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WARD BLANKET ASSEMBLY
PRE-TEST PREDICTIONS BY
SUPERENERGY

by

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Reports and Papers Published under
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1. Background

A heated LMFBR blanket assembly is under test at WARD to determine its thermal-hydraulic characteristics. This test section has nominal design parameters given in Table 1. A step cosine shape heat input and the rod bundle instrumentation planes labeled as A to F, are illustrated in Figs. 1 and 1-A in Ref. (1).

The purpose of this topical report is to report predictions of part of the WARD test results using the SUPER-ENERGY⁽²⁾ code with the aid of existing flow split and salt mixing data.⁽³⁾ A significant part of this effort is the detailed consideration of the uncertainties introduced in temperature field predictions due to as-built bundle tolerances.

2. Subchannel Parameters and Flow Condition of WARD Assembly

The subchannel parameters of WARD assembly, i.e., subchannel area and hydraulic equivalent diameters, are not uniquely determined from the bundle design parameters (Table 1). They also depend on the individual rod position which varies within the limits of the total bundle tolerance. This tolerance called the flat tolerance is defined as the clearance between flat to flat distance and the sum of the rod and wire diameters in the flat-to-flat direction (refer to Fig. 1 for details). For pin and wire diameters at their nominal dimensions, the flat to flat tolerance is 0.045 inches, which is evaluated as follows:

$$0.045" = 4.49 - (\sqrt{3} * 4) (0.519" + 0.037") - (0.519" + 0.037" \times 2)$$

The potential range of various parameters for each type subchannel can be obtained using the F-factor (looseness factor) method.⁽⁴⁾ For $F=1$, the flat tolerance is uniformly distributed among the interior channels. For $F=0$, all the flat tolerance accumulates at the edge channels. The potential range of these geometric parameters based on the .045 inch bundle tolerance are tabulated in Table 2A. As shown in this table, the relative large flat tolerance, 0.045 inches, can lead to large variations of subchannel geometric parameters over the possible F-factor range of 0 to 1.

A particular geometric configuration, i.e., particular value of F is of interest for later calculation. This is the configuration where the total tolerance is distributed such that the clearance between wire and duct in the edge channels equals the clearance between wire and pin in each interior subchannel. This configuration is defined as "nominal". The F value of 0.78 and the associated geometric parameters are listed in Table 2B.

If the maximum diameters of rods or of wires are used, the flat tolerance will be different from 0.045, the value used in Table 2. The worst case with respect to the variation of subchannel parameters is found for the case of smallest parameters is found for the case of smallest rod and wire diameters, i.e., 0.5715" and 0.0365" respectively. With this condition, the flat tolerance is 0.061 and the subchannel parameters are tabulated in Table 3 for various F values including the "nominal" configuration, $F=0.78$.

The influence of the variation of the subchannel parameters on the flow condition can be analyzed by Novenstern's method.⁽⁵⁾ The flow split results corresponding to the subchannel parameters obtained over the F range for the flat tolerance of 0.045 inches (Table 2) are tabulated in Tables 4 and 5 for turbulent and laminar flow respectively. From this table, it is worth noting that the flow splits among subchannels are highly dependent on the possible model of rod alignment.

3. Categories of WARD Experiments

Eleven test runs are planned utilizing differing power skews and inlet flow rates⁽¹⁾ with the WARD blanket assembly. It is important to recognize that these test runs fall into different classifications with regard to hydrodynamic and thermal behavior. Hence we would expect that analysis methods of varying sophistication and associated code input data having differing values in different flow regimes are required to cover the whole test range. Recognizing this we separate the tests into categories of "Forced Convection" and "Mixed Convection", as well as categories of "Turbulent", "Transition" and "Laminar". These arrangements are illustrated in Table 6. The terminology "Turbulence" in this table means that both the flow split and the mixing parameters, i.e., x_1 , x_2 , x_3 , ϵ_1^* and C_1 saturate to constant values. The Reynolds number at which saturation occurs is approximately

about 15,000.⁽³⁾ On the other hand, for the cases having $2,300 \leq Re \leq 16,000$, the term "Transition" is used. Below 2,300, the traditional term "Laminar" is used. From MIT experiments with this blanket bundle geometry, the variation of all flow split and mixing parameters with Reynolds number have been determined.⁽³⁾

4. Input Parameters

The input parameters for the ENERGY code necessary to predict each category listed in Table 6 are based on the experimental data of ϵ_1^* , C_1 and flow splits obtained with the MIT 61 pin blanket assembly.⁽³⁾ The source of input parameters and the specific code in the ENERGY family which should be used to predict the WARD results are tabulated in Table 7.

