rod cluster and which occur at each grid location.

Due to the flexural stiffness of the four rod cluster being greater than that for a single rod, fewer grids are required for each fuel assembly. Coupled with this is the more open design of grid which assists in reducing the coolant pressure drop over a fuel assembly.

The comments given in Section 3.4 concerning grid manufacture apply also to this design.

Because of the unknown torsional fixation produced by the spring fingers and dimples within a grid cell on the clustered rod assembly, it was considered desirable to increase, by a small amount, the clearance between the fuel rods and the grid straps for the P/D ratio of 1.16. This was done by reducing the P/D ratio for the clustered rod assemblies to 1.12 while maintaining the pitch of adjacent assemblies. This gave a P/D ratio of 1.20 between adjacent rods of adjacent assemblies.

Reference Drawings: Figures 3-4 through 3-6.

3.4 HONEYCOMB GRID

The grid consists of formed straps attached together to provide hexagonal "cells" for restraining the fuel rods. The strap form has a continuous corrugated channel section. The top and bottom faces of the strap envelope are machined to approximately one half of the strap material thickness. The straps are then assembled and brazed with machined surfaces in contact, forming the hexagonal cells. If required, peripheral straps can be attached to the grid.

With this type of grid, there is only a single wall thickness of strap material between any two adjacent rods after the straps are brazed into a grid assembly. The grid requires a peripheral boundary member (a strap or can) to provide bending stiffness in a direction normal to the grid plane (i.e., bending stiffness in the flow direction).

The strap corrugations can be formed either by progressive stamping of individual straps or form-rolling sheet material from which straps are then cut. The dimples and spring fingers are formed by stamping and machining. Machining the top and bottom faces of the formed strap to one half metal thickness is critical, and must be done in a fixture that holds the formed strap rigid during machining. The straps are in a fixture during the brazing operation.

The face type of brazed joint used in this design is superior to that of edge-brazing intersecting straps.

Reference Drawings: Figures 3-7 through 3-12.

3.5 SPRING FINGER DESIGN

Beam-type grid spring fingers have been the subject of a considerable design and development effort within the Westinghouse PWR Division. The operating limitations of this type of spring are now well established. A minimum operating height of approximately 0.05 inch is required to produce a spring with an acceptably low spring rate. If the height is reduced significantly below this value, the constraint effect at the ends of the spring becomes the dominant factor influencing the spring rate. Since the total rod to rod spacing available in a typical FFTF fuel rod array is only of the order of 0.04 inch, a PWR type spring design is obviously unworkable.

An alternative grid spring finger design, incorporating a torsion bar as the principal spring element, has been developed as a part of the Westinghouse LMFBR program. The torsion bar, together with its cantilever loading arm, is an integral part of the basic grid strap. With the spring in its nominal operating position, both the torsion bar and its loading arm lie within the boundary planes of the parent grid strap material. Coolant flow maldistributions

at the spring locations are thus reduced to a minimum. Maximum stresses occur in the torsion bar. These are approximately constant along the length of the torsion bar; hence the problem of limiting the spring load carrying capacity due to local stress peaking is minimized. A low spring rate together with an acceptable load carrying capacity are the principal characteristics of this spring design.

3.6 EFFECTS OF FUEL SWELLING

For both the triplanar tubular and honeycomb grids, the minimum gap between fuel rod and grid strap is reduced by an average of 0.0031 inches per one percent of fuel rod swelling. For the four-rod cluster grid, this minimum gap is reduced by an average of 0.0033 inches per one percent of rod swelling.

The fuel swelling effect also reduces the flow channel area available to the coolant. The flow area reduction between grids varies from as much as 10 percent with a P/D of 1.12 and 3 percent swelling, to 2-1/2 percent with a P/D of 1.20 and 1 percent swelling.

The effect on flow conditions at each grid location has been analyzed in Chapter 5.

3.7 MATERIAL VOLUME FRACTIONS

Material volume fractions are given in Table 3-1.

All values are based on materials contained within the outer periphery of the duct and over the active core length. No account has been taken of the conical core geometry with the consequent variation of duct pitch along the duct length.

The analysis was carried out for two types of fuel assembly arrangement within the duct. One arrangement considered the fuel assembly as an integral part of the duct; the other considered the fuel assembly contained within as an integral part of an Inconel 718 can, which is removable as an assembly from within the duct.

3.8 FABRICATION OF FFTF SUBASSEMBLIES

The salient steps in the fabrication of all three types of subassembly (tripplanar grid, honeycomb grid and clustered rod) can be listed as follows:

1. Fabricate fuel pins
2. Form and braze grids
3. Form and weld cans
4. Weld grids into the cans
5. Load fuel pins into the grids

3.8.1 Fabricate Fuel Pins

The fabrication of the fuel pins for the triplanar and honeycomb type of assembly would be based on extensive W experience in manufacturing single fuel pins. In the case of the clustered rod type of assembly W experience in electron beam welding of control rod guides* could also be applied but a rigorous fabrication development and qualification program would be mandatory. Fixturing of the rods and shear webs would be critical and difficult. Furthermore, assuming six grids in every subassembly, each four rod cluster would incorporate 24 shear web to cladding welds and eight end cap welds, all of which must be of very high quality. Since only one irreparable weld would cause rejection of the cluster, it can be readily seen that the fabricator must be prepared to either fund the development of extremely reliable and sophisticated welding techniques or accept a high rejection rate. Similar reasoning can be applied to other fabrication criteria concerning the fuel loading and fuel to clad sodium bonding.

3.8.2 Form and Braze Grids

The forming of the grids is estimated to require a roughly equivalent effort for all three types. The factors to be considered in the selection of a brazing allow would also be common to all three types: sodium corrosion resistance, effects of brazing thermal cycle on grid mechanical properties,

* Foster, E. S., "Fabrication of Tubular Type Control Rod by Electron Beam Welding," WCAP-2352, March 1963.

nuclear considerations of composition and qualification of brazing parameters would have equal importance and would require similar development programs.

From the standpoint of quality assurance the honeycomb and clustered rod grids have an advantage over the triplanar type of grid, in that the flat parallel-face joints would be comparatively easy to inspect. Ultrasonic inspection techniques are currently being developed for this application by WARD. In summary, the fabrication development, and by inference the fabrication procedures, for all three types of grid would be as follows:

1. Select and qualify a suitable brazing alloy.
2. Design and manufacture punches, dies and fixtures for forming and brazing grid straps.
3. Optimize the following brazing parameters; joint gap, amount of brazing alloy, method of applying brazing alloy, brazing temperature, brazing duration, brazing atmosphere and homogenization treatment.
4. Optimize age-hardening treatments.
5. Develop suitable inspection criteria and techniques for each type of grid.

3.8.3 Form and Weld Cans

Since the can design is similar for all three types of subassembly, the required fabrication effort for each type can be summarized as follows:

1. Design and manufacture punches, dies and welding fixtures.
 2. Quality tungsten inert gas (TIG) welding procedures for all can joints.
 3. Specify inspection criteria and techniques for each type of joint.
- The fabrication of these cans is expected to be straightforward and no extensive development programs should be required.

3.8.4 Weld Grids into Cans

TIG welding procedures for joining all those types of grid into the cans should require only routine qualification provided that close attention is paid to fixturing and detailed joint design.

3.8.5 Load Fuel Pins into Grids

Since the loading of fuel pins into the subassembly is essentially a fixturing problem, the only concern to the fabricator appears to be that of possible damage to grids and therefore require repair techniques to be developed.

TABLE 3-1
MATERIAL VOLUME FRACTION PERCENT (AVERAGE VALUES)

	<u>P/D</u>	<u>Inconel 718 Can</u>	<u>Fuel</u>	<u>304 SS</u>	<u>Inconel 718</u>	<u>Sodium</u>	<u>Fuel/Cladding Gap</u>
PNL Reference Design	1.12	No	43.29	28.13	----	26.10	2.48
Triplanar	1.12	Yes	42.06	26.67	6.32	22.54	2.41
Tubular	1.12	No	42.87	27.06	1.86	25.75	2.46
Grid	1.16	Yes	39.47	25.42	6.14	26.70	2.27
	1.16	No	40.21	25.78	1.81	29.89	2.31
	1.20	Yes	37.13	24.29	5.98	30.47	2.13
	1.20	No	37.80	24.62	1.76	33.65	2.17
Honeycomb	1.12	Yes	41.08	26.20	6.88	23.48	2.36
Grid	1.12	No	41.22	26.27	1.22	28.92	2.37
	1.16	Yes	38.53	24.97	6.80	27.49	2.21
	1.16	No	38.68	25.04	1.19	32.87	2.22
	1.20	Yes	36.23	23.85	6.72	31.12	2.08
	1.20	No	36.36	23.91	1.16	36.48	2.09
4-Rod	1.12		-----	-----	----	-----	----
Cluster	1.16	Yes	38.41	25.65	5.26	28.48	2.20
Grid	1.20	Yes	36.16	24.53	5.12	32.11	2.08
	1.20	No	37.19	25.05	0.58	35.04	2.14

LTR	GR 1 112	GR 2 116	GR 3 120
A	280	290	300
B	4122	4270	4417
C	4202	4350	4497
E	4242	4390	4537
F	4760	4930	5100
G	4852	5023	5193
H	5245	5416	5585
J	001	006	011
K	280 OD	290 OD	300 OD
L	1932	2000	2067
M	574	592	610
N	1443	1493	1542

NOTE
DIM'S GIVEN ARE AT GRID OPERATING
TEMPERATURES

Figure 3-1. FFTF Fuel Pin Spacer
Concept Number 1
(766J211)

SECTION A-A
SCALE 1 1

P/D	112	116	120
M	4 300	4 451	4 601
N	4 340	4 491	4 641

SECTION B-B
SCALE 1 1

SECTION C C
SCALE 1 1

Figure 3-2. FFTF Fuel Assembly
Installation (766J224)

INTERNAL SPRING STRAP
TYP

SCALE 1:1

PERIPHERAL STRAP

INTERNAL DIVIDING STRAP

LTR	GR 1 PID-116	GR 2 PID-120
A	280	300
B	.300	.300
C	4.107	4.248
E	4.147	4.287
F	5.135	5.287
K	.053	.053
L	.2825	.2925
M	.1375	.1425
N	.1847	.1906
R	.1588	.1645
S	.3305	.3420
T	.812	.848
U	.1153	.1188
V	2.135	2.927
W	.1177	.1168
Y	.580	.600
Z	.457	.475
AA	2.034	2.104

ALL MATERIAL TO BE INCONEL 718
ALL DIMENSIONS NOMINAL (GIVEN AT GRID OPERATING TEMP)

Figure 3-3. FTF Fuel Pin Spacer
Concept Number 3
(766J213)

INTERNAL SPRING STRAP
TYP

SCALE 1:1

INTERNAL DIVIDING STRAP

DIM	GR 1 P/D=1.16	GR 2 P/D=1.20
A	280	300
B	300	300
C	4 174	4 312
E	4 214	4 352
F	5 212	5 372
K	053	053
L	2825	2925
M	1375	1425
N	1847	1906
R	1588	1645
S	3305	3420
T	812	848
U	4 094	4 232
V	2 829	2 927

ALL MATERIAL TO BE INCONEL 718

DIMS GIVEN AT GRID OPER TEMP

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Figure 3-4. FTF Fuel Pin Spacer
Concept Number 3
(766J228)

SECTION A-A
SCALE 4:1

SCALE 1:1

DIM.	GR 1 P/D 116	GR 2 P/D 120
A	290	300

SCALE 2:1

Figure 3-5. FFTF Fuel Assembly Concept
Number 3 (766J223)

SECTION A A
SCALE 1 1

DIM	GR 1 P/D 1.16	GR 2 P/D 1.20
A	4 214	4 352

SECTION B-B
SCALE 1 1

SECTION C C
SCALE 1 1

Figure 3-6. FFTF Fuel Assembly
Installation Concept
Number 3 (766J226)

Figure 3-9. FFTF Fuel Assembly (766J247)

P/D	112	116	120
M	.280	.290	.300
N	.109	.112	.115
L	2.231	2.318	2.405
P	.280	.290	.300

SCALE 1:1

SECTION A-A
SCALE 10:1
GROUP NO. 2 ONLY

SECTION B-B
SCALE 10:1
GROUP NO. 2 ONLY

DIM	GR 1 P/D 1.12	GR 2 P/D 1.16	GR 3 P/D 1.2
A	.280	.290	.300
B	4.212	4.362	4.512
C	4.300	4.451	4.601
E	4.760	4.930	5.100
F	.002	.007	.012
G	.3060	.3175	.3290
H	.1325	.1375	.1425
J	.1790	.1848	.1906
K	.1530	.1588	.1645
L	4.1858	4.3359	4.4858

NOTE:

1- ALL DIMENSIONS NOMINAL & GIVEN AT GRID OPERATING TEMPERATURE

2- FOR TOLERANCE DIMENSIONS SEE DWG. NO. 766J216

Figure 3-11. FFTF Fuel Pin Spacers
(766J248)

Figure 3-12. FFTF Fuel Pin Spacers,
Back-up (766J249)

CHAPTER 4

DESIGN ANALYSIS

4.1 GRID STRUCTURE STRESS ANALYSIS

4.1.1 General Approach

Appendix E contains the details of the grid structure analysis. The following is a short outline of the assumptions made and the restrictions of the analysis.

4.1.1.1 Grid Loads

The purpose of the grid spacer is to maintain the rod pitch approximately constant. The grid structure has to be adequately stiff to prevent lateral movement of the fuel rods. Analysis of the fuel rod thermal bow showed that the fuel rods must be clamped in the grids. Clamping of the fuel rods in the grids allows a larger axial distance between the grids, which reduces the pressure drop through the core.

The forces necessary to keep the fuel rods uniformly spaced are small compared with other forces which the grid has to sustain. Forces in the vertical direction on the grid plane are produced by:

1. Fluid drag
2. Friction of the fuel pins in the grid

The pressure induced load on the grid exists whenever the pumps are operating. A friction load on the grid exists only if the fuel pins expand or contract relative to the grid and can/duct. It can be shown that only a small temperature change is sufficient to produce the maximum friction load. If the temperature distribution is constant structural noise and fluid induced vibration will relax the friction load. In addition, the friction load reverses its direction during a temperature cycle.

Consideration of the true load history would require an extensive stress analysis, which is beyond the scope of this study. It has been assumed that the pressure load and a static friction load act in the same direction. This gives the maximum loading and hence gives a conservative result.

The friction load is produced by the nominal spring force at the beginning of life. Due to fuel swelling the spring force may increase during the life in the reactor. On the other hand the spring load will be continuously reduced due to stress relaxation of the spring material. These two effects should be investigated to find their net effect on the resultant spring. This investigation has not been included in the present study.

Thermal stresses in the grid have been calculated with a simplified symmetrical temperature distribution across the grid (Figure E-29). The temperature is assumed to go through 30 cycles during the lifetime of the reactor. Details of the grid analysis can be found in Appendix E.

4.1.1.2 Grid Structural Model

The objective of the grid analysis was to determine the stresses in the structural elements. Simple, straightforward methods have been used. A further analysis using a finite element computer code with provision for including nonlinear behavior would be a logical extension of the work. Further calculations could possibly allow a small reduction of the grid height. The presented calculation proves the feasibility of the grid concept and gives an approximation of the final design.

In the analysis, the stresses in the grid structure have been calculated using a homogenous plate analysis with reduced bending and disk stiffness.^[7,8,9] The hexagonal plate has been replaced by a circular plate with an equivalent radius. The circular plate is loaded by the same load per unit area as the hexagonal grid.

The grid structure, Figures E-22 and G-2, supporting the clustered fuel rods required special consideration. The straight radial webs are stiffer than the wrinkled straps. A major part of the traverse load on the wrinkled straps is transferred to the radial webs.

4.1.1.3 Material Data

The data for the grid material Inconel 718 are listed in Appendix E paragraph 2.3. A major effort in the investigation of the grid has been the construction of a low cycle fatigue curve for Inconel 718.

Two methods have been used to construct a low cycle fatigue design curve which incorporates the effects of irradiation, of a hot sodium environment and of creep and stress relaxation at elevated temperatures.

The first method starts with an experimental fatigue curve for Inconel 718, which can be described by a function

$$S = \frac{E}{4 N^k} + S_e \quad (4-1)$$

S = Fatigue strength

E = Young's modulus at given temperature

N = Cycles to failure

$$k = \ln \frac{100}{100 - RA} \approx \text{ductility}$$

RA = Percent reduction of area

S_e = Endurance limit

The factors which reduce the low cycle fatigue strength are included in successive steps. Details of the procedure can be found in the Appendix E Section 2.4.

Irradiation affects the fatigue strength through a reduction of the ductility. In the above expression for the fatigue strength S , the ductility is decreased by the same ratio as the ductility of a tensile test specimen is reduced due to irradiation.

Exposure of Inconel 718 to hot sodium reduces the rupture strength which will be taken account of in a similar manner, see Appendix E, Section 2.4.

Figure E-6 curve III shows the low cycle fatigue curve constructed using the first method.

With the second method, which has been suggested by Halford and Manson,^[12] a low cycle fatigue curve is constructed using material data from static tensile tests. A low cycle fatigue curve for Inconel 718 has been constructed

which includes all the damaging influences of a reactor environment using static tensile test data.

The test specimens have been exposed to irradiation in sodium at elevated temperatures. Figure E-7 curve III shows the low cycle fatigue curve.

Incorporating a safety factor of 10 on cycles and 1.5 on stress in the curve found with the first (Equation 4-1) method gives curve IV in Figure E-6 and curve V Figure E-7 for the second method. The final design curve, Figure E-9 is a lower bound of these curves.

A comparison of the low cycle fatigue design curve obtained with the two methods is shown in curves III and IV, Figure E-8.

4.1.2 Triplanar Grid

The geometric and thermal hydraulic analysis of the triplanar grid was complete prior to the start date set for the grid stress analysis. The grid was shown to have a number of unsatisfactory performance characteristics. Fuel rod swelling produced severe flow maldistributions at the grid. Flow induced pressure drops and fuel rod temperature gradients were high. As a result of this, the triplanar grid was eliminated as a design choice and no stress analysis was done on it.

4.1.3 Clustered Rod Grid

This grid is structurally less desirable due to the secondary bending stresses induced in the dog-legged grid ligaments by all forms of loading. Both strength and stiffness suffer as a result of this feature. Table 4-1 summarizes the peak stresses in this grid.

4.1.4 Honeycomb Grid

This grid has optimum structural properties. All structural members are straight and fully triangulated. Table 4-2 summarizes the peak stresses in this grid.

TABLE 4-1

MARGIN OF SAFETY CLUSTER GRID DESIGN

Stress Location	Stress Description & Failure Mode	Stress or load	Allowable stress or load	M. S. ^a or U. F. ^b	Ref. Section
	<u>Under vertical static load</u>				
Wrinkled strap across a radial web, Section C, Figure G-3	Combined vertical and lateral bending, highest stress	$14.7 \times 10^3 \text{ lb/in}^2$	$35 \times 10^3 \text{ lb/in}^2$	1.4	E-3.2.2
Wrinkled strap across a radial web, Section A, Figure G-3	Highest shear stress	$18.5 \times 10^3 \text{ lb/in}^2$	$20 \times 10^3 \text{ lb/in}^2$	0.1	E-3.2.2
Radial web in center of the grid	Bending stress	$12.7 \times 10^3 \text{ lb/in}^2$	$35 \times 10^3 \text{ lb/in}^2$	1.8	E-3.2.2
Radial web in center of the grid	Lateral buckling	61 lb/in	122 lb/in	1.0	E-3.2.2
	<u>Thermal Induced loading</u>				
Radial web with rigid can . . .	Buckling stress due to radial, thermal induced compressive and static bending moment	$30.3 \times 10^3 \text{ lb/in}^2$	$50.3 \times 10^3 \text{ lb/in}^2$	0.7	E-4.5
With flexible can.		$61 \times 10^3 \text{ lb/in}^2$	$153 \times 10^3 \text{ lb/in}^2$	1.5	E-4.5
	<u>Low cycle fatigue failure</u>				
Wrinkled straps most critical point B in Figure E-35	Static binding stress and alternating stress	$92 \times 10^3 \text{ lb/in}^2$	---	0.03	E-5.3

a. $M. S. = \frac{\sigma_a}{\sigma} - 1 = R.F.-1$

b. $U. F. = \text{Usage Factor} = \frac{\text{Number of Cycles Applied}}{\text{Number of Cycles to Failure}}$

TABLE 4-2

MARGIN OF SAFETY IN THE HONEYCOMB GRID

Stress Location	Stress Description & Failure Mode	Stress or load	Allowable Stress or load	M. S. ^a or U. F. ^b	Ref. Section
	<u>Under vertical static load</u>				
Web in center of the grid	Bending Stress	$13.1 \times 10^3 \text{ lb/in}^2$	$35 \times 10^3 \text{ lb/in}^2$	1.6	E-3.2.1
Web in center of the grid	Lateral buckling moment	31 lb/in	60.2 lb/in	0.9	E-3.2.1
Torsion bar of a spring in the center of the grid	Shear stress	$2.9 \times 10^3 \text{ lb/in}^2$	$20 \times 10^3 \text{ lb/in}^2$	6	E-3.2.1
Torsion bar of a spring in the center of the grid	Lateral buckling due to buckling of the grid	2.1 lb/in	8 lb/in	2.8	E-3.2.1
	<u>Thermal induced load</u>				
Web in the center of the grid	Buckling stress due to radial thermal induced compression and static bending moment	$37 \times 10^3 \text{ lb/in}^2$	$124 \times 10^3 \text{ lb/in}^2$	2.35	E-4.3
	<u>Low cycle fatigue failure</u>				
Web in the center of the grid	Static stress and alternating thermal stress	$24 \times 10^3 \text{ lb/in}^2$		≈ 0	E-5.1
Braze between the webs in center of the grid	Static stress and alternating thermal stress. Based on fatigue design curve of the braze Figure E-39	$16 \times 10^3 \text{ lb/in}^2$		≈ 0	E-5.2

a. $M. S. = \frac{\sigma_a}{\sigma} - 1 = RF - 1$

b. $U. F. = \text{Usage Factor} = \frac{\text{Number of Cycles Applied}}{\text{Number of Cycles to Failure}}$

4.2 SPRING FINGER ANALYSIS

Details of the spring finger analysis are given in the Appendix G. In the following, the basis for the selection of the spring dimensions is given.