4.1 First Effort Cases (Cases 1, 4, 6 and 9)

According to the recommendation from WARD,⁽⁸⁾ first effort on preprediction was put into cases 1, 4, 9 and 6. The first three cases are categorized as "Turbulent" and the last case as "Transition". The input data of ϵ_1^* and C_1 are directly quoted from Ref. (3) as Table 7 indicates. However, the flow split data which are crucial to determine the hot spot temperature are corrected for the P/D difference between the MIT 61 pin assembly and the WARD assembly. The methodology of this correction can be expressed⁽⁶⁾ as follows.

$$x_1(\text{WARD}) = x_1(\text{MIT mockup}) \frac{x_1(\text{Novendstern's prediction of WARD bundle})}{x_1(\text{Novendstern's prediction of MIT mockup})} \quad (1)$$

$i = 1, 2 \text{ and } 3$

With the above correction, the numerical values of these flow split input data, e_1^* and C_1 are tabulated in Table 8. On the other hand, the flow split results at the looseness extremes, i.e., $F=1$ and $F=0$, are evaluated by the following equation.

$$x_1(\text{WARD}, F=\frac{1}{0}) = \frac{x_1(\text{Novendstern's prediction of WARD at } F=\frac{1}{0})}{x_1(\text{Novendstern's prediction of WARD, nominal})} [x_1(\text{WARD}) \text{ from Eq. 1}] \quad (2)$$

$i = 1, 2 \text{ and } 3$

5. Predictions

The predicted subchannel temperatures for the nominal and extreme bundle configurations are plotted from flat to flat at plane A (refer to Ref. 9) in Figs. 2, 3, 4 and 5 corresponding to cases 1, 9, 4 and 6 respectively.

As shown in these figures, the hot spot temperature differences between $F=1$ and 0 are as much as 10 - 15% of the total axial temperature rise. These temperature profiles of $F=0$ and 1 result in a bracket covering the temperature effects of the possible rod alignment modes. Furthermore, even though the temperature profiles of the nominal rod alignment mode when plotted appear coincident with those of $F=1$, small physical differences, 1°F to 3°F , between these cases.

The side channel temperature predictions at the instrumentation level A together with the crossflat temperature predictions at the instrumentation levels F, C, D, B and E are reported in Appendix A. It is found that the edge channel temperatures as Figures A.21 to A.24 indicate, are somewhat independent of the rod alignment modes. This is primarily due to cancellation of two opposing effects. The first is the geometric effect on the flow split, the other is the enhanced energy mixing between the side channels and the interior channels. For instance, if $F=1$, the axial edge velocity is decreased due to channel closure, resulting in higher edge temperatures. Correspondingly, the energy transferred from the adjacent interior channels to the edge channels will be decreased due to the decreased temperature gradient. The net effect is that edge channel temperatures are relatively independent of the rod alignment mode.

6. Implication of Rod Alignment Mode on Required Accuracy of Predictive Codes

As shown above, the inherent uncertainty in rod alignment modes within the existing flat tolerance results in large interior subchannel temperature uncertainties but small edge subchannel temperature uncertainties.

In design practice, since information is not available on the true F factor value, the nominal value of F equal to 0.78

would be assumed. With this assumption various ϵ_1^* and C_1 values should be used to obtain the same temperature results as would result from using the true F and the nominal ϵ_1^* or C_1 . The band of ϵ_1^* and C_1 values are shown in Figs. 6 and 7. These results are just an alternate way of confirming the conclusions drawn in Section 5--that is, the uncertainty in alignment mode strongly affects the interior subchannel temperatures and weakly affects the edge subchannel temperatures.

7. Recommendations and Conclusions

From the above discussion the importance of the rod alignment mode on the coolant temperature predictions would be recognized. This uncertainty in alignment mode is even further enhanced when power operation considerations are included.⁽¹⁰⁾ It is clear that the most fruitful application of resources is in experimental studies to interpret alignment modes rather than refinements of steady state and probably transient analysis methods. The following experimental tasks are recommended in this respect.

- 1) Physical measurement of the F-factor on a statistical base in real as-built CRBR blanket assemblies such that the nominal F-factor and its confidence level

can be determined for reactor design application. Additionally some strategy to characterize geometry at power with varying burnup needs to be devised.

- 2) Flow split measurements within the F-factor range from 0 to 1 of the MIT blanket assembly mockup. This will provide a data base to which measurements from 1) above can be applied as they become available. On a practical basis, measurements have been made⁽³⁾ at nominal conditions ($F=0.78$) and could be also made at the $F=0$ extreme.
- 3) Experimental determination of the swirl flow parameter, C_1 , versus key bundle parameters P/D and H/D . Where necessary this should take priority over determination of the internal mixing parameters, ϵ_1^* . This is because the peripheral temperatures can be accurately determined with the aid of the C_1 even though the actual rod alignment mode of the subject bundle is unknown.

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TABLE 1
Radial Blanket Test Model Parameters

As-Built Duct Inside Dimension, Flat-to-Flat	in.	4.490 (as-built per Ref. 9)
Number of Rods		61
Sheath Material of Rods		20% CW SS316
Length of Heated Region	in.	45
Length of Rod	in.	90.4
Rod Outside Diameter	in.	0.519 $\pm$ 0.0015
Triangular Rod Pitch, Nominal	in.	0.560
Pitch-to-Diameter Ratio, Nominal	in.	1.077
Wire Wrap Diameter/Lead	in.	0.037/4.0
Axial Power Distribution - Chopped Cosine	max/avg	1.40
Radial Power Distribution - Variable	max/avg	2.15
Radial Power Distribution - Variable	min/avg	0.41
Radial Power Distribution - Variable	max/min	4.48 to 1.00
Range to Peak Linear Power	kw/ft	0.4 to 14.8
Coolant		Sodium
Coolant Inlet Temperature	°F	500 - 800
Coolant Outlet Temperature	°F	700 - 1100
Coolant Flow Rate	gpm	1 to 150 max.
Coolant Velocity	ft/sec	0.1 to 11
Coolant Reynolds Number	---	400 to 40,000
Coolant Grashof Number	---	10,000 to 500,000
Assembly Total Power	kw	20 to 1100
Duct Corner Radius	in.	.125

TABLE 2
Subchannel Geometric Parameters of WARD Blanket Assembly

TABLE 2.A

Flat Distance = 4.4900"
Rod Diameter = 0.5190"
Wire Diameter = 0.0370"
Flat Tolerance = 0.045"
P/D Ratio = 1.071

F-Factor	Type	Area	Wetted Perimeter	Hydraulic Dia
0.0	Interior	0.0275 in ²	0.8757 in	0.1256 in
	Edge	0.0710	1.4317	0.1983
	Corner	0.0233	0.6602	0.1411
0.20	Interior	0.0281	0.8757	0.1285
	Edge	0.0689	1.4330	0.1923
	Corner	0.0216	0.6550	0.1322
0.40	Interior	0.0288	0.8757	0.1313
	Edge	0.0668	1.4343	0.1863
	Corner	0.0200	0.6498	0.1232
0.60	Interior	0.0294	0.8757	0.1342
	Edge	0.0647	1.4356	0.1802
	Corner	0.0184	0.6447	0.1143
0.80	Interior	0.0300	0.8757	0.1371
	Edge	0.0626	1.4369	0.1741
	Corner	0.0169	0.6395	0.1054
1.00	Interior	0.0306	0.8757	0.1400
	Edge	0.0604	1.4382	0.1680
	Corner	0.0153	0.6343	0.0965

TABLE 2.B

F-Factor	Type	Area	Wetted Perimeter	Hydraulic Dia
0.78	Interior	0.0299 in ²	0.8575 in	0.1367
	Edge	0.0628	1.4368	0.1749
	Corner	0.0170	0.6401	0.1065

TABLE 3
Subchannel Geometric Parameters of WARD Blanket Assembly
($D_p = 0.5175''$, $D_w = 0.0365''$)

Flat Distance = 4.4900"
Rod Diameter = 0.5175"
Wire Diameter = 0.0365"
Flat Tolerance = 0.061"
P/D Ratio = 1.071

F-Factor	Type	Area	Wetted Perimeter	Hydraulic Dia
0.0	Interior	0.0272 in ²	0.8725 in	0.1245 in
	Edge	0.0748	1.4265	0.2098
	Corner	0.0261	0.6672	0.1563
0.20	Interior	0.0280	0.8725	0.1284
	Edge	0.0720	1.4283	0.2016
	Corner	0.0238	0.6601	0.1442
0.40	Interior	0.0289	0.8725	0.1323
	Edge	0.0691	1.4301	0.1934
	Corner	0.0215	0.6530	0.1320
0.60	Interior	0.0297	0.8725	0.1363
	Edge	0.0663	1.4318	0.1851
	Corner	0.0194	0.6459	0.1198
0.80	Interior	0.0306	0.8725	0.1402
	Edge	0.0634	1.4336	0.1768
	Corner	0.0172	0.6388	0.1077
1.00	Interior	0.0314	0.8725	0.1442
	Edge	0.0604	1.4354	0.1685
	Corner	0.0151	0.6318	0.0955
0.78	Interior	0.0305	0.8725	0.1397
	Edge	0.0637	1.4334	0.1778
	Corner	0.0175	0.6397	0.1091

TABLE 4

Flow Split Results of the Novendstern's Prediction
Over the F Range for the Turbulent Flow Condition

Flow Regime	= Turbulent
Flat Distance	= 4.4900"
Rod Diameter	= 0.5190"
Wire Diameter	= 0.0370"
No. of Rods	= 61
Flat Tolerance	= 0.045"
Lead Length	= 4.000"
P/D Ratio	= 1.071

F-Factor	x ₁	x ₂	x ₃	Pitch
0.0	0.870	1.206	0.945	0.556
0.1	0.880	1.197	0.927	0.557
0.2	0.890	1.187	0.908	0.557
0.3	0.900	1.177	0.889	0.558
0.4	0.909	1.067	0.869	0.559
0.5	0.918	1.156	0.848	0.559
0.6	0.927	1.145	0.827	0.560
0.7	0.936	1.133	0.805	0.561
0.8	0.945	1.121	0.783	0.561
0.9	0.953	1.108	0.760	0.562
1.0	0.961	1.095	0.737	0.562