4.2.1 Grid Loads

At the beginning of life in the reactor, the spring finger in the hexagonal grid is preloaded to three pounds, the spring in the cluster grid is preloaded to five pounds. The load on the spring finger increases gradually due to swelling of the fuel. At the end of life in the reactor (after one year) the springs have to sustain the preload and the deflection produced by three percent fuel swelling.

4.2.2 Limiting Stress

The maximum stress in the spring produced by the preload and by the fuel swelling must not exceed the yield stress of Inconel 718 at 1100°F.

$$\sigma_y = 125,000 \text{ lb/inch}^2$$

The maximum stress will be attained after a three percent swelling of the fuel rod. The stress produced by the preload of three pounds in the spring of the honeycomb grid and the preload of five pounds in the cluster grid should not exceed 2/3 of the yield stress:

$$\sigma = 83,400 \text{ lb/inch}^2$$

The actual stress in the spring of the end of life will not reach the yield stress because stress relaxation will reduce the spring load (See Appendix G).

4.2.3 Tolerances

In the honeycomb grid and the cluster grid a nominal material thickness of 0.015 inch has been chosen (0.0105 inch for the triplanar grid). Corrosion

may reduce the thickness by one mil. The upper tolerance for the grid material is 0.016 inch. The nominal pitch of the fuel rod is 0.29 inch. A pitch tolerance of ± 0.001 inch is allowed.

4.2.4 Parametric Study

The spring rate has been plotted in Figures G-8 through G-35 as a function of the width of the torsion bar A and the length of the torsion bar, (See Figure G-3). Plots have been produced for each combination of material thickness t and pitch P. Incorporated in the curves are the yield stress limits which are produced by a three pound preload (five pounds for the cluster grid) and a deflection due to three percent fuel swelling. The points marking the dimensions and the spring rate of the selected spring have stresses lower than the yield stress at the end of life.

4.2.5 Selection of the Spring Dimensions

The spring dimensions have been selected as follows:

1. The stress in the springs must not exceed the previously defined stresses.
2. The length of the torsion bar C should be as short as possible, because the length of the torsion bar determines the height of the grid.
3. The width of the torsion bar A is limited by the size of the grid cell.

4.2.6 Spring in the Honeycomb Grid

Figures G-1, G-3 and G-6 show the shape of the spring. The dimensions of the spring have been selected from the carpet plots on Figures G-8 to G-19. They show the spring rate as a function of the length C and the width A of the torsion bar. The hatched line marks a stress limit. Below the line the stress produced by a three pound preload and by an additional deflection of 11.25×10^{-3} inches due to three percent fuel swelling is less than the yield stress. Above the line, the maximum stress in the spring is higher than the yield stress.

Graphs have been plotted for a three-grid thicknesses -- 0.014 inch, 0.015 inch and 0.016 inch and for three rod pitches -- 0.280 inch, 0.290 inch and 0.300 inch. A spring with the following nominal dimensions has been selected (See Figure G-3 and Figure G-6 for notation):

Pitch	P = 0.290 inch
Grid thickness	t = 0.015 inch
Width of the torsion bar	A = 0.12 inch
Width of the bending cantilever	B = 0.24 inch
Length of the torsion bar	C = 0.26 inch
The spring constant is 360 lb/inch	

With all other data given, the width B of the cantilever can be calculated using the formula given in the Appendix G. B is approximately given by $B = 2 A$. The width of the torsion bar has to be smaller than full length of one web, which is 0.167 inch.

The points with the selected spring dimensions have been marked on the graphs. All points are located below the stress limit line. For the selected spring Figure G-27 shows spring load vs. deflection; the preload of three pounds requires a predeflection of 8.4×10^{-3} inch. The swelling induced deflection increases the load to 7.1 pounds. The total deflection is a result of the bending deformation of the cantilever and twisting of the torsion bar. A dimple in the middle of the torsion bar prevents bending deformation which would add to the stress in the torsion bar. Details of the analysis can be found in the Appendix G.

4.2.7 Spring in the Grid of the Clustered Rod Design

Figures G-2 and G-17 show the shape of the spring and the notation of the dimensions. The dimensions of the spring have been selected from the carpet plots on Figures G-19 through G-24. A spring with the following nominal dimensions has been selected, see Figure G-17 for notation:

Pitch	P = 0.29 inch
Grid thickness	t = 0.015 inch
Width of the torsion bar	A = 0.14 inch
Width of the bending cantilever	B = 0.15 inch
Length of the torsion bar	C = 0.35 inch
The spring constant is 320 lb/inch	

Figure G-28 shows the springload vs. deflection. The preload of five pounds requires a predeflection of 15.5×10^{-3} inches. The swelling induced deflection of 10.25×10^{-3} inch increases the load up to 8.4 pounds. The total deflection for this load is 25.5×10^{-3} inch.

Details of the analysis can be found in the Appendix G.

4.2.8 Spring in the Triplanar Grid

Figure G-4 shows the shape of the spring and gives the notation for the dimensions. The dimensions of the spring have been selected from the carpet plots on Figures G-30 through G-35. They show the spring rate as a function of the length C of the torsion bar and its angle α , see Figure G-4. The angle between the applied load P and the center of the torsion bar which determine the width A is

$$\alpha/2 + \beta = 90^\circ$$

In this configuration the load applies a maximum moment to the torsion bar. The hatched lines in Figures G-30 through G-35 mark the stress limit. Below the line the stress produced by a three pound preload and by an additional deflection due to three percent fuel swelling, is less than the yield stress. Above the line the maximum stress in the spring is higher than the yield stress. Graphs have been plotted for two grid thicknesses 0.0095 inch, 0.0115 inch and for three pitch distances 0.28 inch, 0.29 inch, 0.30 inch.

A spring with the following nominal dimensions has been selected, see Figure G-4 for notation.

Pitch	P = 0.29 inch
Grid thickness	t = 0.0105 inch
Width of the torsion bar	$\alpha = 60$ degrees
Width of the bending cantilever	B = 0.25 inch
Length of the torsion bar	C = 0.2 inch

The spring constant of the selected spring = 155 lb/inch.

The points with the selected spring dimensions have been marked in the plots. All points are located below the stress limit line. Figure G-29 shows the springload vs. deflection. The springload of three pounds requires a predeflection of $\delta = 0.0195$ inch. The swelling induced deflection of 0.009 inch increase the load to 4.4 pounds. The total deflection for this load is 28.5×10^{-3} inch.

Details of the analysis can be found in Appendix G.

4.3 FLOW INDUCED VIBRATIONS

The analysis of the fluid induced vibration is based on Burgreen's correlation.^[24] An extension of this theory has been made. It includes the effect of an axial compressive load on the vibration amplitude; the amplitude grows considerably with increasing axial load in the fuel rod. The axial load is produced by friction between the rods and the grids. Only the vibration of single rods has been considered.

The vibration amplitude is a function of the elastic modulus. Since the tube is subject to plastic hoop strains produced by swelling of the fuel, it was necessary to determine an effective modulus governing the vibration characteristics of the cladding material. Details of the analysis can be found in Appendix F.

A maximum vibration amplitude of 0.001 inch has been proved by experience with PWR reactors to be a safe upper limit. The vibration analysis revealed that both linear and rotational restraint must be supplied to the rod by the grid in order to keep the vibration amplitude below 0.001 inches. For a fluid velocity of 30 ft/sec and a 9.7 inch distance between the grids, the maximum vibration amplitude during the lifetime of the fuel rods in the reactor is 0.00085 inch.

An additional analysis shows that a spring force of three pounds is sufficient to clamp the fuel rod. The spring load clamps the fuel rod against dimples 0.5 inch above and below the spring. The clamping moment provided by the spring is

$$M_{cl} = 0.6 \text{ lb/inch} \quad (4-2)$$

The maximum bending moment at the clamped end of the fuel rod due to vibration with 0.001 inch amplitude is, in the upper limit,

$$M_v = 0.22 \text{ lb/inch} \quad (4-3)$$

The lowest reserve factor is

$$\frac{M_{cl}}{M_v} = 2.7 \quad (4-4)$$

4.4 FUEL ROD AXIAL CREEP BUCKLING

4.4.1 Statement of Problem

The fuel rod support grids in the proposed fuel assembly designs are attached either directly to the duct wall, or to a structural can located within the duct. In either case the temperature of the majority of the fuel rods will be higher than that of the grid locating structure. The differential thermal expansion associated with this temperature distribution is accommodated by allowing the fuel rods to slide through the grids. Frictional loads generated at the grids by this sliding action are felt by the fuel rods as compressive end loads. The constraints associated with these compressive loads are held constant for long periods of time: the resulting mode of deflection therefore, includes elastic, plastic and creep deformations. A further factor influencing the creep buckling behavior is the state of stress set up in the clad by fuel swelling. It should be noted that the strut loading described above is strain controlled; the post buckling strut behavior thus consists of a drop

in the compressive loading to an equilibrium value, accompanied by a finite transverse deflection of the strut center line. The transverse deflection of the fuel rods can produce significant changes in the geometry of the coolant flow channels. This latter effect is the primary reason for the interest in fuel rod axial creep buckling. A means is required therefore to predict the creep buckling load for a fuel tube and to establish the transverse deflections in the pre and post buckling conditions.

4.4.2 Method of Analysis

4.4.2.1 Basic Strut Buckling Equation

The dominant loading mechanism influencing the state of stress in the clad is fuel swelling. This can produce plastic hoop strains of up to three percent. The behavior of struts subject to this type of loading is being investigated experimentally as a part of the Westinghouse LMFBR program. Compression tests are being performed on hollow pressurized struts, using pressures giving considerable plastic strains in the hoop direction. Initial results from these tests indicate that the buckling load can be predicted by using a modified form of the Euler buckling equation. The modification is contained in the effective elastic modulus (E') used. E' is found to be the secant modulus obtained by entering the stress strain curve with the generalized stress in the clad, i.e.

$$P_{crit} = \frac{C \cdot \pi^2 \cdot E' \cdot I}{l^2} \quad (4-5)$$

In order to account for the extended on-load dwell period the static stress-strain curve is replaced by an isochronous stress-strain curve. All points on the latter type of curve incorporate elastic, plastic and creep strains accumulated over a given period under conditions of constant load. A unique curve exists for each on load dwell period. The value of E' obtained from this curve is considerably lower than that obtained using the static stress-strain curve.

4.4.2.2 Clad State of Stress

Let ϵ_R be the clad terminal swelling strain and T the time period over which this swelling strain is produced.

$$\text{Straining rate} = \dot{\epsilon}_R = \frac{\epsilon_R}{T} \quad (4-6)$$

A low strain rate stress-strain curve is constructed for each strain rate of interest. These curves are produced when strain is increased at a uniform but low rate throughout the test: the increase in stress produced by an increase in strain is accompanied by a decrease in stress due to stress relaxation.

The resultant stress is considerably lower than that obtained in a short time tensile test. In practice, it is not possible to produce low strain rate stress-strain curves directly by test. A means of constructing these curves using available stress relaxation data is given in Section 4.5.

Entering the appropriate low strain rate stress-strain curve with the reference swelling strain, the effective hoop stress σ_H is obtained. Using this stress the effective interface pressure between the fuel pellet and the clad is obtained as:

$$\Delta P = \frac{t \cdot \sigma_H}{R} \quad (4-7)$$

The mean radial compressive stress is now obtained as:

$$\sigma_R = -\frac{\Delta P}{2} \quad (4-8)$$

It should be noted that if the fuel rod is subject to an internal pressure in excess of the value of ΔP given by equation 4-7, then this value replaces ΔP in equation 4-8 and is used to calculate σ_H using membrane theory. In general, this will not be the case.

The longitudinal stress due to axial loading, bending and temperature gradients is obtained as σ_L . Depending upon the assumptions made regarding the axial constraint of fuel swelling, the value of σ_L may include a contribution due to clad axial straining. Since the stress associated with axial swelling strains is tensile, a conservative buckling result is obtained by neglecting it. The generalized stress required is the average value for the tube wall, hence thermal stresses produced by heat transfer are not included in the stress components.

The generalized stress in the clad is now obtained using equation (4-9).

$$S = \sqrt{\frac{(\sigma_H - \sigma_R)^2 + (\sigma_R - \sigma_L)^2 + (\sigma_L - \sigma_H)^2}{2}} \quad (4-9)$$

4.4.2.3 Strut Transverse Deflections

The Westinghouse pressurized strut tests have shown that the equation of the Southwell plot correlates well with experimental results under conditions of high circumferential plastic straining. This equation is adopted therefore, as the basic analytical model for calculating fuel rod transverse deflections, i.e.

$$S = \frac{a}{1 - \frac{P}{P_{crit}}} \quad (4-10)$$

In order to be made applicable to the fuel rod operating conditions, equation (4-10) as given above requires modification to account for the self limiting nature of the applied loading and the effect of stress relaxation.

Consider the geometry of a pin jointed strut in the buckled condition (Figure 4-1). The neutral axis of the buckled strut is assumed represented by an arc of a circle. In deflecting from the straight to the curved condition with no movement of points A and B an increase in center line length must occur. This increase in length is produced by the partial release of the compressive strain in the strut.

Figure 4-1. Pin Jointed Street Buckled Condition Geometry

$$\ell_c = \ell_F (1 - \varepsilon_c) \quad (4-11)$$

$$\ell_c = 2.R.\sin \theta \quad (4-12)$$

$$\ell_F = 2.R.\theta \quad (4-13)$$

$$\frac{\ell_c}{\ell_F} = \frac{\sin \theta}{\theta} \quad (4-14)$$

Expand $\sin \theta$ using Maclaurin's theorem and substitute in equation (4-14). Since θ is small terms involving θ^n may be neglected for $n > 5$. This gives

$$\theta^4 - 20 \theta^2 + 12 \theta \varepsilon_c = 0$$

Solving for θ and eliminating non-meaningful solutions we obtain

$$\theta = (10 - (100 - 120 \varepsilon_c)^{1/2})^{1/2} \quad (4-15)$$

For small values of θ we can write

$$\delta' = \frac{\ell_F}{2} \cdot \frac{\theta}{2} \quad (4-16)$$

Hence

$$\delta' = \frac{\ell_F (10 - (100 - 120 \varepsilon_c)^{1/2})^{1/2}}{4} \quad (4-17)$$

Equation (4-17) has been used to produce the dimensionless deflection plot given in Figure 4-2. This plot indicates that an alternative form for representing the dimensionless transverse deflection is

$$\frac{\delta'}{\ell_F} = C \cdot \epsilon^D \quad (4-18)$$

Constants C and D are obtained from Figure 4-2 to give

$$\delta' = 0.613 \cdot \ell_F \cdot \sqrt{\epsilon_c} \quad (4-19)$$

Equation (4-19) gives the transverse deflection of a pin jointed strut. By means of a similar analysis it can be shown that the corresponding deflection for a strut with fixed ends is given by

$$\delta' = 0.433 \cdot \ell_F \cdot \sqrt{\epsilon_c} \quad (4-20)$$

The decrease in the constraint induced compressive end load produced by the transverse deflection δ' can now be written as

$$\Delta P' = \left(\frac{\delta'}{0.613 \cdot \ell_F} \right)^2 \cdot E \cdot A_s \quad (4-21)$$

Implied in equation (4-21) is the fact that the final unloading produced by strut buckling will be sufficiently rapid such that the elastic stress-strain curve will govern the loss of compressive stress. The equilibrium value of δ' is found by substituting $P' = P - \Delta P'$ in equation 4-10 and noting that:

$$\delta' = \delta - a$$

Stress relaxation prior to buckling will reduce the initial elastic constraint strut loading from P_R to P . In Section 4.5 the analytical model proposed to predict the rate of stress relaxation is of the form

Figure 4-2. Limiting Transverse Deflection of a Buckled Fuel Rod

$$R_s = B.T^A \quad (4-22)$$

Using this model the residual strut loading at time T is obtained as

$$P = \frac{P_R (100 - B.T^A)}{100} \quad (4-23)$$

Substituting the value of P from (4-23) into (4-10) we obtain the final strut deflection equation, as

$$\delta = \frac{a}{1 - \left[\frac{\frac{P_R}{100} (100 - B.T^A) - \left(\frac{\delta - a}{K \cdot \ell_F} \right)^2 \cdot E \cdot A_s}{P_{crit}} \right]} \quad (4-24)$$

For pin jointed struts $K = 0.613$

For fixed end struts $K = 0.433$

In equation (4-24) both P and P_{crit} are time dependent. A complete analysis of the strut transverse deflections thus involves a lifetime survey.

4.4.3 Notation for Section 4.4

Axial load decrease (constraint induced)	$\Delta P'$
Buckled strut radius of curvature	R inch
Clad inside-radius	R inch
Clad-pellet interface pressure	ΔP P.S.I.
Clad thickness	t inch
Compressive strain released by buckling	ϵ_c inch/inch
Effective modulus	E' lb/inch ²
Elastic modulus	E lb/inch ²
End fixity factor	C
Generalized stress	S lb/inch ²

Hoop stress	σ_H lb/inch ²
Initial elastic constraint strut load	P_R lb
Longitudinal stress	σ_L lb/inch ²
Radial stress	σ_R lb/inch ²
Rate of stress relaxation	R_s lb/inch ² /hr.
Reference swelling strain	ϵ_R inch/inch
Rotation of the strut ends	θ rads
Second moment of area	I inch ⁴
Strut (fuel rod) applied axial comp. loading	P lb
Strut (fuel rod) critical axial loading	P_{crit} lb
Strut (fuel rod) cross sectional area	A_s inch ²
Strut (fuel rod) initial eccentricity	a inch
Strut (fuel rod) transverse deflection	δ inch and δ' inch
Strut length (grid pitch)	ℓ inch
Strut length in the buckled condition	ℓ_F inch
Strut length prior to buckling	ℓ_c inch
Time on load	T hrs.

4.5 TIME DEPENDENT STRESS-STRAIN CURVES

4.5.1 Low Strain Rate Stress-Strain Curves

Straining rates associated with fuel swelling induced straining of the clad lie in the range 1×10^{-6} to 5×10^{-6} inch/inch/hr. At temperatures within the creep range, stress-strain curves produced at these low strain rates are influenced considerably by stress relaxation. No practical means is available for producing these curves directly by means of low strain rate tensile tests at elevated temperatures. A means is required therefore to construct the curves using currently available data on the short time stress-strain properties and stress-relaxation properties of the material.

Stress relaxation data for Type 304SS at clad operating temperatures are available. As with load controlled creep, stress relaxation is known to take place at a rate which varies with time. Under conditions of continually increasing strain it is reasonable to assume that the stress-relaxation behavior contributing most to the formation of the low strain rate stress-strain

curve, will be that analogous to primary creep on the conventional creep curve. The available data do not extend into this region. The data required are thus obtained by extrapolating the available data using equation (4-25).

$$R = B \cdot t^A \quad (4-25)$$

Where

R = Rate of stress relaxation, percent/hr

t = Time between the last application of a strain increment and the measurement of stress, hrs.

A and B are constants obtained from test data.

Stress relaxation data for Type 304SS at temperatures ranging from 1000°F to 1300°F have been produced at the Westinghouse Astronuclear Division. Typical test values obtained at 1200°F are

Time	0 hrs.	100 hrs.	10 ⁴ hrs.
σ lb/inch ²	20,000	13,000	4,000
Percent reduction	0	35	80
Percent reduction/hr.	0	0.35	0.008

Substituting the two sets of data given above into equation (4-25) and solving for A and B we obtain

$$A = -0.8205$$

$$B = 15.31$$

Assume the strain to be applied in small increments, i.e. as a step function rather than a ramp increase. The stress σ_2 immediately following the application of the strain increment ϵ_I is known in terms of the previous stress σ_1 and the known characteristics of the short time stress-strain curve. The stress increment produced by the application of strain ϵ_I is thus defined as

$$\Delta\sigma_I = \sigma_2 - \sigma_1 \quad (4-26)$$

Two basic methods for constructing the low strain rate stress-strain curves can now be proposed.

1. The stress governing the stress relaxation behavior is assumed to be that existing immediately following the application of a strain increment. The stress relaxes from this value until a further strain increment is applied. This behavior is summarized in Figure 4-3A. The low strain rate stress-strain curve given by this straining model is summarized in equation (4-27).

$$\sigma_3 = \frac{\sigma_2 (100 - 15.31 t^{0.1795})}{100} \quad (4-27)$$

Where $t = \epsilon_I / \dot{\epsilon}$

2. The stress produced by the instantaneous application of each strain increment ϵ_I is assumed to relax in the manner described by equation (4-27). In this case, however, the value of t used is the total time lapsed since the application of the strain increment, and stress σ_2 is replaced by stress increment $\Delta\sigma_I$, the resultant stress corresponding to strain N . ϵ_I is now obtained as

$$\sigma_N = \sum_{I=1}^{I=N} \frac{\Delta\sigma_I (100 - 15.31 \{t[N - I + 1]\}^{0.1795})}{100} \quad (4-28)$$

Both methods have been used to construct curves. Method (1) is found to be oversensitive to the length of the strain increment ϵ_I used. As the increment length is reduced the resulting curves become very flat, with stress becoming almost independent of strain. This erroneous result derives from the

assumption that the total stress will at any instant relax at the same uniform rate regardless of how the stress has been built up. Since the assumed rate of relaxation starts again at the primary rate following the addition of each strain increment, the effect of primary relaxation is overestimated.

Method (2) is designed to eliminate this defect. Test curves constructed using this method show that as the strain increment length ϵ_I becomes small, the resulting curve becomes insensitive to changes in increment length.

The ISOC 2 computer code has been written to construct low strain rate stress strain curves using method (2) above. Using this code, the length of the strain increment ϵ_I can be shortened to the point at which the resulting stress-strain curve becomes virtually independent of ϵ_I . The resulting curves are given in Figures 4-4 and 4-5.

4.5.2 Isochronous Stress-Strain Curves

Lifetime studies of buckling require the use of a wide range of isochronous stress-strain curves. In order to consider a cycle of 70 days, isochronous stress-strain curves would be needed for a minimum of eight time periods, all less than 70 days in order to define the behavior.

The data available were for time periods of 0.5 years to 3 years. The curves required would be in the range 0 to 0.2 years. A means is required to extend the available data to other time periods.

A mathematical model is chosen to best fit the available data. It should be time dependent in the secondary creep range. The strain model adopted is

$$\epsilon = \frac{\sigma}{E} \quad \sigma \leq A_2 \quad (4-29)$$

$$\epsilon = \frac{\sigma}{E} + K_2 (\sigma - A_2)^{B_2} T \quad A_2 \leq \sigma \leq A \quad (4-30)$$

$$\epsilon = \frac{\sigma}{E} + K_2 (\sigma - A_2)^{B_2} T + K_1 (\sigma - A)^B \quad A \leq \sigma \quad (4-31)$$

A. Construction Number 1

B. Construction Number 2

Figure 4-3. Low Strain Rate Stress-Strain Curve Constructions

Figure 4-4. Low Strain Rate (1, 2, 3, 4, 5 % Per Year) Stress-Strain Curves for Type 304 SS at 1100°F.

Figure 4-5. Low Strain Rate (6,8,10% Per Year) Stress-Strain Curves for Type 304 SS at 1100°F

This model is chosen since it best fits the curves for times $0.5 \text{ year} \leq t \leq 3 \text{ year}$. Using the constants found by fitting equations (4-29) to (4-31) to the available curves, the required additional curves were generated. These are given in Figure 4-6.

4.5.3 Notation for Section 4.5

Constants	A & B & K
Elastic modulus	E lb/inch ²
Increment identification	N
Rate of stress relaxation	R percent/hr
Strain	ϵ inch/inch
Stress	σ lb/inch ²
Time	T hrs

4.6 LOCALIZATION OF CIRCUMFERENTIAL STRAINS

4.6.1 Statement of Problem

Under the action of fuel swelling a contact pressure P psi is generated at the fuel-clad interface. In a uniform tube this pressure P produces uniform straining of the clad. The interface pressure is self-limiting and the resulting clad straining ceases when the clad membrane stresses are in equilibrium with pressure P . The effect of the shear webs in the rod cluster design is to produce a circumferential variation of clad strain. An idealized fuel clad-shear web geometry is shown in Figure 4-7. At the tube-shear web junction the tube wall is locally thicker and hence stiffer than the adjacent wall structure. Under the action of a uniform load therefore, segment 0 will not strain as much as segments 1, 2, etc. As a result of this action, segments 1, 2, etc. will receive additional strains and slide over the fuel pellet towards segment 0. This sliding action will produce resisting frictional shear forces q_j which will in turn produce shear deflections in both the clad and the fuel pellet. Due to this action the additional strains in segments 1 to n are not uniformly distributed and strain peaking occurs in segment 1.

Figure 4-6. Isochronous Stress-Strain Curves for Type 304 SS at 1100°F

Figure 4-7. Idealized Fuel Clad-Shear Web Geometry

4.6.2 Method of Analysis

As shown in Appendix E, the resulting tangential force distribution satisfies the following first-order difference equation:

$$F_j - F_{j-1} = -g_j \ell_j, \quad (4-32)$$

with solution

$$F_j = F_0 - \sum_{i=1}^j g_i \ell_i \quad (4-33)$$

The mean stress in segment j is

$$\sigma_j = \frac{F_j + F_{j-1}}{2t_j} \quad (4-34)$$

where t_j is the thickness of segment j . The corresponding strains ϵ_j are determined by the stress-strain characteristics of the clad material.

Several low strain rate stress-strain curves are input as libraries of points. Each input curve gives strain as a function of stress for a given strain rate. Assuming that all strains occur in the same time U , the strain ϵ_j caused by a stress σ_j is equal to the strain value at the intersection of the line $\sigma = \sigma_j$ with the stress-strain curve which corresponds to a rate of ϵ_j/U . In most cases, neither the required strain ϵ_j nor the required strain rate ϵ_j/U is one of input values, so interpolation is necessary. Linear interpolation is used between points on each stress-strain curve. This method is not very accurate for interpolating between curves for different strain rates; a more exact approximation is required. Details of the interpolation method are given in Appendix D. In the earlier version of the code (SISINUC Phase I) the interpolation between curves corresponding to different strain rates was not included. The strain rate was assumed constant for all segments at an input reference rate, ϵ_{IR}/U , which was chosen to correspond exactly to one

of the input curves. In the final version, SISINUC Phase II, the logarithmic interpolation developed in the appendix was used.

Equation (4-33) does not describe a unique solution until the constant F_0 is evaluated. Its value is determined by the pressure loading P . However, it is more convenient to specify a reference strain ϵ_{IR} and then compute P , the pressure required to produce this strain in an equivalent uniform tube. The stress in such a tube is given by $\sigma_R = \frac{PR}{t}$. The constant F_0 is then evaluated by finding that segment $K-1$ for which

$$F_{K-1} = \sigma_R t \quad (4-35)$$

and using equation (4-33) with $j=K$. The segment $K-1$ is that for which the average strain in segments 0 to $K-1$ is equal to ϵ_{IR}

$$\sum_{j=0}^{K-1} \epsilon_j \ell_j = \epsilon_{IR} \sum_{j=0}^{K-1} \ell_j \quad (4-36)$$

The value of K is determined by iteration. Initially, $K=2$; then K is incremented and a set of strains ϵ_j is calculated using equations (4-33) and (4-34) with the assumption that condition (4-35) is true for the present value of K . If equation (4-36) is not satisfied by this computed set of strains, K is incremented again and the procedure is repeated. The problem is completed when the assumption that (4-35) holds for a given K leads to a computed set of strains which satisfies (4-36), to within an error caused by the finite size of the segments.

The SISINUC code has been extended to handle two strain localization mechanisms in addition to the shear web problem: 1) a local defect producing a thinning of the cladding; and 2) a local increase in temperature causing a local weakening of the cladding. The modifications required for analyzing these problems are straightforward and are discussed Appendix D.

4.6.3 Results

All three types of problems were considered and solutions were determined by SISINUC for several values of the input parameters. In all cases, the following parameters were fixed: 1) the mean radius and thickness of the clad were 0.117 inch and 0.016 inch, respectively; 2) the straining time, U , was one year; 3) the number of segments, N , was 1800; and 4) eight input stress-strain curves generated by ISOC², at strain rates of 1 percent, 2 percent, 3 percent, 4 percent, 6 percent, 8 percent, and 10 percent per year were input for each temperature.

Twenty-seven shear web problems were run, at a temperature of 1100°F and with the width of the shear web equal to 0.02 inch. These 27 cases were all possible combinations of three values assigned to each of the remaining parameters: 1) the effective diffusion length of the shear web, X , took on values 0.0035, 0.0070 and 0.0105 inches; 2) the reference strains, ϵ_{IR} , were 1 percent, 2 percent, and 3 percent; and 3) the coefficients of friction were 0.4, 0.8, and 1.2. The strain distribution calculated by SISINUC for a typical set of parameters ($X = 0.007$ inches, $\mu = 1.2$, all values of ϵ_{IR}) is plotted in Figure 4-8 for the region outside the shear web. Also shown in Figure 4-8 is the strain distribution determined by SISINUC Phase I for the shear web problem with $X = 0.007$ inch, $\mu = 1.2$, and $\epsilon_{IR} = 1$ percent. As expected, the peak strain calculated by this method is higher than that calculated by ISINUC Phase II, since the variation of the strain rate is not taken into account. Table 4-3 lists the peak strain values for all 27 cases and these results are plotted in Figure 4-9. Figure 4-10 is a carpet plot of the peak strain concentration, $\epsilon_{max}/\epsilon_{IR}$, vs. coefficient of friction, μ , and reference strain, ϵ_{IR} , for a typical value of $X(0.007$ inches).

Nine local defect problems were run with ϵ_{IR} and μ taking on the same values as for the shear web problems and X , now a measure of the equivalent thinning caused by the defect, was constantly - 0.00012 inches or 3/4 percent of the clad thickness. This value was determined by averaging the maximum permissible defect thickness over a length $\lambda^{-1} = (1.285/RT)^{-1}$ in the axial direction. The width of the defect was also the maximum allowed by the clad inspection standard, 0.004 inches. Table 4-4 lists the peak strains computed by SISINUC

TABLE 4-3
PEAK STRAIN PERCENT FOR SHEAR WEB PROBLEM

Reference Strain (percent)	X(inch) μ	0.4	0.8	1.2
1.0	0.0035	1.6064	1.8031	1.9408
	0.0070	1.7253	2.0414	2.2969
	0.0105	1.7315	2.0565	2.3441
2.0	0.0035	2.9897	3.3378	3.5556
	0.0070	3.3102	3.8614	4.2784
	0.0105	3.3761	4.0079	4.5521
3.0	0.0035	4.3716	4.8220	5.1633
	0.0070	4.8147	5.5659	6.1401
	0.0105	4.9793	5.8583	6.5643

TABLE 4-4
PEAK STRAIN PERCENT FOR DEFECT PROBLEM

Reference Strain (percent)	X(inch)	Peak Strain %		
		$\mu = 0.4$	$\mu = 0.8$	$\mu = 1.2$
1.0	-0.00012	1.0060	1.0009	0.9910
2.0	-0.00012	2.0015	2.0015	1.9860
3.0	-0.00012	3.0136	3.0021	2.9788

Figure 4-8. Typical Strain Distribution Produced by the Shear Web in a Fuel Rod Cluster

Figure 4-9. Influence of Assumed Values of x and μ on Calculated Peak Clad Strain in a Clustered Rod Design

Figure 4-10. Peak Clad Strains in the Clustered Rod Design

for the local defect problems. Since $\epsilon_{\max}$ is so close to ϵ_{IR} , the strain distribution is essentially uniform.

Twenty-seven hot spot problems were run, with ϵ_{IR} and μ varying as above and X taking on the values 0.0, - 0.00004 inches, and -0.00012 inches. The variations of X made it possible to investigate the straining distribution resulting from a hot spot coincident with a local defect. The width of the hot spot region was taken as 0.0125 inches or about 6 degrees. Figure 4-11 is the strain distribution for a typical case ($X = 0.0$, $\mu = 0.8$, all values of ϵ_{IR}). Table 4-5 presents the peak strain as a function of the three variable parameters and Figure 4-12 is a graph of these results. Figure 4-13 is a carpet plot of the peak strain concentration vs. μ and ϵ_{IR} for a typical value of $X(0.0)$.

4.7 FUEL ROD THERMAL BOW

The computer program FATBO2 analyzes the fuel assembly thermal bow. It considers the single rod configuration with honeycomb grid type rod spacing. The number of grid supports, the grid spacing and the number of rods across the assembly are variables.

The temperature distribution is input to the code by giving the average temperature of each sublength (a length of rod between adjacent grids) in each flow channel, the flow channel being a row parallel to one set of flats of the hexagonal cross section. Plots of the initial set of temperatures used are given for two axial locations of the assembly in Figure 4-14. The modulus of elasticity of each rod sublength is calculated since it is a function of the temperature.

The rods may be fixed or pin supported by the grids. The grids may be fixed to the duct directly or to a 'can' which is pin supported by the duct. The grids must be fixed to the duct or can to maintain grid spacing. A benefit of the can is that it acts as a flow baffle and decreases edge temperature gradients in the fuel assembly.

Duct deflections calculated by PNL are considered to be rigid and when the canned assembly is analyzed, the can must deform within the duct. This means

TABLE 4-5

PEAK STRAIN PERCENT FOR HOT SPOT PROBLEM

Reference Strain (percent)	X(inch) μ	0.4	0.8	1.2
1.0	0.0	1.4149	1.3025	1.2306
	-0.00004	1.4269	1.3122	1.2325
	-0.00012	1.4502	1.3304	1.2361
2.0	0.0	2.6584	2.4997	2.3955
	-0.00004	2.6784	2.5099	2.3969
	-0.00012	2.7203	2.5481	2.4395
3.0	0.0	3.9769	3.7148	3.5718
	-0.00004	4.0168	3.7303	3.5735
	-0.00012	4.0829	3.7918	3.6151

Figure 4-11. Typical Strain Distribution Produced by the Hot Spot in a Wire Wrap Design

Figure 4-12. Influence of Assumed Values of X and μ on Calculated Peak Clad Strain in a Wire Wrap Design

Figure 4-13. Peak Clad Strain Concentrations in the Wire Wrap Design

Figure 4-14. Coolant Temperature Distribution for Honeycomb Grid with Flow Baffle

that the can deflections must be within the limits of the duct deflections plus or minus a given gap.

The code uses the displacement method of matrix analysis to find interaction forces and final displacements. The full specification of the code is given in Appendix A. First the grid displacements of the assembly are found. Each rod region is then allowed to bow freely due to its temperature distribution and then is brought back to match the transverse deflection pattern of the grids. In this manner the slopes and transverse deflections of each rod are found at all grid locations and the latter is also found at all center span locations.

Axial loads on the rods due to friction between the rod and its spring finger support are calculated. Before using the modified Southwell Equation (developed in Section 4.4) the effect of relaxation of the axial load is allowed for (Figure 4-15).

The Euler buckling load (using the secant modulus to the isochronous stress-strain curve), the creep deflection (using the modified Southwell equation) and flow channel changes due to buckling are calculated.

Figures 4-16 and 4-17 show the results of the in-reactor time study on the secant modulus. Results are given for 1 percent, 2 percent and 3 percent end of life swelling strains for the third of three 70-day cycles. The secant modulus is dependent on the isochronous stress-strain curve for the period considered, thus for a given state of stress, the secant modulus decreases in time. Figures 4-16 and 4-17 differ slightly due to the difference in equivalent states of stress of the pinned and fixed rods.

The life study included a plot maximum deflection for the 1 percent, 2 percent and 3 percent end of life swelling strains for the third of three 70-day cycles. Figures 4-18 and 4-19 give the fuel rod maximum mid-span transverse deflection for the pinned and fixed rods. These are found using the modified Southwell equation and are, therefore, dependent on the Euler buckling load and the axial load which are both time dependent. The figures show the beginning of the cycle to be the critical time as far as creep deflection is concerned.

Figure 4-15. Variation of Strut Constraint Loading with Time

Figure 4-16. Variation in Fuel Rod Secant Modulus (10^6 PSI)
Under Full Power Loading Condition; Rods Pinned at Grids

Figure 4-17. Variation in Fuel Rod Secant Modulus (10^6 PSI)
 Under Full Power Loading Conditions: Rods Fixed at Grids

Figure 4-18. Typical Life Time Variation in Fuel Rod Mid-Span Transverse Deflection: Grid Supports Pinned

Figure 4-19. Typical Life Time Variation in Fuel Rod Mid-Span Transverse Deflection: Grid Supports Fixed

Figures 4-20, 4-21 and 4-22 show the maximum center span transverse deflection for the beginning of the third cycle for 1 percent, 2 percent, and 3 percent end of life strains and various grid spacing. Figure 4-20 gives the deflections for the pinned rods, while Figure 4-21 gives fixed rod deflections. The inflection points in these figures on lines of constant end of life strain have been found to be due to the non-linear effect of the axial loading where

$$P_{\max} = \begin{cases} \frac{N1}{2} (\Delta P) & N1(\text{even}) \\ \frac{N1-1}{2} (\Delta P) & N1(\text{odd}) \end{cases} \quad (4-36)$$

where:

$P_{\max}$ = axial load in center region or regions

$N1$ = number of grids

ΔP = increase in load due to friction at one grid

Figure 4-22 is a plot for the fixed rods case in which we adjust the load such that

$$P_{\max} = \frac{N1}{2} (\Delta P) \quad \text{for all } N1. \quad (4-37)$$

Figures 4-23 through 4-26 give some flow channel profiles for case 3 (see specification step 17 in Appendix A for the description of case 3) for the beginning of the third cycle.

Figure 4-20. Variation to Fuel Rod Mid-Span Transverse Deflection

Figure 4-21. Variation to Fuel Rod Mid-Span Transverse Deflection

Figure 4-22. Hypothetical Curves Showing Effect of Linear Variation in Axial Loading on Mid-Span Transverse Deflections

Figure 4-23. Typical Coolant Flow Channel Profiles

Figure 4-24. Typical Coolant Flow Channel Profiles

Figure 4-25. Typical Coolant Flow Channel Profiles

Figure 4-26. Typical Coolant Flow Channel Profiles

CHAPTER 5

THERMOFLUIDS DESIGN ANALYSES

5.1 DISCUSSION

This section presents the thermofluids performance characteristics of the three grid spacer concepts considered in this study. These grid concepts are the honeycomb, rhombus-cluster, and tri-planar grids; their descriptions are presented in Chapter 3 of this report. Whenever possible, comparative evaluations of the three grid concepts were made. These comparisons, along with non-comparative analyses, are presented in the following sub-sections of this report. Comparisons with a wire wrap fuel assembly are also made where possible.

The results of some of these comparisons are presented in Table 5-1. These results indicate that the honeycomb grid concept offers the most promise; the rhombus-cluster and tri-planar grids are calculated to have some serious thermofluids limitations. Because the honeycomb grid offered more promise than the other grid concepts, it was decided about half-way through this study to place the major remaining emphasis on this concept. As a result, Sections 5.2 to 5.5 and Section 5.8 are general and offer comparisons of the concepts while Sections 5.6 and 5.7 are concerned only with the honeycomb grid.

In order to make this study meaningful, current parameters for the FFTF were used. These thermofluids data were obtained from Reference 33 and were the current recommended design parameters at the start of this study. The thermofluids data, presented in Table 5-2, were used for nominal design conditions. The engineering hot channel factors are from Reference 33 and not those presented in this report; these hot channel factors are presented here because they were used for preliminary studies before those tabulated in Section 5.7 were available. Perturbations and parameter studies were made around this reference condition.

TABLE 5-1

COMPARISON OF THERMOFLUIDS ANALYSES
OF THE THREE GRID CONCEPTS

Grid Concept	Honeycomb	Rhombus-Cluster	Tri-Planar
Pressure Drop	Good	Very Good	Fair
Coolant Temperature Distribution	Good	Poor	Good
Circumferential Temperature Variations	Fair	Not Available	Poor

TABLE 5-2
FFTF THERMOFLUIDS DATA

GENERAL

Reactor Power, MWt	350 - 450
Inlet Temperature, °F	550
Avg. Outlet Temperature, °F	900
Flow Rate/Subassembly, 10^6 lb/hr	0.144

DIMENSIONS

Fuel Pin OD, inches	0.250
Fuel Pin ID, inches	0.218
Fuel Rod Pitch to Diameter Ratio ^a	1.12, 1.16, and 1.20
Fuel Pins Per Assembly ^d	217
Active Core Length, Inches	32.0
Unheated Pin Length Below Active Fuel, Inches ^b	0.75
Unheated Pin Length Above Active Fuel, Inches ^b	36.4
Total Fuel Pin Length, Inches	69.15

HOT CHANNEL FACTORS^c

Peak to Average within Subassembly	1.17
Peak Subassembly Power to Core Average Power	1.11
Total Nuclear Radial, F_R^N	1.30
Nuclear Axial, F_Z^N	1.24
Engineering Heat Flux, F_q^E	1.14
Total Heat Flux, F_q^t	1.84
Enthalpy Rise Factor, $F_{\Delta H}^E$	Not Established

TABLE 5-2 (Cont)

HEAT FLUX CONDITIONS^d

Average Linear Power Rating, kw/ft	7.85
Maximum Linear Power Rating, kw/ft	14.4
Average Heat Flux, 10^6 BTU/hr-ft ²	0.409
Maximum Heat Flux, 10^6 BTU/hr-ft ²	0.750

- a. The 1.16 P/D is the reference design. P/D's of 1.12 and 1.20 will also be studied.
- b. Not including end plugs.
- c. This information does not include overpower conditions
- d. 192 Pins for rhombus cluster concepts

5.2 FUEL ASSEMBLY PRESSURE DROP

Current day pump technology limits the FFTF fuel assembly pressure drop to 100 psi.^[33] Evaluation of the three grid concepts indicate that both the honeycomb and rhombus-cluster grided assemblies yield a pressure drop within this limit while the tri-planar concept is slightly in excess of 100 psi. Using a duct flow rate of 1.44×10^5 lb/hr, which produces an average core temperature rise of 350°F,^[33] the average velocity is approximately 20 ft/sec. The pressure drop for a P/D of 1.16 is presented in Figure 5-1 for all three concepts and the wire wrap assembly, calculated using the analysis of deStordeur.^[34] As expected, if velocity is increased above the 20 ft/sec design value, the pressure drop is also increased. The rhombus-cluster and wire wrap fuel assemblies have similar pressure drops, for a given velocity, and are approximately 40 percent less than the honeycomb grid, which was selected to be the most promising concept.

The pressure drops presented, Figure 5-1, do not take advantage of the expansion of the duct flow area, from hexagonal, in the active core region, to circular, a few inches above the top of the active core. The pressure drop is based on a solid flow baffle surrounding the fuel pins for their total length. The effect of holes in the flow baffle, above the active core, is discussed in

Figure 5-1. Fuel Assembly Pressure Drop for FFTF Fuel Assembly

Section 5.2.3. The results show that the pressure drop will be reduced to approximately 60 percent of its value when ten percent of the baffle has holes in it. Thus, the pressure drops presented could be reduced, using a baffle with holes and even the tri-planar grid would be within the 100 psi limit.