Table 5

Flow Split Results of the Novendstern's Prediction
Over the F Range for the Laminar Flow Condition

Flow Regime	= Laminar
Flat Distance	= 4.4900"
Rod Diameter	= 0.5190"
Wire Diameter	= 0.0370"
No. of Rods	= 61
Flat Tolerance	= 0.045"
Lead Length	= 4.000"
P/D Ratio	= 1.071

F-Factor	x_1	x_2	x_3	Fitch
0.0	0.635	1.582	0.801	0.556
0.1	0.660	1.560	0.763	0.557
0.2	0.685	1.535	0.725	0.557
0.3	0.711	1.509	0.687	0.558
0.4	0.736	1.481	0.648	0.559
0.5	0.761	1.450	0.610	0.559
0.6	0.787	1.418	0.571	0.560
0.7	0.811	1.384	0.532	0.561
0.8	0.836	1.349	0.494	0.561
0.9	0.860	1.312	0.457	0.562
1.0	0.883	1.273	0.420	0.562

TABLE 6
Categorization of WARD Eleven Test Runs

TURBULRNT			TRANSITION			LAMINAR		
Forced Convection (Gr* ≤ 0.025)						Mixed Convection (Gr* > 0.025)		
Case No.	Gr*	Re	Case No.	Gr*		Case No.	Gr*	Re
1	0.009	28,400	5	0.010	6,830	3	0.139	1,015
4	0.002	17,100	10	~0.0	6,830	7	0.116	926
9	0.003	17,100	6	0.025	5,260	11	0.129**	926
2	0.005	11,352						
8	0.004	11,121						

$$Re = \frac{\rho v \overline{D_e}}{\mu}, \quad Gr^* \equiv \text{modified Grashof Number}^{(7)}$$

** — Modified Grashof Number for flat radial power profile and low Re.
Refer to appendix B for derivation.

TABLE 7

Sources of Input Parameters of WARD Preprediction

Case No.	ϵ_1^*	C_1	x_1	x_2	x_3
1 ⁽¹⁾	E	E	E	E	E
2 ⁽¹⁾	E	E	I	I	E
3 ⁽²⁾	I	I	I	I	I
4 ⁽¹⁾	E	E	E	E	E
5 ⁽¹⁾	I	I	I	I	E
6 ⁽¹⁾	I	I	I	I	I
7 ⁽²⁾	I	I	I	I	I
8 ⁽¹⁾	E	E	I	I	E
9 ⁽¹⁾	E	E	E	E	E
10 ⁽¹⁾	I	I	I	I	E
11 ⁽²⁾	I	I	E	E	E

(1) SUPERENERGY is used

(2) ENERGY III is used

E: Extrapolation from experimental data (MIT Mockup)

I: Interpolation from experimental data (MIT Mockup)

Flat Tolerance $\equiv$ Flat to Flat Distance - A

Figure 1 Definition of Flat Tolerance and Rod Alignment Modes of F=1 and F=0

Opt. Condition Case 1 (Level A)	Fab. Geometry	Ana. Parameters
Q = 900 KW M = 61500 lbm/hr $\Delta T_{axial} = 164^{\circ}F$	P/D = 1.077 Dp = 0.519" H = 4.0"	$\epsilon_1^* = 0.333$ $C_1 = 0.38$ x_1, x_2 and x_3 = variable

Figure 2: Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 9 (Level A)		
Q = 660 KW	P/D = 1.077	$\epsilon^* = 0.333$
M = 37000 lbm/hr	$D_p = 0.519"$	$C_1 = 0.38$
$\Delta T_{axial} = 200^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variable

Figure 3: Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 4 (Level A)		
Q = 600 KW	P/D = 1.077	$\epsilon_1^* = 0.333$
M = 37000 lbm/hr	$D_p = 0.519"$	$C_1 = 0.38$
$\Delta T_{axial} = 200^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variable

Figure 4: Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition Case 6 (Level A)	Fab. Geometry	Ana. Parameters
$Q = 132 \text{ KW}$ $M = 7,400 \text{ lbm/hr}$ $\Delta T_{\text{axial}} = 198^\circ\text{F}$	$P/D = 1.077$ $D_p = 0.519"$ $H = 4.0"$	$\epsilon_1^* = 0.254$ $C_1 = 0.35$ $x_1, x_2 \text{ and } x_3 =$ variables

Figure 5 Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Figure 6A ϵ_1^* and x_1 Combinations which Yields
the Same Coolant Temperature

Figure 6B C_1 and x_2 Combinations which Yields the Same Maximum Coolant Temperature (within $\pm 4^\circ\text{F}$)

APPENDIX A

The WARD bundle pre-test predictions of crossflat coolant temperatures at the instrumentation levels of B, C, D, E and F together with the side channel temperatures at the instrumentation level A are illustrated in this Appendix.