Also not included in the pressure drop presented in Figure 5-1 is the effect of fuel rod swelling. This effect is not considered significant and a three percent swelling condition leads to an increase in pressure drop of eight psi for the honeycomb grid with a P/D of 1.16. This effect will be discussed in greater detail in a separate section.

The effect of decreasing the pitch to diameter ratio is expected to increase pressure drop, at a given velocity, because the equivalent diameter is smaller and grid form loss coefficients are higher. This effect is shown in Figure 5-2 for the three grid concepts. Also shown on this graph are operating lines for a given core temperature rise. Since the number of fuel pins are the same, regardless of the pitch to diameter ratio, the flow area is decreased and velocities are increased for a decrease in the pitch to diameter ratio. Thus, the operating line intersects the pressure drop line at different velocities. A small pitch to diameter fuel assembly has the hydraulic disadvantage of higher pressure drop, at a fixed velocity, as well as higher pressure drop due to a higher velocity. However, it has a nuclear advantage of a larger fuel volume fraction.

The pressure drop presented in the two previous figures includes the frictional grid form loss and gravitational pressure drop. The pressure drop due to fluid acceleration is small and was neglected. End plate losses are not included, but are estimated to be approximately 1.5 psi, and are considered small in comparison to the total fuel assembly pressure drop. The pressure drop is calculated as follows.

$$\Delta P = \left(f \frac{L'}{D} + NK \right) \frac{\rho V^2}{2g} + \rho L \quad (5-1)$$

Figure 5-2. Effect of Fuel Rod Pitch and Velocity on Fuel Assembly ΔP

where:

f = Friction factor
L' = Total ungridded fuel rod length
D = Equivalent diameter
N = Number of grids
K = Form loss coefficient per grid
 ρ = Density of sodium
V = Average fuel assembly velocity
L = Total fuel rod length
g = Gravitational constant.

The friction factor, f, is calculated by an approximation to smooth pipe Moody curve. [35]

$$f = \frac{0.184}{Re^{0.2}} \quad (5-2)$$

In the range of Reynold's numbers expected during FFTF steady state operation, this approximation is very good.

The calculation of the form loss coefficient is discussed in the following section. However, these calculations should be verified experimentally upon selection of a final design.* The prototypic fuel bundle pressure drop should also be evaluated experimentally.

5.2.1 Form Loss Coefficient

The calculation of fuel assembly pressure drop requires a value for the form loss coefficient of the grid. This section will present form loss coefficients for the honeycomb, rhombus-cluster and tri-planar grid concepts. It is usually best to evaluate the form loss experimentally. However, as no

*See Development programs.

experimental results were available for preliminary design, calculational models were used to estimate the loss coefficient. Methods of calculation of the form loss coefficient are also discussed. The form loss coefficient, K, is defined as:

$$K = \frac{\Delta P_{\text{grid}}}{\frac{\rho V^2}{2g_c}} \quad (5-3)$$

Thus, the form loss coefficient, is the ratio of the pressure drop across the grid divided by the dynamic head of the sodium in an ungridded flow channel.

Figure 5-3 illustrates the form loss coefficients, for the three grid concepts, at reference conditions. The tri-planar has a grid tube around every fuel rod and the highest form loss because it blocks the largest percentage of flow area. The rhombus-cluster concept has a grid strap around every four rods, with shear webs holding the four rods together. Thus, it has the lowest flow area contraction and a low form loss coefficient. The honeycomb grid falls between the two previously mentioned concepts. The dashed lines on this graph illustrate those pitch to diameter ratios which cannot be built. Three percent swelling, the maximum considered in this design study, causes interference between the rod and grid and makes these concepts unacceptable.

The following paragraphs will describe in more detail the form loss coefficients for each type of grid.

5.2.1.1 Honeycomb Grid Form Loss Coefficient

The honeycomb grid has a grid strap surrounding every fuel rod. This strap is of uniform thickness and is joined to an adjacent strap by grinding straps to half thicknesses and brazing. Details of this grid are shown in Figure 3-8 (Dwg 766J216).

Many parameter studies were performed for this grid. Perturbations for these studies were made about the reference conditions for the honeycomb grid which are presented in Table 5-3. These analyses included the effect of changes in

Figure 5-3. Effect of Rod Spacing on Grid Performance

pitch to diameter ratio, fuel rod OD (i.e., swelling), velocity, and grid thickness. The results of these studies are summarized in Table 5-4 in the form of a sensitivity of the grid performance to the change in a given parameter. The parameter studies used for this table will be discussed in greater detail in the following paragraphs.

TABLE 5-3
REFERENCE CONDITIONS FOR HONEYCOMB GRID

Velocity	20 ft/sec
Pitch to Diameter Ratio	1.16
Fuel Rod OD	0.250 inches
Grid Thickness	0.015 inches
Grid Length	1.125 inches

Grid form loss coefficients are best evaluated from experimental results. However, when experimental results are not available, as is the case for the honeycomb grid, they may be estimated for preliminary design purposes using analytical techniques. The theoretical model used is discussed in detail in Section 5.9. Basically, it assumes that the flow area within the grid is divided into a number of parallel subchannels. Twelve subchannels were used to analyze the honeycomb grid as shown in Figure 5-4. The velocity in each subchannel is then adjusted such that equal pressure drop occurs across each. Effects of friction, contraction, expansion and other obstructions to flow (i.e. dimples) on pressure drop are considered.

The honeycomb grid has four subchannels with "half-dimples" and one with a "complete dimple" as shown in this figure. Thus, these dimples are the "additional form loss" mentioned in Section 5.9. The value of the "additional form loss" was based on the form loss for an orifice. The following

TABLE 5-4

SENSITIVITY ANALYSIS OF THE HONEYCOMB GRID LOSS COEFFICIENT

Parameter Varied	Derivative	Value of Derivative	Change in Form Loss Coefficient for a 1% Increase in Nominal Value ^a
Clad Diameter	$\frac{\partial K}{\partial D}$ (inch ⁻¹)	120	0.3
Grid Thickness	$\frac{\partial K}{\partial t}$ (inch ⁻¹)	225	0.034
Pitch to Diameter Ratio	$\frac{\partial K}{\partial (P/D)}$ (unitless)	-25.8	-0.30
Velocity	$\frac{\partial K}{\partial V}$ ($\frac{\text{sec}}{\text{ft}}$)	-0.014	-0.003

a. Nominal values shown in Table 5-3.

Figure 5-4. Subchannel Division for Honeycomb Grid

paragraphs will discuss the value of the honeycomb grid form loss coefficient, based on the above model, as well as perturbations of important parameters.

The effect of the fuel rod pitch to diameter ratio on the grid loss coefficient can be seen to be appreciable as shown in Figure 5-5. The large increase in the grid loss-coefficient at a low pitch to diameter ratio, may be attributed to smaller flow areas within the grid and a larger contraction ratio (because grid thickness is fixed). This figure also illustrates the small effect that velocity has on grid form loss. This small effect is caused by the slight change in the friction factor with velocity. Contraction and expansion loss coefficients are assumed to remain constant. Since the friction factor decreases slightly with velocity ($f \sim V^{-0.2}$), the grid loss goes down slightly. However, total pressure drop increases drastically with velocity since it is proportional to the velocity squared.

Thickness of the hexagonal grid surrounding the fuel pin is another important parameter affecting the grid performance. Thicker grid straps produce smaller flow areas, higher velocities within the grid and the expected result of higher loss coefficients as shown in Figure 5-6.

Swelling also influences the form loss coefficient of the grid. Basically, it has been treated as an increase in the fuel rod diameter with other parameters remaining constant. As shown in Figure 5-7, the form loss coefficient increases approximately 50 percent at a $P/D = 1.16$ and three percent swelling (the maximum swelling expected in the FFTF fuel assembly). However, this does not imply that the total pressure drop across all grids will be increased this much because the maximum swelling only occurs at one location. The average swelling along the core is considerably less than three percent and no swelling occurs in the 37 inches above the top of the active core, where grids also exist.

5.2.1.2 Tri-Planar Grid Loss Coefficient

The tri-planar grid is described in Figure 3-1 (Dwg 766J211). With this grid, coolant entering the grid can flow either through the annulus between a fuel rod and its concentric grid tube, or flow through the open area between grid tubes as shown in Figure 5-8. These flow splits will occur at three different

Figure 5-5 Form Loss Coefficient for Honeycomb Grid

Figure 5-6 Effect of Grid Thickness on Form Loss for Honeycomb Grid

Figure 5-7. Effect of Swelling on Honeycomb Grid Performance

FREE STREAM FLOW AREAS

FLOW AREAS AFTER INTERCEPTING
FIRST LEVEL OF GRID TUBES FLOW AREAS AFTER INTERCEPTING
SECOND LEVEL OF GRID TUBES FLOW AREAS AFTER INTERCEPTING
THIRD LEVEL OF GRID TUBES

Figure 5-8 Triplanar Grid Flow Areas

levels within the grid. The separated streams will later recombine at junctions farther down-stream in the grid.

The tri-planar grid flow channels can be represented by the model shown schematically in Figure 5-9. The coolant entering the grid will distribute in such a way as to cause equal pressure drop in two parallel streams with common end points, such as streams 7 and 2-3-4 in Figure 5-9. This analysis is similar to a previous analysis presented in another report.^[36] The grid configuration is slightly different than the previous tri-planar grid and flow distribution is obtained by considering both friction and form loss pressure drop in this analysis compared with only friction in the previous analysis.

The overall pressure drop and flow distribution is calculated by considering the pressure loss in each section where the mass flow rate is constant. For each of these sections:

$$\Delta P = (K_{\text{form loss}} + f L/De) \frac{\rho V^2}{2g} \quad (5-4)$$

where:

- ΔP = static pressure drop from section inlet to section outlet
- g = gravitational constant
- ρ = coolant density
- V = coolant velocity
- $K_{\text{form loss}}$ = contraction + expansion form loss coefficients
- f = Moody friction factor
- L = section length
- De = section hydraulic diameter

Also, at any junction of two or more streams,

$$\sum_{i=1}^n \rho_i V_i A_i = 0 \quad (5-5)$$

STREAMS 6, 7, 8 REPRESENT FLOW
THROUGH THE ANNULUS BETWEEN A
FUEL ROD AND ITS CONCENTRIC
GRID TUBE

STREAMS 1 TO 5 REPRESENT FLOW
THROUGH THE OPEN AREA BETWEEN
GRID TUBES

Figure 5-9. Model Used To Calculate Pressure Drop
and Flow Distribution in Triplanar Grid

where:

A = stream flow area

n = the number of streams at the junction

i = subscript which refers to a particular stream

These two equations, written for each stream or junction form a set of non-linear simultaneous equations in terms of the stream velocities. To solve this set of equations, the following procedure was used:

1. An initial estimate of the stream velocities is used to calculate the $(K + f L/De)$ term.
2. Solve the set of equations for new velocities.
3. If the difference between new velocities and old velocities is negligible, stop iterating; otherwise use the new velocities and return to Step 1.

Once the correct flow distribution has been determined, the overall pressure drop can be found by calculating individual pressure drops of streams 1 through 5 in Figure 5-9 and summing up these pressure drops.

For various grid geometries, the form loss coefficient depends on pin pitch, pin diameter, grid tube wall thickness, surface roughnesses, amount of brazing material and coolant velocity. The form loss contribution, consisting of an expansion and a contraction loss coefficient is calculated from the ratio of projected flow area within a grid to the flow area outside the grid using the curves presented by Kays and London.^[37] The form loss coefficients are taken as constant for a given geometry. The friction factor, f , is the Moody friction factor calculated at the final velocities.

Using the procedure outlined, the overall pressure drop and flow distribution was calculated for several geometries of the tri-planar grid. For each geometry, the free stream velocity outside the grid was varied parametrically. Figure 5-2 shows the grid form loss as a function of grid geometry and free stream velocity. Table 5-5 presents typical values of stream velocities.

TABLE 5-5
TYPICAL TRI-PLANAR GRID STREAM VELOCITIES

<u>Stream Number^a</u>	<u>Velocity ~ Ft/Sec</u>
1	31.5
2	43.5
3	75.7
4	43.2
5	31.3
6	18.1
7	21.3
8	20.8

Pin Pitch = 0.29 inches
Pin Diameter = 0.25 inches
Tube Wall = 0.015 inches
Thickness
Free Stream = 15 Ft/Sec
Velocity

a. See Figure 5-9

5.2.1.3 Rhombus Cluster Grid Form Loss Coefficient

The rhombus cluster grid concept provides rod support without direct pressure contact on the rod surface. This is accomplished by attaching four rods into a rigid structure by means of a shear web. Each group of rods are then held in place by a grid which applies pressure to the shear web. Details of this grid are shown in Figure 3-3 (Dwg 766J213) and are discussed in more detail in Section 3.4.

From the standpoint of pressure drop, this type of grid configuration is very desirable, since the effect of the grid straps and dimples are smeared over a larger flow area thus resulting in lower overall grid loss coefficients. Parameter studies were conducted for this grid using reference conditions tabulated in Table 5-3, with the exception of grid length which was increased to 1.4 inches.

The theoretical model used in the analysis of pressure drop is essentially the same as used for the honeycomb grid and is presented in Section 5.9. Each characteristic flow area within the grid is divided into a number of parallel flow subchannels whose boundaries are determined from positions of minimum velocity. The velocity for each subchannel is then adjusted such that the pressure drop across the grid for each subchannel is a constant. The breakdown of the subchannels is shown in Figure 5-10 for a uniform P/D ratio.

After consideration of fuel rod swelling effects, it was determined that for a P/D ratio of 1.12, interference would exist at the end of life between the rod and grid strap, thus this P/D ratio was eliminated from further consideration. A similar problem of possible interference was found to exist for a P/D ratio of 1.16. However, this was circumvented by decreasing the P/D ratio of the four rod cluster to 1.12, while keeping the pitch of individual clusters the same. For a P/D ratio of 1.2, no interference was found; thus a uniform pitch was maintained.

Grid loss coefficient and fuel assembly pressure drop are presented in Figures 5-11 and 5-12 as a function of velocity. The results for both the even and uneven P/D ratio of 1.16 illustrate that both configurations are hydraulically equivalent based on the system performance limitation of

Figure 5-10. Rhombus Cluster Subchannel Designation for Pressure Drop Calculation

Figure 5-11. Rhombus Cluster Grid Loss Coefficient

Figure 5-12. Rhombus Cluster Fuel Assembly Pressure Drop

pressure drop. Figure 5-13 shows the individual subchannel velocity profiles. It may be noted that the uneven P/D ratio of 1.16 represents a slight localized hydraulic advantage in velocity. More important however, is the very notable non-uniformity of velocity within a unit cell for all P/D ratios considered. Thermally this introduces large thermal gradients across individual rows within the grid structure. The magnitude of the difference was not analyzed since other non-uniformities in temperatures were found along the bare rods, and were deemed to be more critical. This is discussed in more depth in Section 5.4.1.

5.2.2 Effect of Swelling on Pressure Drop

During burnup of the fuel rods, the mixed oxide fuel will swell due to irradiation damage. This swelling causes an increase in the cladding diameter, thus decreasing flow channel area. For a fixed mass flow rate (lb/hr), which is necessary to achieve an average core temperature rise of 350°F, the velocity of the coolant increases. Therefore, swelling will cause an increase in pressure drop.

Based on a PNL review of swelling experiments,^[38] data from Pin V171, irradiated in early fast irradiation studies by the British,^[39] was used to determine the effect of swelling on pressure drop. Values of swelling, with up to three percent peak diametral swelling, were evaluated axially along the FFTF core. The curves used are shown in Figure 5-14. They were obtained by ratioing the British data.

The pressure drop may be evaluated by the following relationship.

$$\Delta P = \left(f \frac{L}{D} + N K_{\text{grid}} \right) \frac{\rho V^2}{2g} + \rho L \quad (5-6)$$

where:

- f = friction factor
- L' = ungridded length
- L = length
- D = equivalent diameter

Figure 5-13 Relative Velocity Ratios in Grid Unit Cell

Figure 5-14. Fuel Element Swelling for FFTF Fuel Assembly

N = Number of grids
K_{grid} = grid form loss coefficient
ρ = density
V = velocity
g = gravitational constant

Because swelling is not uniform along the total length of the active core, as shown in Figure 5-14, this equation must be evaluated locally and integrated over the total core length. This was done for the honeycomb gridded concept dividing the core into numerous increments and evaluating each increment individually. Grid loss coefficients must also be evaluated locally based on the swelling data presented in Figure 5-14 and the form loss coefficients presented in Figure 5-7.

Due to swelling, the flow area is a minimum at the position of maximum swelling and increases at locations away from this position. Thus a contraction and expansion of flow area occur axially along the core due to swelling. This effect is small and can be neglected for pressure drop calculations. If treated as a diffuser, one can easily see that the rate of flow area change per unit length is very small, the "diffuser angle" is very small, and this pressure loss can be neglected.

By numerical integration, the additional pressure drop was determined for peak swelling up to three percent and pitch to diameter ratios of 1.12 to 1.20 at velocities which give an average core temperature rise of 350°F. This additional pressure drop occurs in the active core, where the fuel swells, and not in the fission gas plenum. Figure 5-15 presents these results. For a 1.16 pitch to diameter ratio and three percent maximum swelling, the increase in pressure drop will be only eight psi. This is not appreciable. However, for lower pitch to diameter ratios, the increase in pressure drop becomes more severe.

5.2.3 Effect of Holes in Flow Baffle on Fuel Assembly Pressure Drop

As previously mentioned, the pressure drops for the fuel assembly, presented in Figures 5-1 and 5-2 are based on a solid flow baffle up the total length of the core and fission gas plenum.

Figure 5-15. Effect of Swelling on Pressure Drop

This causes the velocity at the active core and fuel assembly outlets to be approximately the same. A reduction of the velocity in the region of the fission gas plenum would reduce the pressure drop while not adversely influencing coolant temperature distribution. Two advantages of reducing pressure drop are lower pumping power and a smaller pressure gradient across the flow baffle. This results in a thinner baffle and thus increases the fuel volume fraction.

As shown in Figure 3-6 (Dwg 766J226), the duct surrounding the flow baffle, expands from a hexagonal geometry in the active core area to a circular shape above the top of the active core. If one could utilize the area, between the hexagonal flow baffle and the circular duct, for coolant flow, the pressure drop in the fission gas plenum region could be reduced. This possibility exists by putting holes in the flow baffle, as shown in Figure 3-10 (Dwg 766J222).

The analysis was performed for holes in the flow baffle and reference conditions with the honeycomb grid. It is expected results will be similar for different pitch to diameter ratios and different grid configurations. The hydraulic model was based on the following assumptions:

1. No flow in the annulus between the duct and flow baffle due to the ball and cone seat located at the bottom of the flow baffle.
2. Holes were uniformly spaced in the flow baffle, starting directly above the top of the active core.
3. The duct expanded from hexagonal to circular geometry at the core outlet. It is realized that this assumption is slightly incorrect because the expansion actually occurs several inches above this location. However, it was made such that the analysis could be done quickly and easily.
4. The resistance to cross flow, from the rods to the area between the duct and baffle, is dependent upon coolant velocities in these regions.

Using a digital computer code, which solves the energy, momentum and continuity equation, results were obtained for up to ten percent opening in the flow baffle. Results are presented in Figure 5-16. If ten percent of the baffle has holes, the fuel assembly pressure drop is reduced from approximately 70 to 42 psi, a reduction of 40 percent. It is expected that this percentage reduction in pressure drop would be similar for all types of FFTF fuel assemblies if the can were perforated above the active core region.

It is interesting that the additional decrease of pressure drop is small in the range of ten percent, for increases in the can perforations beyond 10 percent. Thus, additional perforations would not appreciably reduce the pressure drop and might structurally affect the baffle performance.

The maximum pressure drop across the flow baffle is also shown in Figure 5-16. It also decreases similar to the fuel assembly pressure drop. Since the ball and cone seat is axially located at the bottom of the active core, the maximum can pressure drop also occurs at this location, which is at inlet temperature conditions.

The results indicated that flow started leaving the rodded region just above the active core. Additional coolant continued flowing into the area between the duct wall and flow baffle as one moved farther axially away from the top of the active core. Eventually, when no lateral pressure gradient existed across the flow baffle, no additional flow passed through the holes in the flow baffle.

5.3 MIXING CHARACTERISTICS

A knowledge of the mixing characteristics induced by grids has a two-fold purpose in the design of reactor fuel assemblies. 1) It provides information which is required to accurately calculate bulk coolant temperature distributions in the various subchannels of an assembly. 2) It is required for a reliable assessment of the magnitude of the enthalpy rise hot channel factor which directly affects the maximum available power that may safely be obtained from a reactor core.

Figure 5-16. Effect of Holes in Flow Baffle on Pressure Differential

A number of experimental tests have been conducted at Westinghouse [40,41,42] and elsewhere to evaluate mixing for a particular fuel assembly configuration. Some of this data has been generalized for bare rods by Rogers and Tarasuk [43] in the form of a correlation which applies to both square and triangular pitch rod arrays. This work provides a valuable tool for analyzing temperature distributions at large distances downstream from a grid. Some method however, is needed for analyzing the turbulence immediately downstream from a grid. This effect, if taken into account, can increase the average turbulence level from two to three times the value for bare rods, thereby, reducing the enthalpy rise of the hot channel.

To the author's knowledge there has not been any experimental data presented in the literature on the decay of turbulence downstream from a grid in a rod bundle, let alone any generalized models. It is the purpose of this section to present such a model which can be used in the interim until a definitive correlation is developed from actual experimental results.

The work of Frenkiel [44] has been used as a basis for the model development. The theoretical equations developed by Frenkiel are generalized to the extent that they may be applied to any type of isotropic turbulence promoters provided that the initial characteristics are known at some instant of time.

Frenkiel concluded from his analysis that grid produced turbulence in an open pipe may be separated into two parts, a filter and wake effect. The filter effect changes the spectrum of turbulence, without changing the total turbulent energy available for mixing while the wake effect increases the intensity of turbulence with a corresponding decrease in the scale of turbulence. Frenkiel's analysis was developed from the von Karman and Howarth [45] treatment of the Navier - Stokes equation given by:

$$\frac{\partial (\bar{u}'^2 R_x)}{\partial t} + 2 (\bar{u}'^2)^{3/2} \left(\frac{\partial T_h}{\partial r} + \frac{4}{r} T_h \right) = 2 \gamma \bar{u}'^2 \left(\frac{\partial^2 R_x}{\partial r^2} + \frac{4}{r} \frac{\partial R_x}{\partial r} \right) \quad (5-7)$$

where: R_x and T_h are the second and third order correlation coefficients

u', t = turbulent velocity and time

r = distance of two points between which correlation coefficient is determined.

Applying the theories of Loitsianskii,^[46] Millionshtchikov,^[47] and Kolmogoroff,^[48] Frenkiel obtained the following approximate solution for the case of large Reynold Numbers of turbulence;

$$\frac{u'^2}{u_o'^2} = (x/x_o)^{-10/7} \quad (5-8)$$

where: subscript o refers to initial value and x is distance downstream from the grid

In terms of turbulent intensity (defined as the ratio of turbulent velocity to main stream velocity) Frenkiel's approximate solution may be rewritten as

$$\frac{u'}{v} = A \left(\frac{x}{b}\right)^{-5/7} \quad (5-9)$$

where:

$$A = \frac{u_o'}{v} \left(\frac{b}{x_o}\right)^{-5/7}$$

and

b = strap or bar width which may be considered as a reference length characteristic of any grid configuration

v = main stream velocity.

Baines and Peterson^[49] were able to verify Frenkiel's decay law using air as the working fluid for seven types of grid configurations. Sparks and Hoelscher^[50] also verified the same decay law using water for four types

of grid configurations. This excellent agreement indicates that the behavior of the decay is independent of the fluid properties. Both of these sets of experimental data are shown in Figure 5-17 and are noted to correlate reasonably well by Equation (5-9) with $A = 1.12$.

In addition to a verification of Frenkiel's law of decay, the results of Baines and Peterson also provide two further points of interest.

1. The maximum turbulent intensity is greatest for those configurations which have the greatest solidity ratio. Solidity ratio in this case is defined as the fractional degree to which the grid obstructs the flow or $s = \text{area with grid} / \text{area of free stream}$.
2. The maximum turbulent intensity does not occur directly downstream from the grid, but appears to be a function of the solidity ratio. The distance is independent of the relative size of the strap as is noted in the results for strap widths of 1/3, 1, and 3 inches.

Assuming that the distance and magnitude of the maximum turbulent intensity is only a function of the solidity ratio, the results of Baines and Peterson were correlated using a digital computer code "OMNIFIT."^[51] These results, presented in Figures 5-18 and 5-19, are as follows:

$$\text{Maximum Turbulent Intensity } \frac{u'}{v} = 0.5899 (s)^{1.102} \quad (5-10)$$

$$\begin{array}{l} \text{Distance Downstream of} \\ \text{Maximum Turbulent Intensity } \frac{x}{b} = 3.634 (s)^{-1.236} \end{array} \quad (5-11)$$

Before these results can be applied to a grid within a rod bundle, several points must be considered.

1. Will the decay of grid turbulence in a rod bundle obey Frenkiel's law?
2. What effect will the rods have on the initial turbulence level and position downstream from the grid?

Figure 5-17. Variation in Intensity of Turbulence Produced by Lattice-Type Screens, Referred to Bar Size

Figure 5-18. Effect of Solidity Ratio on Location Downstream of Maximum Turbulence

Figure 5-19. Effect of Solidity Ratio on Initial Turbulence for Screens

The first point has been experimentally verified by Westinghouse for a type of grid which utilized unsymmetric mixing vanes on its downstream edge. These vanes were used to physically increase the turbulence level by deflecting flow from one subchannel to another, thereby, creating a region of non-isotropic turbulence immediately downstream from the grid. The results of the Westinghouse tests are shown in Figure 5-17, and illustrate that this region of non-isotropic turbulence decays rapidly to a condition of isotropic turbulence which obeys Frenkiel's decay law. Since the Westinghouse tests used a dye injection and sampling technique, it was not possible to measure the turbulent intensity at short distances >1.57 inches downstream from the grid, thus little can be said regarding the region of non-isotropic turbulence. It is possible that the maximum turbulent intensity shown in Figure 5-17 may have had a greater value at some location upstream. However, for the purpose of this present model, this represents a conservatism as will be shown later. The second point is not as clearly defined, however, again referring to the Westinghouse result, it is apparent that the fuel rods do affect the location of the maximum turbulent intensity. The possible explanation for this behavior is:

1. The region of non-isotropic turbulence perpetuates downstream until large scale eddies are dissipated, thus the maximum turbulent intensity will occur farther downstream in a rod bundle.
2. In the initial stage of turbulence, the rods act as a barrier or provide additional shear surface to the coalescing jets exiting from each obstructed channel thus delaying total interaction.

Since the grids for the FTR have solidity ratios less than the mixing vane grid, the turbulent intensities for these configurations should fall somewhere in between the grid data for the rod bundle and the open duct.

Some idea of how the fuel rod could affect the initial turbulence of the grid, may be obtained by comparing the free stream turbulence of a pipe and a rod bundle. For this purpose the work of Rogers and Tarasuk^[43] is used for rod bundles and the work of Martin^[52] is used for pipes.

Rogers and Tarasuk developed a generalized correlation using experimental data and a macroscopic analytical model based on subchannel average conditions. Their results for turbulent intensity are shown in Figure 5-20 for a range of P/D ratios from 1.12 to 1.24 and a Reynolds Number range from 10^4 to 10^5 .

Martin's results for turbulent intensity are shown in Figure 5-21 as a function of Reynolds number. Also shown are the results of three other experimental sources. [53,54,55,56]

The magnitude of the turbulent intensity for the open pipe is seen to be (in the Reynolds Number range of interest) about two to three times the value for a rod bundle. This same behavior has been assumed to occur at the grid within a rod bundle in the same ratio as the free stream results, thus the initial turbulence is defined as:

$$\frac{u_o'}{v} = \left[\frac{u' / v \mid \text{rod bundle}}{u' / v \mid \text{pipe}} \right]_{N_{RE}} \times 0.5899(S)^{1.102} \quad (5-12)$$

where

$$\frac{u'}{v} \mid \text{rod bundle} = \frac{0.0503(P/d-1)^{-157} \left(1 + \left(\frac{de_j}{de_i}\right)^{1.133}\right) \frac{de_i}{d}}{2 N_{RE}^{0.32}}$$

and

$$\frac{u'}{v} \mid \text{pipe} = 0.2415 (N_{RE})^{-0.189}$$

p, d = rod pitch and diameter

N_{RE} = Reynolds Number

de = Subchannel equivalent diameter, subscripts i and j indicate two adjacent subchannels

Figure 5-20 Free Stream Turbulence in Rod Bundles

Figure 5 21 Relative Intensity as a Function of Reynolds Number

Since the decay law of Frenkiel has been shown to apply to grids within rod bundles (at least in the isotropic region), the following form of the decay law is proposed:

$$\frac{u'}{v} = C \left(\frac{x}{x_0} \right)^{-0.691} \quad (5-13)$$

where

$$C = \frac{u'_0}{v}$$

The term x_0 represents the point of maximum turbulent intensity which for this present evaluation has been assumed to be the same as for the Westinghouse grid, thus

$$\frac{u'}{v} = C \left(\frac{x}{1.75} \right)^{-0.691} \quad (5-14)$$

It is recognized that all the assumptions made to develop this model must be justified experimentally,* however, until this information becomes available this model can provide interim design characteristics.

Applying Equations (5-12) and (5-14) for the FTR honeycomb grid concept, results in the turbulent intensity characteristics as shown in Figure 5-22.

As discussed in Section 5.4, very little difference is noted in the temperature distributions when an average turbulent intensity is used in place of an axial varying distribution, thus also shown in Figure 5-22 are the average values for the three P/D ratios investigated. In determining the average value it has been assumed that the maximum turbulent intensity decreases

*See development program

Figure 5-22 Flow Interchange Velocity Ratio Downstream from a Grid

linearly to zero at the grid. Based on Baines and Peterson's results, this should represent a design conservatism. The average turbulent intensity for the wire wrap is seen to vary from 1.5 to 3 times that for the grid thus illustrating the effectiveness of wire wrap in promoting mixing.

5.4 COLLANT TEMPERATURE DISTRIBUTION

Coolant temperatures were determined within the subassembly for the rhombus-cluster and honeycomb grid concepts. It is expected that coolant temperatures for the triplanar grid would be similar to the honeycomb grid concept. No analysis was performed for the triplanar grid because by this time in the study, the honeycomb concept was selected as the most promising.

Temperatures were determined for the hottest fuel assembly.^[57] The conditions for this fuel assembly are:

$$\frac{\text{Peak Power Within Assembly}}{\text{Average Power Within Assembly}} = 1.17 \quad (5-15)$$

$$\frac{\text{Hot Assembly Power}}{\text{Avg. Assembly Power}} = 1.11 \quad (5-16)$$

These conditions yield a nuclear radial factor, F_R^N , equal to 1.3. This condition of power distribution is expected to give the most severe temperature gradient across the fuel assembly. The average linear power rating of 7.85 Kw/ft was used when no hot channel factors were applied. The coolant flow rate was consistent with a 350°F increase in coolant temperature across the core with an average power fuel assembly.

Using a digital computer program, which solves the energy, momentum and continuity equations simultaneously, the temperature and flow distributions within the fuel assembly were determined. Effects of conduction and mixing, due to turbulence, were also included. Analysis indicated that mixing, which varied axially along the fuel assembly, gave almost the same results as the axial average of the turbulent intensity. This variation occurs, due to the grid, which promotes turbulence and decays at axial distances downstream from the grid.

Analysis of temperature distribution with the honeycomb grid indicated that severe temperature gradients exist across the outer row of rods due to the large quantity of coolant flowing between these rods and the duct wall. In a wire-wrap fuel assembly, the gradients are similar to a gridded concept. It was found that this temperature gradient could be substantially reduced by the insertion of a "flow baffle" in this area. This effect is discussed in the section on the honeycomb grid. The addition of a flow baffle for the rhombus-cluster design is not discussed. However, it is expected that a flow baffle would also cause improvements.

5.4.1 Rhombus Cluster Temperature Distributions

Coolant temperature distributions were computed for the rhombus-cluster grid concept using a computer code which solves the mass, energy and momentum equations simultaneously.

Shown in Figure 5-23 is the subchannel breakdown and power distribution which was used in the analysis. The subchannel breakdown was selected on the following basis:

1. Symmetry was utilized to provide sufficient detail of the actual fuel assembly temperature distribution within the limits of the computer code (a maximum of 36 subchannels). This was not entirely possible to achieve since the fuel assembly is made up of both square and triangular subchannels. The section selected does, however, represent the most severe condition which was anticipated to occur for the radial power distribution shown in Figure 5-23.
2. The effect of the square subchannels was assessed by considering these subchannels separately.
3. The effect of the duct walls was assessed by a finer division of the subchannels near the wall.

Figure 5-23. Subchannels and Power Distribution for Analysis of Coolant Flow and Temperature for FTR Fuel Assembly using Rhombus Cluster Grid Concept

The results obtained for the uneven P/D ratio of 1.16 are shown in Figure 5-24 for two types of square subchannels. The large square subchannels have the same pitch as the triangular subchannels while the small square subchannels are reduced to a minimum pitch consistent with mechanical restraints of grid straps and swelling. Both types of subchannels are seen to produce large thermal gradients across the central row of rods $\sim 120^{\circ}\text{F}$ (large subchannels) and $\sim 90^{\circ}\text{F}$ (small subchannels). The effect of the uneven pitch can be seen by referring to the results for subchannel 2 which has a peak temperature of 1075°F and subchannel 6 which has a peak temperature of 1025°F . The thermal gradient produced in other subchannels is not seen in the figure because the effect is smeared due to the method of analysis. However, it is expected to be the same as subchannels 2 and 6.

To provide a more detailed picture of the temperature characteristics across the fuel assembly, the results are presented in a slightly different form in Figures 5-25 and 5-26 for P/D ratios of 1.16 (uneven) and 1.2. Only the results for the small square subchannels are shown, since nothing additional can be obtained from the results for the large subchannels. It is noted that even for a P/D ratio of 1.20, the square subchannels adversely affect the thermal characteristics of the fuel assembly. Thus for the basic rhombus-cluster configuration, severe thermal gradients resulting in high thermal stresses seriously limit the design feasibility.

Some idea of what square subchannel rod spacing would be required in order to obtain desirable temperature characteristics can be obtained by referring to Figure 5-27. It is apparent that for the small P/D ratios being considered in the current study, rod spacings less than the thickness of a grid strap would be required. Thus some other method of baffling the flow is required, which adds to the design complexity of this configuration.

Severe thermal gradients are also noted for those subchannels which border a duct wall, however this problem can be rectified by the insertion of a flow baffle as was done with the honeycomb configuration.

Figure 5-24. Coolant Temperatures in Rhombus Cluster Fuel Assembly

Figure 5-25. Temperature Distribution of Coolant Flow for FTR Fuel Assembly using Rhombus-Cluster Grid Concept

Figure 5-26. Temperature Distribution of Coolant Flow for FTR Fuel Assembly Using Rhombus-Cluster Grid Concept

Figure 5-27. Effect of Square Channel Geometry on Enthalpy Rise

5.4.2 Honeycomb Grid Coolant Temperature Distribution

Many analyses were performed to determine the coolant temperature distributions in a honeycomb grid concept. The basic method of attack used a digital computer program which solves the energy, momentum and continuity equations simultaneously. This code uses a subchannel analysis; the subchannels and the relative power distribution are presented in Figure 5-28. Thirty-six subchannels were used (due to a limitation of the code) and they were finely divided near the outer row of rods, where steep temperature gradients could be expected to occur. Only one-half of a 217 pin fuel assembly was analyzed because of symmetry.

In order to evaluate the temperature distribution, it is necessary to determine the effect of mixing on temperature distribution. Figure 5-29 represents the curve used for the axial distribution of turbulent intensity. It should be noted that this curve is slightly different than the recommended values discussed in Section 5.3. This difference arises because this curve was generated before the mixing analysis was completed. This curve was used to generate the coolant temperature distribution. It is felt that this difference is of little consequence since it and the results in Section 5.3 are fairly close.

Using this mixing curve, coolant temperatures were generated for two conditions. The first case assumed the fuel assembly was within the duct as shown in Figure 3-7 (dwg 766J212). Figure 5-30 shows the results for no mixing and variable mixing (variable mixing will later be shown to be equivalent to three percent mixing). Since very large temperature gradients existed, it was also decided to look at coolant temperatures using a flow baffle between the outer row of rods and the duct. Details of the flow baffle may be found on Figure 3-8 (dwg 766J216). As shown in Figure 5-30, a flow baffle reduces the absolute value of maximum temperature as well as the peak temperature gradients. The effect of mixing is also shown on this curve; the recommended value of mixing does reduce the temperature gradients below the condition of no mixing and only conduction. This curve is for conditions at the top of the active fuel, where both temperatures and gradients will be at their maximums values.

Figure 5-28. Subchannels and Power Distribution for Analysis of Coolant Flow and Temperature for FFTF Fuel Assembly

Figure 5-29 Flow Interchange Velocity Ratio Downstream from Mixing Grid

Figure 5-30 Effect of Flow Baffle on Coolant Temperatures at Core Outlet

The maximum temperature gradient across one row of rods is illustrated in Figure 5-31. It shows that for any given amount of mixing, the maximum gradient across a row of rods without mixing is considerably larger than the case with a flow baffle. Even with 15 percent mixing (this might possibly be obtained by inserting mixing promoters at the expense of pressure drop), the gradient is still more severe than the condition with variable or three percent mixing and a flow baffle. This figure also shows how results for variable mixing closely agree with the average axial mixing.

The axial temperature profile for the two hottest subchannels is shown in Figure 5-32. No mixing is included in these results. However, Figure 5-33 illustrates the effect of mixing on temperatures for these two subchannels. This figure demonstrates how much mixing decreases the temperature in these two channels. As expected, the greater the mixing is, the larger the decrease in the temperature of these two channels. It is very interesting to inspect the curves with three percent mixing and variable mixing. Both yield basically the same results, except the variable mixing curve is not smooth. This is because more heat transfer due to mixing occurs near the grids, where the turbulent intensity is greater in magnitude than farther downstream.

The effect of pitch to diameter ratio on coolant temperature was also evaluated and results are shown in Figure 5-34. These results indicate that pitch to diameter ratio does not appreciably affect the coolant temperature distribution, or the severity of the gradients. The condition of a tighter pitch is slightly more severe. These results are presented for the honeycomb grid concept with a flow baffle. However, if no flow baffle did exist, it is believed that similar trends would exist (i.e., no appreciable change in coolant temperatures for different pitch to diameter ratios).

Swelling can also affect the coolant flow distribution. Because the fuel pin remains on two fixed dimples, swelling tends to move all the fuel pins in one direction as well as decreasing the pitch to diameter ratio. Basically, swelling can move the hexagon drawn through the outer row of fuel rods. Two extreme conditions which were evaluated are shown in Figure 5-35. These conditions are that the fuel assembly would partially close the outer coolant channel at the hottest part of the fuel assembly (Case 2) or the lowest power

Figure 5-31 Effect of Mixing on Maximum Temperature Gradient Across a Row of Rods

Figure 5-32 Axial Temperature Distribution of Hottest Subchannel

Figure 5-33. Effect of Mixing on Coolant Temperature

Figure 5-34 Effect of P/D on Coolant Temperatures

Figure 5-35. Position of Fuel Rods within Flow Baffle

portion of the fuel assembly (Case 1). These represent extremes, based on the configuration of the fixed dimples in the grid. The results of this analysis, for a flow baffle, indicated swelling does not adversely affect the bulk coolant temperature distribution. These results are presented in Figure 5-36. If swelling caused the flow channels near the hottest part of the fuel assembly (Case 2) to close, the maximum coolant temperature would be increased only 15°F above the condition of no swelling. If swelling caused movement in the opposite direction, temperatures would not be increased. It should also be noted that these results are probably extremely conservative. The model assumed uniform swelling of three percent for the total axial length of the active core. This is unrealistic as shown in Figure 5-36 and would predict higher coolant temperatures than the actual condition.

5.5 CIRCUMFERENTIAL TEMPERATURE VARIATIONS

In a liquid metal cooled reactor, a variation of temperature will exist around fuel rods. This variation may be severe enough to limit reactor performance. This circumferential temperature distribution exists around "bare rods" (i.e., rods not in the vicinity of the grid) as well as in grid regions. Analytical results are presented but should be verified by experiments.* The analyses were performed for the hot spot, operating at a linear power of 14.4 Kw/ft. This linear power is based on PNL hot channel factors (see Table 5-2) and not the WARD hot channel factors presented in Table 5-7. Table 5-6 indicates the magnitude of the maximum temperature increase around the clad for the conditions analyzed. The conditions which caused these increases are discussed in the following sections. Circumferential variations in the vicinity of grids for the honeycomb and triplanar grids are presented; no work was done on the rhombus-cluster temperature variations.

*See development programs

Figure 5-36. Effect of Swelling on Coolant Temperatures

TABLE 5-6

INCREASE IN CLAD TEMPERATURE DUE TO
CIRCUMFERENTIAL TEMPERATURE VARIATIONS

Condition	Increase in Temperature
Bare Rods ^a - P/D = 1.12	23°F
Bare Rods ^a - P/D = 1.16	15°F
Bare Rods ^a - P/D = 1.20	10°F
Honeycomb Grid-Swelled Condition - P/D = 1.16	50°F
Tri-Planar Grid - P/D = 1.16	240°F

a. Based on Axford's laminar flow analysis

5.5.1 Bare Rod

At close pitch to diameter ratios, the velocity in the vicinity where the rods are closest is lower than the average. This effect, coupled with the sodium's physical properties leads to circumferential temperature variations around a fuel tube in a nuclear reactor. These variations have been investigated analytically by Axford^[58] and Dwyer.^[59] Subbotin et al^[60,61] have presented experimental evidence of this phenomenon. All of the above investigators have looked at tight lattice spacings with a triangular pitch and a central cell. A central cell is defined as a coolant channel surrounded by identical coolant channels (i.e., the coolant channel in the center of a large fuel assembly).

The results of these investigators are presented in Figure 5-37 for pitch to diameter ratios ranging from 1.0 to 1.2. All the results of the independent investigators appear to be in good agreement. The following paragraphs will discuss the individual investigators results.

5.5.1.1 Axford^[58]

Axford presented an exact solution of the energy equation. His method coupled this equation in coolant, clad and fuel regions with a triangular lattice. No gap resistance was considered. Velocity profiles were assumed for both laminar and turbulent flow. The laminar profile was calculated and allowed to vary with angle and distance from the rod surface. The turbulent flow solution considered a slug flow (uniform velocity) model. Slug flow would most likely occur at extremely high Reynolds Number. The transfer of heat due to turbulence (eddy diffusivity) was not considered. However, Dwyer states that eddy diffusivity does not appreciably effect the circumferential temperature variations. The results for laminar and slug flow should bracket the limits of the variation expected and are shown in Figure 5-37. These results were calculated for the maximum linear power using the digital computer program AXTHERM.^[62] Actual circumferential temperatures are plotted for slug flow in Figure 5-38. As shown, the change from maximum to minimum temperature occurs every 30 degrees. Thus, peak temperatures occur six times around the rod circumference.