Figures A.1 to A.20, categorized in Table A.1 by the case number and the identification of the instrumentation planes, are the pre-test predictions of the crossflat temperatures. On the other hand, Figures A.21 to A.24 are the predictions of the side channel temperatures.

Case No. \ Levels	F	D	C	B	E
1	Fig. A.1	Fig. A.2	Fig. A.3	Fig. A.4	Fig. A.5
9	Fig. A.6	Fig. A.7	Fig. A.8	Fig. A.9	Fig. A.10
4	Fig. A.11	Fig. A.12	Fig. A.13	Fig. A.14	Fig. A.15
6	Fig. A.16	Fig. A.17	Fig. A.18	Fig. A.19	Fig. A.20

TABLE A.1

Categorization of Figures in Appendix A

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 1 (Level F)		
$Q = 900$ KW	$P/D = 1.077$	$\epsilon_1^* = 0.333$
$M = 67,500$ lbm/hr	$D_p = 0.519$ "	$C_1 = 0.38$
$\Delta T_{axial} = 80^\circ F$	$H = 4.0$ "	x_1, x_2 and $x_3 =$ variables

Figure A.1 Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 1 (Level D)		
Q = 900 KW	P/D = 1.077	$\epsilon_1^* = 0.333$
M = 61,500 lbm/hr	$D_p = 0.519"$	$C_1 = 0.38$
$\Delta T_{axial} = 119^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.2 Crossflat Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 1 (Level C)		
Q = 900 KW	P/D = 1.077	$\epsilon_1^* = 0.333$
M = 61,500 lbm/hr	$D_o = 0.519"$	$C_1 = 0.38$
$\Delta T_{axial} = 142^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.3 Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 1 (Level B)		
Q = 900 KW	P/D = 1.077	$\epsilon^* = 0.333$
M = 61,500 lbm/hr	$D_p = 0.519"$	$C_1^1 = 0.38$
$\Delta T_{axial} = 163^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.4 Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 1 (Level F)		
Q = 900 KW	P/D = 1.077	$\epsilon_1^* = 0.333$
M = 61,500 lbm/hr	$D_p = 0.519''$	$C_1 = 0.38$
$\Delta T_{axial} = 163^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.5 Crossflat Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 9 (Level P)		
$Q = 660 \text{ KW}$	$P/D = 1.077$	$\epsilon_f = 0.333$
$M = 37,000 \text{ lbm/hr}$	$D_p = 0.519''$	$C_1 = 0.38$
$\Delta T_{\text{axial}} = 99^\circ\text{F}$	$H = 4.0''$	$x_1, x_2 \text{ and } x_3 = \text{variables}$

Figure A.6 Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 9 (Level D)		
Q = 660 KW	P/D = 1.077	$\epsilon_1^* = 0.333$
M = 37,000 lbm/hr	$D_o = 0.519"$	$C_1 = 0.38$
$\Delta T_{axial} = 146^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.7 Crossflat Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 9 (Level C)		
Q = 660 KW	P/D = 1.077	$\epsilon_1^* = 0.333$
M = 37,000 lbm/hr	$D_p = 0.519"$	$C_1 = 0.38$
$\Delta T_{axial} = 174^\circ F$	H = 4.0"	$x_1, x_2 \text{ and } x_3 =$ variables

Figure A.8 Crossflat Coolant Temperatures of JARD Blanket Test Pre-Test Predictions

Opt. Condition	Tab. Geometry	Ana. Parameters
Case 9 (Level B)		
Q = 660 KW	P/D = 1.077	$\epsilon_1^* = 0.333$
M = 37,000 lbm/hr	$D_p = 0.519"$	$C_1 = 0.38$
$\Delta T_{axial} = 200^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.9 Crossflat Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 9 (Level E)		
Q = 660 KW	P/D = 1.077	$\epsilon_1^* = 0.333$
M = 37,000 lbm/hr	$D_p = 0.519"$	$C_1 = 0.38$
$\Delta T_{axial} = 200^\circ F$	H = 4.0"	x_1, x_2 and x_3 variables

Figure A.10 Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 4 (Level F)		
Q = 600 KW	P/D = 1.077	$\epsilon^* = 0.333$
M = 37,000 lbm/hr	D _p = 0.519"	C _f = 0.38
$\Delta T_{\text{axial}} = 98^\circ\text{F}$	H = 4.0"	x ₁ , x ₂ , x ₃ = variables

Figure A.11 Crossflat Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 4 (Level D)		
Q = 600 KW	P/D = 1.077	$\epsilon^* = 0.333$
M = 37,000 lbm/hr	$D_p = 0.519"$	$C_1 = 0.38$
$\Delta T_{axial} = 145^\circ F$	H = 4.0"	$x_1, x_2, \text{ and } x_3 =$ variables