Figure 5-37. Circumferential Temperature Distribution for FFTF Fuel Rod

Figure 5-38 Circumferential Temperature Variation Around a Fuel Rod

5.5.1.2 Dwyer^[59]

Dwyer also developed an analytical method to predict circumferential temperature distribution around a fuel rod. With the assumption of uniform heat flux (Axford assumed more realistically that there was uniform heat generation within the fuel and the solution of the energy equation yielded the heat flux), the energy equation was solved using a velocity profile calculated by a complex technique. Effects of eddy diffusivity were included. Because Dwyer's technique is difficult to use, no additional points were calculated. Instead, data presented by him for pitch to diameter ratios of 1.10 and 1.20 were used and are plotted on Figure 5-37.

5.5.1.3 Subbotin et al^[60,61]

Subbotin has experimentally investigated heat transfer in liquid metals. Circumferential variations of temperature around the cladding have been correlated for a pitch to diameter ratio of 1.04.

$$\frac{T_W^{\text{Max}} - T_W^{\text{Min}}}{q''R} K_f = 1.5 \text{ Pe}^{-0.32} \quad (5-17)$$

for $20 < \text{Pe} < 300$

where

T_W = wall temperature

q'' = heat flux

R = tube radius

K_f = coolant thermal conductivity

Pe = Peclet number

His results indicated that a fin (which is used to maintain rod spacing and is similar to wire wrap) only caused a 5 to 10 percent increase in the temperature variation. Thus, even though the above correlation is for a finned tube, it can also be used for a bare tube.

Experiments were also done by Subbotin for rods that were touching ($P/D = 1.0$). The temperature variation can be represented by the following relationship.

$$\left(\frac{T_W^{\text{Max}} - T_W^{\text{Avg}}}{q''R} \right) (\lambda_f) \left(\frac{\lambda_c}{\lambda_f} \right) = 0.18 \quad (5-18)$$

All symbols have been previously defined except

$$\lambda_c = \text{clad thermal conductivity}$$

Subbotin's experimental data are out of the pitch to diameter range of interest in FFTF. However, they do illustrate that Axford's analytical approach predicts results sufficiently close to the experimental data. Thus, one has some confidence when Axford's results are used in the 1.12 to 1.20 pitch to diameter range. Dwyer's work gives added confidence, even though it is analytical, because of its good agreement using an independent approach.

The upper limit of the circumferential temperature variation can be estimated using the laminar flow condition. The slug flow condition appears to give the minimum temperature increase and is probably closer to reality than the laminar flow condition. However, the laminar flow solution puts an upper bound on results and is the temperature presented in Table 5-6.

5.5.2 Effect of Swelling on Honeycomb Grid Temperature Distribution

When the fuel rod swells, with an increase up to three percent of its original dimension, the pitch to diameter ratio will decrease from 1.16 to 1.13. This will cause an increase in circumferential temperature variation around a bare rod from 15°F to 20°F. The honeycomb grid is designed such that all fuel rods swell in the same direction, causing the pitch to diameter ratio to remain constant and all central subchannels to be equal in area.

However, the swelling does cause one "non-uniformity" in the vicinity of the grid. Since the grid contains two fixed dimples and a spring finger, which may move, the flow areas within the grid will become non-uniform with swelling.

The largest flow area within a given area occurs between the fixed dimples and the smallest area occurs in the vicinity of the spring. This causes a decrease in velocity and mass flow rate within the small flow area and an increase in the large area. Thus, swelling within the grid will cause non-uniform coolant temperature rises around the fuel rod.

An analytical model, which treats each of these subchannels individually is incorrect because it neglects conduction across the grid strap; this type of analysis would lead to a very severe temperature variation around the rod. Therefore, a model was used which did include the conduction across the grid and into adjacent sub-channels. The velocity distribution was obtained by dividing the flow area of the grid into twelve subchannels and using the method described in Section 5.9. Then, the digital computer program, MANTA,^[63] was used to solve this problem. Figure 5-39 illustrates the mesh used to analyze the problem.

It is interesting to note that the minimum area channel always occurs adjacent to a maximum area channel and is separated only by a 0.015 inch grid strap. Thus, the larger channel conducts heat away from the smaller channel.

Results indicate that the most severe circumferential temperature gradient occurs at the outlet of the grid. Figure 5-40 illustrates results for a rod operating at 14.4 Kw/ft with three percent swelling. The inlet temperature to the 1.125-inch grid was assumed to be 750°F. The difference between the maximum and minimum temperature of the coolant and cladding are 37°F and 52°F, respectively, based on the above set of conditions. The temperature gradients will be less severe at lower swelling rates. The magnitude of the temperature gradient can also be reduced by placing the fuel rod eccentric prior to swelling. This could be done by changing the size of the dimples. Then, as the rod swelled, it would become concentric and then eccentric again but this eccentricity would not be as severe.

It is assumed that after the sodium leaves the grid, the circumferential coolant temperature distribution will be dissipated and will not appreciably affect "bare rod" clad temperatures. This dissipation occurs because a small coolant channel is adjacent to a large channel; after leaving the grid area,

Figure 5-39. Mesh Used to Study Three Percent Swelling Condition at Grid

Figure 5-40. Circumferential Temperature Variation Caused by Swelling

these channels have no physical barrier (i.e., a grid strap) and mix rapidly. The other channels are also arranged such that a smaller channel is next to a large channel, even though they are not the smallest and largest subchannels.

5.5.3 Tri-Planar Grid Temperature Distribution

In the tri-planar grid, the velocities in the annulus between the fuel rod and grid tube were found to be quite low for several combinations of grid geometry and free stream velocity. These low velocities occurred at tight pitches or with a swollen fuel rod, or both, and low free stream velocities. Such low annular velocities along with the small annular flow area, result in very low mass flow rates through these annuli. If the annular coolant cannot transmit the rod heat output to the main coolant flow outside the tubes, the heat must be absorbed by the annular stream. The high heat output along with low flow rate could result in excessive annular coolant temperature rises.

Heat transfer to the main coolant stream is discouraged by the presence of brazing material at the points of contact of grid tubes. Heat being conducted through the grid tube wall must travel a longer path either circumferentially around the tube wall to a non-brazed area or through the brazing material and then out to the main coolant stream.

In order to determine the temperature distribution in the grid, the MANTA^[63] computer code was used. This code considers the combined effects of convection and heat conduction. The thermal network used for this study is shown in Figure 5-41.

Film coefficients for the main coolant stream outside the grid tubes were calculated by Subbotin's^[64] correlation for liquid metals flowing in round tubes:

$$Nu = 5.0 + 0.025 Pe^{.8} \quad (5-19)$$

Figure 5-41. Triplanar Grid Thermal Network

where:

Nu = Nusselt number

Pe = Peclet number

Since both turbulent and laminar flow could be experienced in the annulus, the film coefficient resulting from both flow conditions was considered.

Dwyer^[65] analyzed the case of laminar flow in concentric annuli. He gave the Nusselt number, Nu, at the inner and outer radii of the annulus as a function of the ratio of outer to inner radius. The Nusselt numbers for the tri-planar grid annuli will depend on grid and fuel rod dimensions but not on annular stream velocities.

In the turbulent regime, Dwyer and Tu^[66] have correlated the results of their investigations by semiempirical equations of the form:

$$Nu = \alpha + \beta (\psi Pe)^\gamma \quad (5-20)$$

where:

α , β and γ are functions of outer radius/inner radius

and

$$\psi = \frac{1.82}{1 - \frac{\text{Pr}(E_M/\nu)^{1.4}}{\text{Max}}} \quad (5-21)$$

where:

Pr = Prandtl number

E_M = eddy diffusivity of momentum transfer

ν = kinematic viscosity subscript Max refers to the maximum value in the channel cross section.

Film coefficients were calculated as outlined above and these, along with the thermal properties of the fuel rod and grid tube were used as input to the MANTA code. Figures 5-42 and 5-43 show the clad and grid tube temperatures for a $P/D = 1.16$, 3 percent fuel rod swelling and 20 ft/sec free stream velocity.

In addition, the effective film coefficient at the braze was assumed to be one percent of the free stream film coefficient. This "effective film coefficient" is the thermal resistance due to the braze. The circumferential temperature variation in the fuel rod cladding and in the grid tube shows a difference of about 240°F from minimum to maximum in an angle of 60 degrees around the circumference. This increase severely penalizes the tri-planar concept.

5.6 THERMOFLUIDS PERFORMANCE WITH THERMAL BOW

Bowing of the fuel element will occur in a fast reactor due to initial fabrication, manufacture, and/or thermal gradients. This fuel rod bowing will cause changes in the coolant and cladding temperatures. Generally, the smaller flow channel (i.e., the subchannel into which the fuel rod has bowed) will get hotter and the larger channel will have a lower temperature.

Two different analyses were performed to study this phenomenon for the honeycomb gridded fuel assembly. The first of these studies included iteration between the mechanical design and thermal and hydraulic calculation. The procedure used was basically as follows:

1. Temperatures were calculated for nominal conditions.
2. Closing and opening of subchannels, due to bowing, was calculated using the temperature distribution.
3. New temperatures were determined from the new dimensions.
4. Steps 2 and 3 were repeated until the iteration converged.

Unfortunately, time did not allow us to finish this procedure until it converged. Two iterations were completed and it appeared that convergence was close. Fuel assemblies with and without a flow baffle were analyzed.

Figure 5-42 Fuel Cladding Surface Temperature

Figure 5-43. Grid Tube Temperature for Triplanar Grid

The second approach to the bowing problem did not include any mechanical input. A given amount of bow was assumed and circumferential coolant and clad temperature was determined. Both of these approaches are discussed in the following sections of this document.

5.6.1 Thermofluids and Mechanical Considerations to the Bowing Problem

The analysis of thermal bow is extremely complicated and truly requires both mechanical and thermofluid interaction. This interaction is required because the bowing of a rod changes the temperature distribution and a change in temperature distribution causes a mechanical response of the rod. Rather than directly couple both the mechanical and thermofluids equation, it was decided to iterate between both sets of equations. The method used has been previously presented in Section 5.6.

The physical condition chosen to represent the fuel assembly is considered to be very severe. The swelling was assumed to be three percent axially along the total length of the fuel assembly. The fuel assembly was assumed to have a flow baffle. It was also assumed that the cold bow between the first two rows was in the direction that the flow channel received a maximum decrease in flow area. The next flow channel received a maximum increase in flow area due to the initial bow. This pattern was repeated across the complete fuel assembly.

The coolant temperatures were determined from a code which simultaneously solves the momentum, energy and continuity equations. Results are shown in Figure 5-44 and are based on the first mechanical design calculations, which indicated a maximum gap closing of 6.3 mils for a fuel assembly with a flow baffle. As shown, the temperatures at the core outlet vary from row to row of rods. One coolant subchannel is cold and the two adjacent subchannels are hotter. At the fuel assembly outlet, mixing and conduction tends to dampen out this effect and the temperature profile seems much smoother.

Figure 5-45 illustrates the coolant temperatures determined from the results of the first mechanical iteration in a fuel assembly with no flow baffle. The maximum gap closing used to calculate these temperatures is 12.3 mils compared with 6.3 mils for a fuel assembly with a flow baffle. As expected,

Figure 5-44. Effect of Bowing on Coolant Temperatures with Flow Baffle

Figure 5-45 Effect of Bowing on Coolant Temperatures without Flow Baffle

the thermal gradients across the outer row of rods are more severe when no flow baffle exists; the maximum coolant temperatures are also greater for this condition. However, gradients away from this region are also more severe when no flow baffle exists; the maximum coolant temperatures are also greater for this. The maximum temperature gradient is approximately 80 percent greater when the flow baffle is removed. Unfortunately, time prohibited any additional work in these thermo-fluids-mechanical iterations.

5.6.2 Bowing Effects on Coolant and Clad Temperatures

The temperature distribution around a fuel element was analytically investigated by Woodrow.^[67] This model does not include the effects of adjacent coolant channels, mixing, and conduction. Thus, it overestimates the temperature distribution and is judged to be conservative. This analysis presents the effect of a given amount of bow on temperatures; no attempt is made to determine how this affects the mechanical dimensions.

Results of the analysis are presented in Figure 5-46 for a linear power of 14.4 Kw/ft, the hot spot condition. For a bow of 5 mils, with the honeycomb grid spaced 9.7 inches apart, the cladding and coolant temperatures are increased approximately 100°F above the condition of no bow. Both cladding and coolant temperatures are close in value because the coolant is a liquid metal, which causes a high film coefficient and low film drop temperatures. Mechanical calculations indicate that the maximum gap closing will be approximately 7.5 mils with a flow baffle, thus causing an equivalent bow of two adjacent channels of 3.8 mils. However, this gap closing is sinusoidal in nature (the gap closing at the grids is non-existent) and the average bow along an axial length between grids is about 2.5 mils. This causes an increase in temperature, compared to nominal conditions of 45°F.

Several assumptions made in this analysis are:

1. The displacement of the fuel rod, δ , is small compared to the average distance between rods.
2. The flow area around one rod can be approximated by an annulus, with inner radius a , and a mean annular thickness of δ_o .

Figure 5-46. Effect of Bowing on Coolant and Clad Temperatures

Thus, the local clearance is $\delta_o (1 + C \cos \theta)$ where C is the ratio of the eccentricity to the mean clearance, δ/δ_o . The model is shown in Figure 5-47.

Treating the flow through the eccentric annulus as a set of parallel channels, the following relationship can be derived when the friction factor is approximated by $f = 0.184/Re^{0.2}$ and the pressure drop across each of the "eccentric parallel channels" is assumed to be equal. The velocity, v , is proportional to the equivalent diameter to the two-thirds ($De^{2/3}$) power. Thus, a first order approximation yields

$$V = \bar{V} \left(1 + \frac{2}{3} C \cos \theta \right) \quad (5-22)$$

Using the heat transfer coefficient presented by Subbotin^[68]

$$\frac{h De}{K} = 0.58 \left(\frac{De}{D_o} \right)^{0.55} Pe^{0.45} \quad (5-23)$$

where:

- h = film coefficient
- De = equivalent diameter
- K = thermal conductivity
- D_o = fuel rod diameter
- Pe = Peclet number

For a given coolant, h varies as $V^{0.45}$. Thus, the heat transfer coefficient varies as

$$De^{0.45} \times \frac{2}{3} = De^{0.3} \quad (5-24)$$

The first order approximation yields an equation for the local heat transfer coefficient.

$$h = \bar{h} (1 + 0.3 C \cos \theta) \quad (5-25)$$

UNBOWED CONDITION

BOWED CONDITION

Figure 5-47 Model Used For Bowing Analysis

In calculating the coolant heat balance, the product of velocity and area gives a flow rate. For a rod displaced δ , the mass flow is

$$\dot{M} = \rho A V = \rho a d \theta \delta_o (1 + C \cos \theta) \bar{V} (1 + \frac{2}{3} C \cos \theta) \quad (5-26)$$

Neglecting mixing and conduction, the heat balance using a first order approximation, yields the following:

$$\dot{M} C_p \frac{\partial T}{\partial Z} = q'' a d \theta \quad (5-27)$$

$$\frac{\partial T}{\partial Z} = \frac{q''}{\rho C_p \bar{V} \delta_o (1 + \frac{2}{3} C \cos \theta)} \quad (5-28)$$

This is the same equation as derived by Woodrow except that it was assumed that the bow was uniform for the complete axial span, rather than having a slope (i.e., this analysis assumes $C = \text{constant}$ and $\frac{\partial T}{\partial Z} = 0$). The results of this analysis have been presented in Figure 5-47. Because of assumptions made for this model, the results are considered to be conservative.

5.7 HOT CHANNEL FACTORS FOR HONEYCOMB GRID

Any deviation from nominal conditions must be accounted for in any system design to ensure system safety. The problem becomes more critical in a reactor system because of the number of variables which can influence the design and how these variables may be combined so as to still have a reliable design.

Two basic approaches to hot channel factors are currently being used by reactor designers. The first approach, presently used at Westinghouse Advanced Reactors Division, was first proposed by Le Tourneau and Grimble^[69] and further expanded by Chelemer and Tong.^[70] Separate factors are defined for each of the three important thermal parameters of heat flux, enthalpy rise, and film temperature drop. Each of these factors, consisting of both nuclear

and engineering contributions, include effects which account for deviations of a particular parameter from the ideal case which is commonly referred to as the average channel.

The second approach at times referred to as a statistical method^[71] of analysis, utilizes some of the same basic individual factors as used in the first approach, but applies these factors to the individual ΔT 's which make up the overall ΔT of interest. The term "statistical" however can be misleading when referring to these two approaches since the first approach can also be thought of in these same terms. A more fitting name for this second method might be the "temperature drop" approach.

For the two approaches considered above, two methods are commonly used for combining the individual factors: 1) worst possible value 2) statistical values. Both methods have their shortcomings, however, if strictly applied. For example, the worst possible value method results in a very conservative or pessimistic design since the stochastic character of some of the variables is not considered. At the opposite extreme is the statistical method which considers all variables to have a distinct frequency and distribution function. This is not true for factors such as flow and flux distributions or any other factor which is either a correction factor or a factor of a more systematic nature. Thus this method may result in a design which is not necessarily optimistic but a design which is unrealistic and difficult to evaluate.

The first approach will be considered in more detail in the following paragraphs.

The definition of the three important thermal parameters are as follows:

$$F_q^T = \frac{Q_{\max}}{Q_{\text{avg}}} \quad (\text{Heat flux hot channel factor}) \quad (5-29)$$

$$F_{\Delta H}^T = \frac{\Delta H_{\max}}{\Delta H_{\text{avg}}} \quad (\text{Enthalpy rise hot channel factor}) \quad (5-30)$$

$$F_{\theta}^T = \frac{\theta_{\max}}{\theta_{\text{avg}}} \quad (\text{Film temperature drop hot spot factor}) \quad (5-31)$$

In terms of engineering and nuclear factors where superscript N refers to nuclear contributions and superscript E to engineering contributions, those factors are further defined as

$$F_{\Delta H}^T = F_R^N F_{\Delta H}^E \quad (5-32)$$

$$F_q^T = F_R^N F_Z^N F_q^E \quad (5-33)$$

$$F_{\theta}^T = F_{\theta}^E F_R^N F_Z^N \quad (5-34)$$

where: F_R^N and F_Z^N are the nuclear radial and axial factors, respectively.

The first factor considers all adverse effects which occur in a single coolant channel, the second considers effects that occur at a given axial position, while the last factor applies to single point location on the clad.

A brief description of each of these factors will be presented with major emphasis placed on those subfactors which represent a major contribution to the total. The other subfactors tabulated have been generated from Westinghouse experience on other reactor systems or obtained from PNL documents. It is recognized that for the final design, these factors must be verified by actual manufacturing and experimental test data.

5.7.1 Engineering Heat Flux Hot Channel Factor

The individual subfactors which influence the heat flux are: 1) fuel and clad dimensions, 2) fuel composition (density and enrichment), 3) power measurement and control,^[33] and 4) uncertainties in flux and burnup. Items 1 and 2 are due to manufacturing tolerances and have been found to behave

statistically; thus, these subfactors are evaluated and combined in a statistical manner. Item 3 is also considered to be of a statistical nature since it involves inaccuracies in instrument readings and human error potential. The last item represents a systematic error which is attributed to uncertainties in the nuclear analysis.

5.7.2 Engineering Enthalpy Rise Hot Channel Factor

The individual subfactors described in the preceding section also have an influence on the enthalpy rise. However, each subfactor represents an integrated affect along the length of the fuel column.

Flow maldistribution effects must also be considered and include the following:

1) inlet flow distribution per fuel tube, 2) flow redistribution within a fuel assembly, 3) outlet back pressure and outlet nozzle locations, 4) flow mixing which represents a reduction in the overall factor and 5) orificing effects.

Items 2 and 4 were evaluated using results from a digital computer code which solves the mass, energy and momentum equations simultaneously. This is discussed in more detail in Section 5.4 where fuel assembly temperature characteristics are discussed.

Of considerable importance in the enthalpy rise hot channel factor is the effect of fuel rod bow which includes both initial and operational thermal bow. Figure 5-48 illustrates how the local enthalpy rise hot channel subfactor is affected by rod bow where the bow is assumed to be toward the center of the subchannel. Three different cases have been considered where either one, two or three rods are bowed. In the development of this figure, it was assumed that an equal pressure gradient existed for all parallel channels and that velocity was proportional to the two-thirds power of equivalent diameter. Thus:

$$F_{\Delta H}^{'E} = \frac{\Delta H \text{ (hot channel)}}{\Delta H \text{ (nominal)}} = \frac{\text{mass flow rate in nominal channel}}{\text{mass flow rate in hot channel}} \quad (5-35)$$

Figure 5-48. Local Enthalpy Rise Hot Channel Subfactor Due to Statistical Tolerances and Rod Bow

or

$$F_{\Delta H}^{'E} = \frac{(d^{2/3})_{\text{hot}}}{(d^{2/3})_{\text{nominal}}} \frac{\left(\sqrt{\frac{3}{2}} P^2 - \frac{\pi d^2}{4}\right)_{\text{nominal}}^{5/3}}{\left(\sqrt{\frac{3}{2}} P^2 - \frac{\pi d^2}{4}\right)_{\text{hot}}^{5/3}}$$

where

D = rod diameter

P = rod pitch

Defining the rod bow by δ , the expression for $F_{\Delta H}^{'E}$ for the three cases considered is as follows where P and d are the initial nominal and hot channel conditions:

$$F_{\Delta H}^{'E} = \frac{(D^{2/3})_{\text{hot}}}{D^{2/3}_{\text{nominal}}} \frac{\left(\frac{3}{2} P^2 - \frac{\pi d^2}{4}\right)_{\text{nominal}}^{5/3}}{\left(\frac{\sqrt{3}}{2} P^2 - P\delta \frac{\pi d^2}{4}\right)_{\text{hot}}^{5/3}} \quad (1 \text{ rod bowed}) \quad (5-36)$$

$$F_{\Delta H}^{'E} = \frac{(D^{2/3})_{\text{hot}}}{(D^{2/3})_{\text{nominal}}} \frac{\left(\frac{\sqrt{3}}{2} P^2 - \frac{\pi d^2}{4}\right)_{\text{nominal}}^{5/3}}{\left((P - \sqrt{3}\delta) \left(\frac{\sqrt{3}}{2} P - 1/2\delta\right) \frac{\pi d^2}{4}\right)_{\text{hot}}^{5/3}} \quad (2 \text{ rods bowed}) \quad (5-37)$$

$$F_{\Delta H}^{'E} = \frac{(D^{2/3})_{\text{hot}}}{(D^{2/3})_{\text{nominal}}} \frac{\left(\frac{\sqrt{3}}{2} P^2 - \frac{\pi d^2}{4}\right)_{\text{nominal}}^{5/3}}{\left((P - \sqrt{3}\delta) \left(\frac{\sqrt{3}}{2} P - 3/2\delta\right)\right)_{\text{hot}}^{5/3}} \quad (3 \text{ rods bowed}) \quad (5-38)$$

The effect of swelling and operational thermal bow on the enthalpy rise hot channel subfactor was determined for one rod using Equation 5-36. This assumption is based upon the bowing model established in Section 4.7. The results are shown in Figure 5-49 for end of life swelling rates of 1, 2, and 3 percent. It is noted that for zero bow, swelling increases this subfactor

Figure 5-49 Effect of Swelling on Enthalpy Rise Hot Channel Subfactor for Bowing of One Rod

from 1.036 (no swelling) to 1.329 (3 percent swelling), thus illustrating the vital need for accurate swelling data.