Figure A.12 Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 4 (Level C)		
Q = 600 KW	P/D = 1.077	$\epsilon^* = 0.333$
M = 37,000 lbm/hr	$D_p = 0.519"$	$C_1 = 0.38$
$\Delta T_{axial} = 173.5^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.13 Crossflat Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 4 (Level B)		
Q = 600 KW	P/D = 1.077	$\epsilon_1^* = 0.333$
M = 37,000 lbm/hr	$D_p = 0.519"$	$C_1 = 0.35$
$\Delta T_{axial} = 200^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.14 Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 4 (Level E)		
Q = 600 KW	P/D = 1.077	$\epsilon^* = 0.333$
M = 37,000 lbm/hr	$D_o = 0.519"$	$C_f = 0.38$
$T_{axial} = 200^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.15 Crossflat Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 6 (Level F)		
Q = 132 KW	P/D = 1.077	$\epsilon^* = 0.254$
M = 7,400 lbm/hr	$D_p = 0.519"$	CI = 0.35
$\Delta T_{axial} = 98^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.16 Crossflat Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition Case 6 (Level D)	Tab. Geometry	Anal. Parameters
$Q = 132 \text{ KW}$ $M = 7,400 \text{ lbm/hr}$ $\Delta T_{\text{axial}} = 145^\circ\text{F}$	$P/D = 1.077$ $D_p = 0.519''$ $H = 4.0''$	$\epsilon_1^* = 0.254$ $C_1 = 0.35$ $x_1, x_2 \text{ and } x_3 =$ variables

Figure A.17 Crossflat Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 6 (Level C)		
Q = 132 KW	P/D = 1.077	$\epsilon_1^* = 0.254$
M = 7,400 lbm/hr	$D_p = 0.519"$	$C_1 = 0.35$
$\Delta T_{axial} = 173^\circ F$	H = 4.0"	x_1, x_2 and $x_3 =$ variables

Figure A.18 Crossflat Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 6 (Level B)		
$Q = 132 \text{ KW}$	$P/D = 1.077$	$\epsilon^* = 0.254$
$M = 7,400 \text{ lbm/hr}$	$D_p = 0.519''$	$C_1 = 0.38$
$\Delta T_{\text{axial}} = 198^\circ\text{F}$	$H = 4.0''$	$x_1, x_2 \text{ and } x_3 =$ variables

Figure A.19 Crossflat Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 6 (Level E)		
$Q = 132 \text{ KW}$	$P/D = 1.077$	$\epsilon_1^* = 0.254$
$M = 7,400 \text{ lbm/hr}$	$D_p = 0.519''$	$C_1 = 0.35$
$\Delta T_{\text{azial}} = 198^\circ\text{F}$	$H = 4.0''$	$x_1, x_2 \text{ and } x_3 =$ variables

Figure A.20 Crossflat Coolant Temperatures of JARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 1 (Level A)		
Q = 900 KW	P/D = 1.077	$\epsilon^* = 0.333$
M = 615,000 lbm/hr	$D_p = 0.519''$	$C_1 = 0.38$
$\Delta T_{axial} = 200^\circ\text{F}$	H = 4.0"	$x_1, x_2 \text{ and } x_3 =$ variables

Figure A.21 Peripheral Coolant Temperatures of WARD Blanket Test Pre-Test Predictions

Opt. Condition Case 9 (Level A)	Fab. Geometry	Ana. Parameters
Q = 660 KW M = 37,000 lbm/hr $\Delta T_{axial} = 200^{\circ}F$	P/D = 1.077 $D_p = 0.519"$ H = 4.0"	$\epsilon_1^* = 0.333$ $C_1 = 0.38$ x_1, x_2 and $x_3 =$ variables

Figure A.22 Peripheral Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition Case 4 (Level A)	Fab. Geometry	Ana. Parameters
$Q = 600 \text{ KW}$ $M = 37,000 \text{ lbm/hr}$ $\Delta T_{\text{axial}} = 200^\circ\text{F}$	$P/D = 1.077$ $D_p = 0.519''$ $H = 4.0''$	$\epsilon^* = 0.333$ $C_1^1 = 0.38$ $x_1, x_2 \text{ and } x_3 =$ variables

Figure A.23 Peripheral Coolant Temperatures of WARD Blanket
Test Pre-Test Predictions

Opt. Condition	Fab. Geometry	Ana. Parameters
Case 6 (Level A)		
$Q = 132 \text{ KW}$	$P/D = 1.077$	$\epsilon_1^* = 0.254$
$M = 7,400 \text{ lbm/hr}$	$D_p = 0.519''$	$C_1 = 0.35$
$\Delta T_{\text{axial}} = 198^\circ\text{F}$	$H = 4.0''$	$x_1, x_2 \text{ and } x_3 =$ variables

~Figure A.24 Peripheral Coolant Temperatures of-WARD Blanket
Test Pre-Test Predictions