In the final evaluation, this particular subfactor must be obtained by numerically integrating stepwise along the length of the core, the effect of swelling and bowing. When this analysis was conducted, maximum bow for a honeycomb gridded assembly with a flow baffle was calculated to be six mils. Since axial variations were not available, this maximum bow was assumed to have a sinusoidal distribution between grids and this was repeated along the length of the rod. Integrating the amount of rod swelling (shown in Figure 5-49) and bowing, this subfactor was determined for each P/D ratio to be as follows:

P/D	$F_{\Delta H}^{'E}$
1.12	1.282
1.16	1.218
1.20	1.182

It has been assumed that the bow was the same for each of the P/D ratios investigated.

Later results presented in Figure 4-23 indicate that the average maximum bow for a condition of 3 percent rod swelling is approximately 4 mils; thus the results presented above represent a design conservatism. The effect of this conservatism on maximum clad surface temperature will be discussed in a later section where the total hot channel factors are discussed.

5.7.3 Film - Temperature Drop Hot Spot Factor

Besides fabrication and flow distribution effects, several localized effects influence the film temperature drop. Some of these effects are: 1) uncertainty in the local heat transfer coefficient, 2) mass transfer, fouling and gas entrainment, 3) hot spot due to obstruction and 4) circumferential temperature variations around the rod. All of these localized effects represent non-statistical contributions to the overall total factor.

The digital computer code "AXTHERM"[57] was used to evaluate the circumferential temperature distribution for the range of P/D ratios considered in this present study. These results are shown in Figure 5-50 and are tabulated below

P/D	F_{θ}'
1.12	1.650
1.16	1.400
1.20	1.250

5.7.4 Total Hot Channel Factor

The results of a hot channel factor analysis are presented in detail in Table 5-7, for a P/D ratio of 1.16. It is recognized that many of the subfactors presented here have been discussed qualitatively in previous paragraphs and are thus open to question.

Thus these subfactors should not be considered as final design numbers but as reasonable estimates which will be verified by analysis or experiment at some later point in the design stage. The final tabulation of engineering hot channel factors for the three P/D ratios considered are as follows:

P/D	F_q^E	$F_{\Delta H}^E$	F_{θ}^E
1.12	1.195	1.633	2.457
1.16	1.195	1.554	2.091
1.20	1.195	1.510	1.867

5.7.5 Maximum Clad Surface Temperature

Clad surface temperature was determined for the hot channel by using a digital computer code which solves for coolant, clad, and fuel temperatures as a function of the axial location. These results are shown in Figure 5-51 as a function of the engineering enthalpy rise hot channel factor, since this factor has the largest effect on the results. It is noted that the maximum clad surface

TABLE 5-7

HOT CHANNEL FACTORS FOR FTR CORE P/D = 1.16

Hot Channel Factors	F_q - Heat Flux	$F_{\Delta H}$ - Enthalpy	F_{θ} - Film Drop
Statistical:			
1. Pellet diameter, density and enrichment	1.080	1.030	1.080
2. Fuel diameter, pitch, and bowing	1.004	1.218	
3. Power measurement and control	1.050	1.050	
Non-Statistical:			
1. Inlet flow distribution per fuel tube		1.100	1.014
2. Flow redistribution		1.017	1.008
3. Outlet back pressure		1.020	
4. Orificing		1.060	1.029
5. Mixing		0.960	0.982
6. Uncertainty in heat transfer corr.			1.25
7. Mass transfer, fouling, entrainment			1.05
8. Hot spot due to obstruction			1.02
9. Circumferential variations around the clad			1.400
10. Maldistribution of flux	1.04	1.04	
11. Burnup uncertainty	1.05	1.05	

TABLE 5-7 (Cont)

	F_q - Heat Flux	$F_{\Delta H}$ - Enthalpy	F_{θ} - Film Drop
Total Engineering Factor	1.195	1.554	2.091
Nuclear Factors:			
1. F_R^N	1.300	1.300	1.30
2. F_Z^N	1.240		1.24
Total Nuclear Factor	1.612	1.300	1.612
Total Hot Channel Factor	1.927	2.020	3.371

Figure 5-50. Film Drop Hot Channel Subfactor to Account for Circumferential Temperature Variation

Figure 5-51 Effect of Enthalpy Rise Hot Channel Factor on Maximum Clad Surface Temperature

temperature ranges from 1285°F for a P/D ratio of 1.20 to 1345°F for a P/D ratio of 1.12, using the design results from Section 5.7.5.

The results for a P/D ratio of 1.16 have been further investigated considering the effect introduced by each important subfactor. These results are tabulated in Table 5-8. It is noted that bow increases the maximum temperature by 350°F while the combined effect of bow and swelling results in a 94°F rise in the maximum temperature. These results indicate that the maximum clad surface temperature can be expected to be in the range from 1230 to 1306°F. To narrow down these results any further would require quantitative information which is just not available at this point in the design stage.

5.8 BYPASS FLOW

In order to minimize stresses caused by thermal gradients across the duct wall, it is necessary to determine the amount of bypass flow outside the duct which will cause this coolant to experience approximately the same axial temperature profile as the main flow of coolant inside the duct. The bypass coolant is heated by:

1. Heat conducted through the duct wall from the main coolant flow.
2. Heat conducted from the duct wall due to gamma heating of the wall.
3. Heat generated due to gamma heating of the bypass coolant.

To calculate the temperature profile of a given amount of bypass coolant, consider a differential axial element of this coolant as shown in Figure 5-52.

Writing a heat balance on the duct wall:

$$q_D = U_1 P_1 dX (T_D - T_1) + U_2 P_2 dX (T_D - T_2) \quad (5-39)$$

where:

q_D = the heat generated

U = overall heat transfer coefficient

P = wetted perimeter

TABLE 5-8

EFFECT OF ROD AND PITCH TOLERANCES ON MAXIMUM
COOLANT AND CLAD TEMPERATURES
P/D = 1.16

Condition	Maximum Temperature °F		Value of Engineering Hot Channel Factor		
	Coolant	Clad	F_q^E	$F_{\Delta H}^E$	F_{θ}^E
1. Nominal, nuclear contributions only	1006	1022	1.0	1.0	1.0
2. Nominal, excluding rod and pitch variations	1165	1206	1.195	1.342	2.091
3. Nominal, same as 2 with 3 percent swelling	1190	1230	1.195	1.394	2.091
4. Minimum Pitch - Maximum rod diameter, no bow or swelling	1171	1212	1.195	1.355	2.091
5. Minimum Pitch - One rod with initial bow	1190	1230	1.195	1.394	2.091
6. Minimum Pitch - Two rods with initial bow	1212	1253	1.195	1.443	2.091
7. Minimum Pitch - All rods of subchannel with initial bow	1237	1278	1.195	1.496	2.091
8. Minimum Pitch - Initial and operational bow	1206	1247	1.195	1.429	2.091
9. Minimum Pitch - Initial and operational bow and swelling effects	1264	1306	1.195	1.554	2.091

Figure 5-52. Temperature Profile of Bypass Coolant.

X = axial coordinate

T = temperature

subscript D refers to duct

subscript 1 refers to main coolant

subscript 2 refers to bypass coolant

Dividing through by dX and rearranging,

$$T_D = \frac{q'_D + U_1 P_1 T_1 + U_2 P_2 T_2}{U_1 P_1 + U_2 P_2} \quad (5-40)$$

where:

q'_D is the linear heat generation rate of the duct.

Similarly, a heat balance on the bypass coolant gives

$$q_2 + U_2 P_2 dX (T_D - T_2) = \rho VA C_p dT_2 \quad (5-41)$$

where:

q_2 = heat generated in bypass coolant

ρ = coolant density

V = coolant velocity

A = flow area at coordinate X

C_p = coolant specific heat

Because of the relatively thin duct wall,

$$P_1 \approx P_2 = P \quad (5-42)$$

After dividing through by dX in Equation (5-41) and substituting Equations (5-50) and (5-42),

$$\begin{aligned}
q'_2 + \left(\frac{U_2}{U_1 + U_2}\right) q'_D + \left(\frac{U_1 U_2 P}{U_1 + U_2}\right) T_1 \\
+ U_2 P \left(\frac{U_2}{U_1 + U_2} - 1\right) T_2 = \rho V A C_p \frac{dT_2}{dX}
\end{aligned} \quad (5-43)$$

Now, assuming that gamma heat generation rate per pound of material is a constant through the core,

$$q'_2 = C_1 \rho A \quad (5-44)$$

where:

C_1 = a constant (Btu/ (hr/lb)

A = the flow area at coordinate X

For the dispersed FTR core,

$$A = A_o + C_2 X \quad (5-45)$$

where:

A_o = bypass flow area at core inlet

then

$$q'_2 = \rho C_1 A_o + \rho C_1 C_2 X \quad (5-46)$$

Also,

$q'_D = \text{constant}$

It can be seen in Figure 5-53 that the temperature of the main coolant stream adjacent to the duct wall can be approximated within the core by a straight line of the form

$$T_1 = C_3 + C_4 X \quad (5-47)$$

Substituting Equations (5-45) and (5-46) into Equation (5-43) and rearranging,

$$\begin{aligned} \frac{dT_2}{dX} + \frac{1}{\rho V A C_p} \left(\frac{U_1 U_2 P}{U_1 + U_2} \right) T_2 = \frac{1}{\rho V A C_p} \left[\rho C_1 A_o + \rho C_1 C_2 X + \left(\frac{U_2}{U_1 + U_2} \right) q_D' \right. \\ \left. + \left(\frac{U_1 U_2 P}{U_1 + U_2} \right) (C_3 + C_4 X) \right] \end{aligned} \quad (5-48)$$

This is a linear first order differential equation of the form:

$$\frac{dy}{dx} + P(x)y = Q(x)$$

The general solution of which is:

$$y e^{\int P dx} = \int Q e^{\int P dx} dx + C$$

where:

$$P(x) = \frac{1}{\rho V A C_p} \left(\frac{U_1 U_2 P}{U_1 + U_2} \right) = K_1 \quad (5-49)$$

Figure 5-53. Coolant Temperature in Hottest Outer Channel versus Axial Length

$$\begin{aligned}
Q(x) = & \frac{1}{\rho VAC_p} \left[\rho C_1 A_o + \left(\frac{U_2}{U_1 + U_2} \right) q_D' + \frac{U_1 U_2 PC_3}{U_1 + U_2} \right] \\
& + \frac{1}{\rho VAC_p} \left[\rho C_1 C_2 + \left(\frac{U_1 U_2 P}{U_1 + U_2} \right) C_4 \right] X = K_2 + K_3 X
\end{aligned} \tag{5-50}$$

Then,

$$T_2 e^{K_1 X} = \frac{K_2}{K_1} e^{K_1 X} + \frac{K_3}{K_1^2} (K_1 X - 1) e^{K_1 X} + C$$

Since

$$T = T_o @ X = 0$$

$$C = T_o - \frac{K_2}{K_1} + \frac{K_3}{K_1^2}$$

The general solution of Equation (5-48) is then:

$$T = \frac{T_o}{e^{K_1 X}} + \frac{K_2}{K_1} \left(1 - \frac{1}{e^{K_1 X}} \right) + \frac{K_3}{K_1^2} \left(K_1 X - 1 + \frac{1}{e^{K_1 X}} \right) \tag{5-51}$$

The gamma heating constants C_1 and q_D' were calculated from overall gamma heating data given for the FTR core. Since the values of the constants calculated from these data would be average values over the core, the appropriate hot channel factors were applied to obtain core maximum values. These factors are 1.11 for the subassembly peak to average factor and 1.17 for the peak to average factor within a subassembly. Thus, the linear heat generation rates used were 1.30 times the values calculated from Reference 72. Figure 5-54

Figure 5-54. Bypass Flow Requirements

shows the percentage of bypass flow required to give the same axial temperature profile as the profile as the main coolant. This percentage is based on a main coolant flow rate of 0.144×10^6 lb/hour. Since Reference 72 indicates an uncertainty of ± 10 percent on the gamma heating data, the effect of increasing the nominal heating rates is shown. The curve shown for zero overall heat transfer coefficient of main coolant to duct wall is an upper limit since the main coolant stream serves as a sink.

5.9 CALCULATION OF THE FORM LOSS COEFFICIENT OF A GRID

A grid may be divided into many subchannels as shown in Figure 5-55. Since the lateral pressure gradients at any level are very small, the assumption of equal pressure gradient across these subchannels is valid. Thus, the pressure drop across each subchannel is equal and can be calculated by:

$$\Delta P_i = K_i \frac{\rho v_i^2}{2g} \quad (5-52)$$

where

ΔP_i = Pressure drop across subchannel i

K_i = Total hydraulic resistance

V_i = Velocity

ρ = Density

g = Gravitational constant

This equation neglects the cross flow component of pressure drop, from region i, to another region, that appears in the momentum equation. This component of pressure drop is usually small compared to ΔP_i .

The loss coefficient, K_i , is made up of several components. These components are:

Figure 5-55. Subchannel Division of a Grid

5.9.1 Contraction Loss

This is a loss due to the contraction of the fluid when it enters the grid. It can be closely related to the graphical relationship presented by Kays and London^[73] with the following equation:

$$K_{i, 1} = 0.4 (1 - R_i)^2 \quad (5-53)$$

where

R_i = Ratio of flow area within grid subchannel to areas before contraction.

5.9.2 Expansion Loss

This occurs when the fluid leaves the grid area and an expansion occurs. The following relationship^[74] was used to calculate this loss

$$K_{i, 2} = (1 - R_i)^2 \quad (5-54)$$

5.9.3 Frictional Pressure Drop

In the grid area, frictional pressure drop may be appreciable because of the large values of the length to diameter ratio. This occurs primarily because additional wetted perimeter exists in the grid area causing the equivalent diameter of individual subchannels to be small. The form loss is:

$$K_{i, 3} = f \left(\frac{L}{D} \right)_i \quad (5-55)$$

where f is the Moody Friction Factor. It may be closely approximated^[75] by the following simple relationship.

$$f = 0.184 / Re^{0.2} \quad (5-56)$$

5.9.4 Additional Form Loss

Additional form loss may occur in individual subchannels. Springs, dimples and positioners are some of the "obstructions" that may cause this phenomenon. This form loss is usually a function of area ratio and can usually be found in many references, such as Reference 76, once a good hydraulic analogy is made of this "obstruction." In many instances, one may treat it as an orifice. The form loss is some constant

$$K_{i, 4} = (\text{constant})_i \quad (5-57)$$

5.9.5 Total Form Loss

The total form loss for subchannel i is the sum of the individual form losses.

$$K_i = K_{i, 1} + K_{i, 2} + K_{i, 3} + K_{i, 4} \quad (5-58)$$

Using the continuity equation for a constant density fluid

$$\sum A_i V_i = V \sum \frac{A_i}{R_i} \quad (5-59)$$

where

A_i = Flow area within grid subchannel

V_i = Velocity within grid subchannel

V = Velocity before the grid region

and combining it with Equation (5-52), the following simultaneous equations can easily be derived.

$$\begin{bmatrix}
 A_1 + A_2 + \dots + A_n \\
 \sqrt{K_1} - \sqrt{K_2} + 0 \dots 0 \\
 \sqrt{K_1} + 0 - \sqrt{K_3} \dots 0 \\
 \sqrt{K_1} + 0 + 0 \dots -\sqrt{K_n}
 \end{bmatrix}
 \begin{bmatrix}
 V_1 \\
 V_2 \\
 \\
 V_n
 \end{bmatrix}
 =
 \begin{bmatrix}
 V \sum \frac{A_i}{R_i} \\
 0 \\
 \\
 0
 \end{bmatrix}
 \quad (5-60)$$

Solution of this set of simultaneous equations yields a velocity distribution within the n subchannels of the grid. However, this velocity distribution is not quite correct because the friction factor was not calculated correctly since the velocity distribution was previously unknown. Using the new velocities, another friction factor is calculated and Equations (5-60) are solved again. This iteration process is repeated until the velocities converge. Convergences usually occur in three or less iterations.

The pressure drop is then calculated using Equation (5-52). The total grid form loss may be expressed as the ratio of grid pressure drop divided by the velocity head in the bare rod region.

$$K_{\text{grid}} = \frac{K_1 \rho V_1^2}{\rho V^2} \quad (5-61)$$

Subchannel one was used in Equation (5-61), but any subchannel could have been used since the pressure drop across each region is equal.

These equations were programed for use on a digital computer and they were used to evaluate grid loss coefficients for both the honeycomb and rhombus-cluster grid. They were not used for pressure drop calculation of the triplanar grid because all of this grid does not lie in one plane. The calculational procedure used for this grid is described in Section 5.2.1.2.

CHAPTER 6

EVALUATION

6.1 GRID MATERIAL

The three types of grids considered for the FFTF fuel rod assembly are:

Triplanar

Cluster Rod

Honeycomb

Recently the choice has been narrowed down to the honeycomb type, although the grid material, brazing alloy and method of joining would be applicable to all three types.

The operating conditions of the FFTF grid have been defined as:

Simultaneous rate past the grid	~2 fps
Temperature	1250°F (max. hot channel)
	1050°F (max. average)

The candidate grid and core materials considered are AISI type 316 stainless steel, Inconel-718, Inconel X-750 and TZM (Mo-base). The chemical composition, tensile properties and stress relaxation properties are shown in Tables 6-1, 6-2 and 6-3, respectively. It has been established that a material with a yield strength, at design temperature, of 90,000 psi or greater is required. Of the materials considered, AISI type 316 stainless steel is eliminated as inadequate. Although TZM would be expected to have a yield strength greatly in excess of 90,000 psi in the 1000°F/1400°F temperature range, it is not currently being considered; the reason being that fabrication of a complex grid assembly of TZM would be extremely difficult using current technology. Comparing Inconel-718 and Inconel-X-750, the tensile and relaxation properties of Inconel-718 are superior to Inconel-X-750 at the design temperatures for FFTF.

TABLE 6-1
TYPICAL COMPOSITIONS OF CANDIDATE ALLOYS FOR GRIDS (w/o)

<u>Alloy</u>	<u>C</u>	<u>Mn</u>	<u>P</u>	<u>S</u>	<u>Si</u>	<u>Cr</u>	<u>Ni</u>	<u>Mo</u>	<u>Ti</u>	<u>Others</u>
316	0.08	2.00	0.045	0.030	1.0	16.00/ 18.00	10.00/	2.00/		Balance FC
Inconel 718	0.04	0.20	---	0.007	0.30	18.6	52.5	3.1	0.90	0.07% Cu, 0.40% Al 5.0% Cb, 18.5% Fe
Inconel X-750	0.04	0.70	---	0.007	0.30	15.0	73.0		2.50	6.75% Fe, 0.05% Cu, 0.80% Al
TZM	0.03	--	---	--	--	--	--	Bal.	0.50	0.1% Zr

TABLE 6-2

TYPICAL SHORT-TIME, HIGH-TEMPERATURE TENSILE PROPERTIES
(First Value is Yield Stress and Second Value is Percent Elongation)

<u>Alloy</u>	<u>Condition</u>	<u>1000°F</u>	<u>1200°F</u>	<u>1400°F</u>	<u>2000°F</u>
316		25,000	23,000	21,000	
Inconel 718	Annealed, Aged Rod Sheet annealed at 1700°F and aged	137,000-26	148,800-22.0	91,000-8	
Inconel X-750	Annealed, Aged Rod Sheet annealed at 1950°F and aged	109,500-29 109,600-29	104,600-9 104,000-10	78,000-5	
TZM	Sheet	--	--	--	77,700-4.3

TABLE 6-3

STRESS RELAXATION AT TEMPERATURES ABOVE 1000°F^a

Alloy	Wire Temper	<u>Heat Treatment</u>		1000 psi Stress	<u>1100°F</u>		<u>1200°F</u>		<u>1300°F</u>	
		Temp, (°F)	Time, (hr)		1 hr	100 hr	1 hr	100 hr	1 hr	100 hr
Inconel Alloy 718	15%	1800	1	60	0.4	1.7	1.5	12.3	----	----
		1325	8 ^a		----	----	1.0	9.7	----	----
Inconel Alloy X-750	Spring	2100	2	60	2.3	12.0	----	----	----	----
		1550	24							
		1300	20	50	0.6	1.8	----	----	----	----
				40	----	----	0.8	4.1		
				30	----	----	0.8	3.9	1.8	8.0
				20	----	----	----	----	3.8	8.0

a. Then furnace cool to 1150°F and hold for total time of 16 hours.