APPENDIX B

Modified Grashof Number for flat radial power profile

The modified Grashof Number, G_Y^* , for the skewed radial power profile case has been derived by Khan in the report⁽⁷⁾ to determine the degree of importance of the buoyancy effect. This modified Grashof number is presented as

$$G_Y^* = \frac{\text{buoyant force}}{\text{viscous force}} = \frac{G_Y}{Mf Re^2} \frac{(\hat{Q}_p - 1)}{2} \frac{1}{\gamma L} \quad (B.1)$$

where

$$G_Y = \frac{g\beta(T_M - T_{IN})De^2}{\nu^2}$$

g = gravity

β = thermal expansion coefficient

$\hat{Q}_p$ = peak to average skew

ν = kinematic viscosity

L = bundle length

f = friction factor for smooth tube

M = friction factor multiplier

$\gamma = A_c D_e (\epsilon^* + \alpha^*) / A_n$

The criteria for the dominance of the buoyancy effect over the viscous effect has also been derived in reference (7) as

$$G_Y^* > 0.025 \quad (B.2)$$

For the flat radial power case, $\hat{Q}_p = 1$ and $G_Y^* = 0$. This leads to a conclusion that buoyancy effect is negligible. However, this is not the case for a flat radial power assembly operated at low Reynolds Number. The difference between the flow rates of the different types of subchannels can result in a temperature difference between edge and interior subchannels. The radial temperature difference thus may cause buoyancy effect.

Usually, the flow rate for the interior subchannels is much less than that for the edge subchannels, the ratio between these two flow rates being approximately 0.5 for laminar flow conditions. Consequently, the temperature of the center region of the assembly is much higher than that of the peripheral region. The following derivation of the modified Grashof Number takes into account this phenomenon, but its application is limited to the flat radial power case.

The basic definition of the G_Y^* reported in reference (7) is

$$G_Y^* = \frac{g\beta[T(Z) - \bar{T}(Z)]}{F_D V^2} \quad (24, \text{reference } 7)$$

where

$$F_D \equiv Mf/D_e$$

$T(Z) \equiv$ fluid temperature in the bundle,

$\bar{T}(Z) \equiv$ mean fluid temperature.

In order to predict bouyancy effects in a conservative way, the $[T(Z) - \bar{T}(Z)]$ in the above equation is replaced by $[T_{\max}(L) - \bar{T}(L)]$, where $T_{\max}$ denotes the maximum temperature at plane L and L denotes the plane at the bundle exit where the $[T(Z) - \bar{T}(Z)]$ has a maximum value. Therefore, the G_Y^* becomes

$$G_Y = \frac{g [T_{\max}(L) - \bar{T}(L)]}{F_D V^2} \quad (B.3)$$

The $[T_{\max}(L) - \bar{T}(L)]$ can be obtained by solving the temperature profile of a two region assembly at uniform radial power. One region contains all the interior subchannels and the other region

contains all the edge subchannels. The governing equations for the fluid enthalpy of these two regions are

$$\dot{m}_1 \frac{dH_1}{dz} = A_c \rho (\epsilon_H + \alpha) \frac{H_2 - H_1}{\eta} + QA \quad (B.4)$$

$$\dot{m} \frac{dH_2}{dz} = A_c \rho (\epsilon_H + \alpha) \frac{H_1 - H_2}{\eta} + QA \quad (B.5)$$

where

$\dot{m}$ $\equiv$ mass flow rate of interior subchannels,

$\dot{m}$ $\equiv$ mass flow rate of edge subchannels,

$H_2 \equiv$ fluid enthalpy of edge subchannels,

$H_1 \equiv$ fluid enthalpy of interior subchannels,

$A_c \equiv$ gap area per unit length = p-D

$\eta \equiv$ centroid to centroid distance between region 1 and 2 (i.e., interior subchannel region and edge subchannel region),

$QA \equiv$ heat added to the subchannel per unit length.

The solutions of $H_1(L)$ and $H_2(L)$ in equations (B.4) and (B.5) are as follows

$$H_1(L) = H_{IN} + \left[\frac{2QA}{\dot{m}_1 + \dot{m}_2} + \frac{QAm_1}{(\dot{m}_1 + \dot{m}_2)\alpha} \right] [1 - e^{-\beta\alpha L}] \quad (B.6)$$

$$H_2(L) = H_{IN} + \left[\frac{2QA}{\dot{m}_1 + \dot{m}_2} + \frac{QAm_1}{(\dot{m}_1 + \dot{m}_2)} \right] [1 - e^{-\beta\alpha L}] \quad (B.7)$$

where

$$\alpha = \frac{A_c \rho (\epsilon_H + \alpha)}{\eta} \quad (B.8)$$

$$\beta = \frac{\dot{m}_1 + \dot{m}_2}{\dot{m}_1 + \dot{m}_2} \quad (B.8a)$$

Subtracting equation (B.6) from (B.7), we obtain

$$H_1(L) - H_2(L) = (m_1 - m_2) \left(\frac{QA}{(m_1 + m_2)\alpha} \right) [1 - e^{-\alpha\beta L}] \quad (B-9)$$