It has been reported that the stress relaxation properties of nickel and the high nickel alloys (70-100% Ni) are degraded in the presence of thermal and fast neutron fluxes.^[1,2] However, Inconel-718 contains significantly less nickel than the alloys investigated and consequently may not behave in a similar manner. In addition, tests conducted by (W) on the Yankee Reactor Inconel-718 grids after irradiation to 2×10^{20} nvt (> 1 MeV) at 500°/600°F did not show any gross relaxation. The measured forces to extract the removable fuel rods from the assemblies gave evidence that the spring forces were still high enough to produce a sizeable holding force on the fuel rods.

High nickel alloys generally exhibit greater corrosion in flowing sodium than the austenitic stainless steels. No data are available on the corrosion of Inconel-718 in flowing sodium but UKAEA data^[3] on comparable alloys, PE 16 (43% Ni), in sodium containing ~10 ppm oxygen at 1200°F and 35 fps gave a corrosion rate of ~1 mil/year. Based on this tentative figure it is not expected that the corrosion of Inconel-718 under FFTF conditions will be a problem.

The reference method of joining (described in Section 6.1.1) the Inconel-718 grid will be brazing. The brazing cycle is expected to be compatible with the complex heat treatment required to develop the optimum mechanical properties of Inconel-718. In addition (W) has considerable experience in fabricating and brazing Inconel-718 grids for PWR core assemblies.

Based on the above considerations Inconel-718 is selected as the reference grid material.

6.1.1 Brazing Alloy

An extensive review of alloys suitable for brazing Inconel-718 grids has been carried out by (W).^[11] Based on the evaluation of the following factors:

1. Sodium corrosion of brazing alloy
2. Base metal melting
3. Diffusion of brazing alloy elements into the base metal
4. Base metal erosion
5. Effects of brazing cycle on the mechanical properties of the base metal.

6. Related experimental reactor experience with brazements.

Nicrobraz-50 appears to be the most promising for brazing Inconel-718 grids for FFTF. Leaching of phosphorous from this alloy in flowing sodium which has previously been reported^[4] is not expected to have any deleterious effect at the FFTF operating temperature. Alternate brazing alloys being evaluated include coast metals -52, -53 and -NP.

6.2 FACTORS INFLUENCING THE MECHANICAL DESIGN

6.2.1 Swelling Induced Clad Strains

No generally accepted theory defining the nature of irradiation induced fuel rod swelling is currently available. In the analysis described in this report the following characteristics have been assumed.

1. Fuel rod swelling may be split into two components
 - a. Swelling of the cladding material
 - b. Straining of the cladding due to the constraint loading generated at the fuel pellet-clad interface.

The exact contribution made by each of the two components is as yet undefined. Only component (b) is of interest in terms of an applied mechanics analysis of the fuel rods. The assumption made in this report is that all the clad diametral change is due to mechanism (b). This assumption leads to a conservative result.

2. Evidence exists which suggests that measurable fuel rod swelling occurs only after the rod has been subject to irradiation for an incubation period. The length of this period has not been defined. In the analysis, swelling induced strains have been assumed to be generated at a uniform rate over the entire rod life. The inclusion of the incubation effect would increase the straining rate $\dot{\epsilon}$. In order to accommodate varying incubation period assumptions, straining rate has been treated as a variable in the analysis. (i.e., A range of terminal swelling strains has been covered, each is assumed to take place in a one-year period.)

6.2.2 Friction Coefficients

Friction coefficients are required in the analysis outlined in Sections 4.6 and 4.7. No data relevant to the stick-slip mode of operation is available for the material combinations and environmental conditions of interest. In the case of the swelling induced clad strain analysis, the coefficient of friction has been treated as a variable. Figure 4-10 shows the peak clad strains to be relatively insensitive to the assumed value of μ : a 100 percent change in μ produces only 15 percent change in the peak strain. The penalty involved in assuming a conservative value of μ in the analysis of Section 4.6 is therefore acceptable. This is not the case in the analysis of fuel rod thermal bow. Here the transverse rod deflections associated with rod buckling are critically dependent upon the compressive fuel rod loads generated at the support grids. These loads vary directly as the grid-fuel rod friction coefficient. The value used in the thermal bow analysis (0.32) was obtained from tests in which Inconel 718 formed both rubbing surfaces. The tests were conducted in sodium at 1200°F (Ref. LMEC Report No. NAA-SR-12520). The μ values quoted are for the *beginning of sliding* condition. Tests using the correct combination of materials, contact loading and ambient conditions are required in order to fix the value of μ for this analysis. This type of test is currently scheduled as a part of the Westinghouse LMFBR development program.

6.2.3 Local Effects at the Grids

Local distortions of the temperature pattern within the fuel rod cladding will occur at each grid location. Additional thermal stresses will be associated with this effect. No analysis of these stresses has been included in the study, however it is anticipated that they will not exceed 5000 lb/inch². Similar local thermal stresses will be produced by any rod support system in which coolant flow patterns are disturbed.

6.2.4 Flow Channel Closure Due to Fuel Rod Swelling

All swelling of the fuel rod is accommodated by further deflection of the grid spring finger. The rods rest against rigid dimples within each grid cell, hence swelling produces a lateral translation of the rod centerline. All flow

channels have the same distorted shape with the exception of those channels adjacent to the fuel assembly can. Rod swelling therefore results in no significant coolant flow redistribution over a major portion of the fuel assembly cross section. This is viewed as a significant feature of the compliant grid pin spacer.

6.2.5 Fuel Rod Axial Creep Buckling

This mode of fuel rod deformation is seen as a possible major limitation in LMFBR fuel assembly design. The tentative strut buckling equation given in Section 4.4.2 represents a very simplified initial approach to the problem. A need exists for more test data obtained under loading conditions more closely representing fuel rod service conditions. Extended on-load hold times at elevated temperatures will be a necessary feature of these further tests. At present the selection of the isochronous stress-strain curve as the curve governing the buckling behavior is somewhat arbitrary. This curve is known to be useful in predicting creep buckling behavior under load controlled conditions; its use in the analysis of strain controlled loading conditions however, may be overly conservative. Further tests would be used to generate a set of creep buckling design curves relevant to the fuel rod service conditions, and hopefully to obtain a fuller understanding of the mechanism of fuel rod creep buckling.

6.2.6 Alternate Design Considerations

Wire wrap pin spacing is the principal design alternative to the use of a compliant grid pin spacer. Some mention of how the problems discussed above relate to the wire wrap design is therefore relevant to an evaluation of the compliant grid pin spacer.

An inherent feature of the wire wrap concept is the existence of 'pinch points' along the length of the rod. These are points at which the centers of the wire wrap and fuel tubes all lie in one plane. At these points interaction between the duct and the rod bundle can produce transverse compressive loads within the bundle. The magnitude of these loads is governed by the transverse stiffness of the rod bundle and the duct stiffness. Frictional restraint of

the free motion of the fuel rod bundle relative to the duct at the pinch points, will produce the loading required to induce fuel rod axial creep buckling. The mode of buckling would not be identical with that proposed for the grid spacer design: a corkscrew buckled mode shape would seem most probable.

Geometric changes due to differential thermal expansion and irradiation induced rod swelling are accommodated in a wire wrap design by distortion of the fuel rod center line. This distortion results in an uneven distribution of coolant flow area within the fuel rod bundle.

Flow induced fuel rod vibration is a problem in both fuel bundle designs.

6.3 THERMAL HYDRAULIC-MECHANICAL DESIGN INTERACTION

The principal area of thermal hydraulic-mechanical design interaction is in the analysis of fuel rod/assembly thermal bow. The objective set for the thermal bow analysis was to obtain the best representation possible of the non-linear mechanical behavior. In order to devote more time to this aspect of the analysis, no attempt was made to couple the coolant temperatures with the deformed flow channel geometries. Temperatures are input into the code. In its present form it is intended that FATBOW II be used in conjunction with a manual iteration in order to converge on a stable bowed configuration for the fuel rods.

CHAPTER 7

DEVELOPMENT PROGRAM

7.1 MECHANICAL DEVELOPMENT PROGRAMS

Table 7-1 lists the development activities selected as key items in the production of a successful fuel assembly mechanical design.

7.2 THERMOFLUIDS DEVELOPMENT PROGRAMS

Thermofluids development programs necessary to support the design of a gridded FFTF fuel assembly are reported in this section. The development programs cover only the scope of the work of this contract and do not include such necessary items as two-phase flow, boiling superheat, burnout, and other liquid metal phenomena. Section 7.1 of this report covers the development area of thermal bow. Since this area requires strong interaction between mechanical and thermofluids areas, it is discussed as one overall program.

7.2.1 Pressure Drop and Flow Distribution

7.2.1.1 Basis

In the design of a reactor, it is necessary to determine the fuel assembly pressure drop and flow distribution within the subchannels of the fuel assembly. Grids, expansion of the duct, and possibly a flow baffle require experimental verification of calculational procedures.

7.2.1.2 Scope of Work

Analytical models will be developed to analyze the fuel assembly pressure drop, grid form loss and flow distribution. Prototypic bundles will be tested in water initially and a firmed or final design will be tested in sodium.

7.2.1.3 Milestones

1. Prototypic pressure drop tests in water.
2. Prototypic flow distribution tests in water.
3. Repeat steps 1 and 2 for firmed-up prototypic bundle using sodium as a fluid.

7.2.2 Heat Transfer

7.2.2.1 Basis

With a sodium cooled environment, predictions of temperatures (under a heat flux condition) require experimental justification. The film coefficient is usually a function of geometry, heat flux and fluid properties. A liquid metal causes circumferential variations around the cladding. The effect of grid hot spots must be determined.

7.2.2.2 Scope of Work

This development program will include experimentation in a liquid metal environment as well as development of analytical techniques. All heat transfer aspects, including effects of grids, will be studied. Some of the developmental areas are discussed in the preceding paragraphs. Small scale tests will be used if possible. Eventually, it will be desirable to run a full mockup of the fuel assembly.

7.2.2.3 Milestones

1. Bare rod heat transfer in a central cell using a small bundle.
2. Bare rod heat transfer in a side cell (i.e. the flow channel between the outer row of rods and the duct) using a small bundle.
3. Circumferential temperature distribution for side and central cells.
4. Temperature distribution within a grid.
5. Repeat step 4 but simulate swelling.
6. Heat transfer tests on a full scale bundle.

7.2.3 Mixing

7.2.3.1 Basis

The turbulence induced by bare rods and grids has a two-fold importance in the design of reactor fuel assemblies. 1) It is needed to accurately calculate bulk coolant temperature distributions in the various subchannels of a fuel assembly, 2) it is needed to reliably assess the magnitude of the enthalpy rise hot channel which directly affects the maximum available core power. Since turbulence is influenced strongly by both grid and rod configuration, experimental data is required to establish analytical methods.

7.2.3.2 Scope of Work

Prototypic fuel assemblies will be tested in water initially to provide a basic understanding of the mixing process. Preliminary analytical methods will then be established for the reference fuel assembly design stage. Later tests will then be performed using sodium as the working fluid to evaluate conduction effects. These later tests, in conjunction with the water results, will provide a basis for the development of final analytical methods.

7.2.3.3 Milestones

Prototypic fuel assemblies will be tested in water initially to evaluate the following:

1. The degree of turbulence or mixing occurring along bare rods for a uniform subchannel geometry.
2. The effect on mixing of subchannel area changes due to fabrication tolerances and bowing.
3. Turbulence decay characteristics downstream from a grid.
4. The influence of duct walls on the mixing characteristics of side subchannels.

Sodium conduction effects will be experimentally determined and combined with the preliminary analytical methods developed from the water tests.

A final method will be developed to adequately describe the mixing process in the FTR fuel assembly.

7.2.4 Analytical and Computational Techniques

7.2.4.1 Basis

It is not economically and/or physically possible to experimentally study all possible conditions that might occur during reactor operation. Analytical techniques are often used to study these varied conditions and results from experimental development programs are used as input for these analytical methods.

7.2.4.2 Scope of Work

At this time, it is not possible to determine all of the analytical capabilities that are necessary to design a FFTF fuel assembly. Some unforeseen problems will require additional techniques to be developed. However, some of the techniques that are now considered necessary are: 1) pressure drop prediction, including swelling, 2) flow coolant and fuel rod temperature distributions, and 3) prediction of heat transfer under varied conditions.

7.2.4.3 Milestones

1. Review current available techniques.
2. Modify techniques is necessary.
3. Develop new techniques if none are available.

TABLE 7-1. FUEL ASSEMBLY MECHANICAL DESIGN SCHEDULE

Activity Objective	Activity Description	Milestones
1. Spring finger development		
(a) Obtain a full understanding of the influence of spring geometry on stress levels and spring characteristics.	(a) Frozen stress and photostress tests on scaled up models of the grid spring finger designs.	(a) Produce a set of spring design rules. Establish load diffusion lengths and stress concentration factors.
(b) Optimize the spring geometry.	(b) Produce an optimum spring design using the results of (a).	(b) Drawings produced for section (c) tests.
(c) Establish the performance characteristics of the optimized spring design.	(c)i Load deflection tests on true scale spring models over the full spring working range.	(c)i Produce a spring load--deflection curve.
	(c)ii Stress/load relaxation tests under simulated in-reactor service conditions.	(c)ii Produce curves of spring load as a function of time and deflection.
	(c)iii Fatigue tests under simulated fuel rod vibration loading.	(c)iii Establish a safe working life for the spring.
2. Grid, can and brazing alloy corrosion/erosion tests		
(a) Establish the corrosion/erosion rates for both structural and braze materials for use in grid construction.	(a) Sodium loop tests simulating in-core conditions on grids fitted with dummy fuel rods and cans. In-reactor environmental conditions to be simulated. Periodic inspection used to monitor corrosion rates.	(a) Produce curves of corrosion/erosion material loss as a function of temperature, flow rate and time for braze and structural materials.
(b) Select the optimum joint geometry	(b) Vary the braze fillet radius in the tests described in (a) above. Establish the effective area of structurally sound braze alloy by metallurgical examination after varying exposure times.	(b) Produce design curves giving the required braze fillet radius as a function of service life.
3. Fuel rod creep buckling tests		
(a) Obtain an understanding of the parameters controlling the buckling of struts subject to circumferential plastic straining.	(a) Extend the initial Westinghouse series of pressurized strut tests.	(a) Fix the elastic constant effective in controlling buckling of the strut.

Activity Objective	Activity Description	Milestones
(b) Established the buckling characteristics of a strut subject to circumferential plastic straining and held under conditions of axial constraint loading at elevated temperatures for extended periods of time.	(b) Pressurized struts will be tested at elevated temperature under axial constraint loading. The time dependent growth of the strut transverse deflection will be monitored. A range of initial eccentricity and P/P_{crit} values will be used.	(b) i. The type of stress-strain curve giving the best fit of analytical results with test results will be selected. (b) ii. The strut transverse deflection equation developed in Section 4-4 will be checked. Empirical factors will be introduced if required.
4. Fuel Assembly Thermal Bow Tests		
(a) Demonstrate acceptable fuel assembly thermal bow characteristics under simulated operating conditions. Obtain fuel rod grid cell fretting and galling information.	(a) A fuel assembly prototype (possibly with a reduced number of fuel rods) will be tested in a sodium loop. Sodium conditions will be representative of in core conditions. Core support conditions and temperature gradients associated with critical core locations will be simulated. Fuel rod and assembly deflections will be monitored, together with clad/coolant temperatures. Frictional loading will be determined.	(a) Produce a report detailing the measured thermal bow characteristics of the fuel assembly. Relate these deflections to the analytical model developed to predict thermal bow.
5. Development of Analytical Models & Techniques		
(a) Create all the design tools necessary for the analysis and optimization of the fuel assembly mechanical design.		
(b) Produce low cycle fatigue design curves for all the fuel assembly materials. Incorporate the effects of environment and irradiation.	(a) Develop the FATBOW II computer code to include more exact fuel rod deformation models. (i.e. treat each rod as a continuous strut on a number of supports). Allow for a variety of duct support conditions. Couple the mechanical and thermal hydraulic effects. (b) Develop methods of allowing for the influence on fatigue life of all the in reactor conditions. From currently available literature obtain the data required to produce the fatigue curves. Identify data not available. Produce fatigue design curves.	(a) Produce an operational version of FATBOW II. (b) Produce a set of fatigue design curves for all fuel assembly materials/ braze alloys and typical joint designs.

Activity Objective	Activity Description	Milestones
(c) Obtain an understanding of the nature of irradiation induced fuel rod swelling.	<p>From currently available literature determine:</p> <ul style="list-style-type: none">i. The length of the swelling incubation periodii. The ratio of clad material swelling to total fuel rod swellingiii. The state of stress associated with clad swellingiv. The mechanical properties of irradiated material.	(a) Produce a set of design rules for the treatment of irradiation effects in fuel rods.
6. Fabrication Development		
(a) To establish the feasibility of all the operations required in fabrication of the sub-assemblies and to optimize fabrication techniques.	<ul style="list-style-type: none">(a) i. Manufacture of dies and the production of prototype grid strapsii. Manufacture of grinding fixtures and the development of grinding techniquesiii. Manufacture of brazing fixtures. Brazing of prototype grids. Development of brazing techniques.iv. Statistical determination of rod pitch tolerance for finished grids.v. Development and qualification of fuel assembly welding procedures.	(a) Produce a manufacturing specification for the sub-assembly together with a statistical analysis of rod pitch tolerances.
7. Fuel Assembly Static and Dynamic Tests		
(a) To establish the fuel assembly strength, stiffness and vibration characteristics. These data are required in order to more readily interpret the results of subsequent tests.	<ul style="list-style-type: none">(a) Load-deflection tests of a complete fuel assembly supported in the core constraint, fuel handling and shipping configurations. Strains to be monitored.(b) Forced vibration tests on the fuel bundle with the fuel assembly supported in the core constraint mode.	<ul style="list-style-type: none">(a) Produce a fuel assembly strength and stiffness report.(b) Produce a vibration report giving fuel bundle natural frequencies and mode shapes.
8. Flow Induced Vibration Tests		
(a) Determine the flow induced vibration characteristics of the fuel assembly under simulated operating conditions.	(a) Water loop tests on a complete fuel assembly. Dynamic similarity with reactor sodium	(a) Produce a preliminary fuel assembly

Activity Objective	Activity Description	Milestones
	<p>flow will be obtained. The full spectrum of reactor operating conditions will be investigated. Response of the fuel bundle (frequency, amplitude and mode shapes) will be monitored.</p> <p>(b) Sodium loop vibration tests on a complete fuel assembly. Test conditions will be those selected during the water loop screening tests. Mode shapes, amplitude and frequency of rod bundle vibrations will be monitored. Tests will simulate a complete fuel assembly life cycle. Extensive post test visual examination will be called for.</p>	<p>vibration report detailing the fuel bundle/fuel assembly vibration characteristics Identify critical operating conditions for further testing in the sodium loop.</p> <p>(b) Produce a report detailing the fuel assembly vibration characteristics under simulated in core conditions.</p>
9. Material and Fuel Assembly Irradiation Tests		
<p>(a) Obtain design data for materials irradiated under simulated FFTF nuclear conditions in flowing sodium.</p> <p>(b) Demonstrate satisfactory performance of a complete fuel assembly under simulated FFTF operating conditions. (Irradiation test)</p>	<p>(a) Irradiation tests coupled where possible with simultaneous mechanical tests. The principal mechanical properties of interest will be:</p> <ul style="list-style-type: none">i. Yield strengthii. Uniform elongationiii. Creep performanceiv. Stress relaxation performancev. Low cycle fatigue propertiesvi. Swelling rates <p>All fuel assembly materials will be tested</p> <p>(b) A fuel assembly (probably with a reduced number of fuel rods) will be tested under simulated FFTF operating conditions. Nuclear, thermal hydraulic and mechanical 'in core' conditions would all be simulated. The fuel life history loading of the FFTF fuel assembly will be simulated.</p>	<p>(a) Produce a design manual detailing irradiated material properties for all the proposed core materials, together with their associated design safety factors.</p> <p>(b) A successful result in the test will generate the confidence required to commit the fuel assembly design to production. A post test 'strip down and inspection' report will be issued.</p>

Activity Objective	Activity Description	Milestones
10. (a) To establish satisfactory performance of the fuel assembly under fuel handling conditions.	(a) Fuel handling tests will be run using a prototype unfuelled fuel assembly. The actual fuel handling machinery will be used in a reactor mockup. All fuel handling operations under both routine and emergency conditions will be performed. Fuel handling techniques will be varied and operator experience built up.	(a) A fuel handling report will be issued. This will establish compatibility of the fuel assembly with the fuel handling machine. Fuel handling techniques will be specified for all routine and emergency fuel handling operations.
(b) Establish compatibility of the fuel assembly with the fuel handling machine.		
(c) Provide the experience required to refine the fuel handling techniques under both routine and emergency conditions.		

CHAPTER 8

CONCLUSIONS

The honeycomb grid pin spacer incorporating a cantilever loaded torsion spring is shown to be a practical means of spacing fuel rods in an FFTF fuel assembly design. The design has advantages over the alternate wire wrap pin spacer design in that:

1. Its structural behavior under operating conditions is more predictable.
2. It ensures an even distribution of the effects of the differential thermal expansion and irradiation induced fuel rod swelling.

In addition, it offers advantages when compared with the other two grid concepts considered. These advantages include an acceptable pressure drop and low localized temperature increases due to the grid.

Materials evaluation is seen as the principal grid development area. Structural materials having a high yield strength and good stress relaxation characteristics coupled with low corrosion/erosion rates in flowing sodium are required. It is anticipated that the proposed development program will show Inconel-718 to be an accepted grid structural material.

It has not been possible to include an assessment of the effect of irradiation induced material swelling in the design analysis of the grid structure, due to lack of swelling data. A review of the designs presented will be required when these data become available.

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