Because $T_{\max}(L) - \bar{T}(L)$ is approximately equal to the temperature of the interior subchannels minus the average fluid temperature which can be expressed as

$$\bar{T} = \frac{T_1(L)N_1\dot{m}_1 + N_2\dot{m}_2T_2(L)}{N_1\dot{m}_1 + N_2\dot{m}_2} \quad (B.10)$$

where

$N_1 \equiv$ number of interior subchannels

$N_2 \equiv$ number of edge subchannels

$$\begin{aligned} [T_{\max}(L) - \bar{T}(L)] &= T_1(L) - \frac{T_1(L)N_1\dot{m}_1 + N_2\dot{m}_2T_2(L)}{N_1\dot{m}_1 + N_2\dot{m}_2} \\ &= \frac{N_2\dot{m}_2T_1(L) - N_2\dot{m}_2T_2(L)}{N_1\dot{m}_1 + N_2\dot{m}_2} \\ &= \frac{N_2\dot{m}_2}{N_1\dot{m}_1 + N_2\dot{m}_2} \end{aligned} \quad (B.11)$$

Inserting equation (B.9) into the above equations, we obtain

$$[T_{\max}(L) - \bar{T}(L)] = \frac{N_2\dot{m}_2}{N_1\dot{m}_1 + N_2\dot{m}_2} (m_1 - m_2) \left(\frac{QA}{(m_1 + m_2)\alpha C_p} \right) [1 - e^{-\alpha\beta L}] \quad (B.12)$$

With the above equation, equation (B.3) can be expanded as

$$G_Y^* = \frac{g\beta \left\{ \frac{N_2\dot{m}_2}{N_1\dot{m}_1 + N_2\dot{m}_2} \frac{(m_1 - m_2)}{m_1 + m_2} \left(\frac{QA}{\alpha C_p} \right) [1 - e^{-\alpha\beta L}] \right\}}{F_D V^2} \quad (B.13)$$

In addition, we know

$$R_e = \frac{\rho V D_e}{\mu} \quad (B.14a)$$

$$m_1 = \rho A_1 V X_1, \quad X_1 \equiv \frac{V_1}{V} \quad (B.14b)$$

$$m_2 = \rho A_2 V X_2, \quad X_2 \equiv \frac{V_2}{V} \quad (B.14c)$$

equation (B.13) thus can be reformed as

$$G_Y^* = \frac{g \rho \left\{ \frac{N_2 A_2 X_2}{N_1 A_1 X_1 + N_2 A_2 X_2} \left(\frac{X_2 A_2 - X_1 A_1}{X_2 A_2 + X_1 A_1} \right) \left(\frac{Q A}{2 \gamma C_p} \right) [1 - e^{-\alpha \beta L}] D_e^3 \right\}}{M \rho R_e^2 V} \quad (B.15)$$

where $V \equiv$ kinematic viscosity $\left(\frac{ft^2}{hr} \right)$.

The above modified Grashof Number is derived from the definition which Khan employed. However, whether the criteria stated in equation (B-2) is applicable or not with respect to the above formula (B.15) has not yet been studied. In any case, if the above G_Y^* formula for flat radial power case is much larger than 0.025, it is reasonable to assume that the mixed convection condition will be experienced.

According to the formula derived above, the Modified Grashof Number for case 11 of WARD test can be evaluated with the following information:

$$X_1 = 0.836 \text{ (Novendstern's theory)}$$

$$X_2 = 1.349 \text{ (Novendstern's theory)}$$

$$A_1 = 0.0299 \text{ inch}^2$$

$$A_2 = 0.0628 \text{ inch}^2$$

$$R_e = 934$$

$$QA = \frac{35KW}{126 \times 80.56 \text{ inches}} = 141.2 \frac{\text{Btu}}{\text{hr-ft}}$$

$$v = 0.01324 \text{ (ft}^2/\text{hr)}$$

$$M \approx 3.0 \text{ (M.I.T. Test)}$$

$$n = 0.822 \text{ inch}$$

$$E^* = \frac{E_H}{VD_e \lambda_a} = 0.0781$$

$$\lambda_a = 0.247$$

$$\beta = 1.5278 \times 10^{-4} \left(\frac{1}{Po} \right)$$

$$f = \frac{64}{Re}$$

The G_Y^* thus equals to

$$G_Y^* = \frac{4.17 \times 10^8 \times 1.5278 \times 10^{-4} \left[\frac{0.836 \times 0.0299 - 1.349 \times 0.0628}{0.836 \times 0.0299 + 1.349 \times 0.0628} \right] \left[\frac{141.2}{2 \times 0.588 \times 0.307} \right]}{3 \times 64 \times 928 \times 0.1324^2}$$

$$\times \left(\frac{0.134}{12} \right)^3$$

$$= 0.129$$

(B.15)

Since this number is much larger than 0.025, it is believed that the buoyancy effect in this case can not be neglected.