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COOLANT MIXING IN LMFBR ROD BUNDLES AND
OUTLET PLENUM MIXING TRANSIENTS

Progress Report

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MIT Coolant Mixing in LMFBF Rod Bundles Project

A. Quarterly Progress Reports (Available from:
National Technical Information Service
U.S. Department of Commerce
Springfield, VA 22151)

COO-2245-1	Period June 1, 1972 - November 30, 1972
COO-2245-2	Period December 1, 1972 - February 28, 1973
COO-2245-3	Period March 1, 1973 - May 31, 1973
COO-2245-6	Period June 1, 1973 - August 31, 1973
COO-2245-7	Period September 1, 1973 - November 30, 1973
COO-2245-8	Period December 1, 1973 - February 28, 1974
COO-2245-10	Period March 1, 1974 - May 31, 1974
COO-2245-13	Period June 1, 1974 - August 31, 1974
COO-2245-14	Period September 1, 1974 - November 30, 1974
COO-2245-15	Period December 1, 1974 - February 28, 1975
COO-2245-23	Period March 1, 1975 - May 31, 1975
COO-2245-25	Period June 1, 1975 - August 31, 1975
COO-2245-26	Period September 1, 1975 - November 30, 1975
COO-2245-28	Period December 1, 1975 - February 29, 1976
COO-2245-30	Period March 1, 1976 - May 31, 1976
COO-2245-31	Period June 1, 1976 - August 31, 1976
COO-2245-34	Period September 1, 1976 - November 30, 1976
COO-2245-38	Period December 1, 1976 - February 28, 1977
COO-2245-50	Period March 1, 1977 - May 31, 1977
COO-2245-53	Period June 1, 1977 - August 31, 1977
COO-2245-60	Period September 1, 1977 - November 30, 1977
COO-2245-63	Period December 1, 1977 - February 28, 1978
COO-2245-64	Period March 1, 1978 - May 31, 1978
COO-2245-65	Period June 1, 1978 - August 31, 1978
COO-2245-66	Period September 1, 1978 - November 30, 1978
COO-2245-69	Period December 1, 1978 - February 28, 1979
COO-2245-70	Period March 1, 1979 - May 31, 1979
COO-2245-71	Period June 1, 1979 - August 31, 1979
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- * Revised - See Section B.2

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COOLANT MIXING IN LMFBR ROD BUNDLES AND
OUTLET PLENUM MIXING TRANSIENTS
Contract AT(11-1)-2245

Quarterly Progress Report

The work of this contract has been divided into the following Tasks:

- TASK I: BUNDLE GEOMETRY (WRAPPED AND BARE RODS)
- TASK IA: Assessment of Available Data
 - TASK IB: Experimental Bundle Water Mixing Investigation
 - TASK IC: Experimental Bundle Peripheral Velocity Measurements (Laser Anemometer)
 - TASK ID: Analytic Model Development - Bundles
- TASK II: SUBCHANNEL GEOMETRY (BARE RODS)
- TASK IIA: Assessment of Available Data
 - TASK IIB: Experimental Subchannel Water Mixing Investigation
 - TASK IIC: Experimental Subchannel Local Parameter Measurements (Laser Anemometer)
 - TASK IID: Analytic Model Development - Subchannels
- TASK III: LMFBR OUTLET PLENUM FLOW MIXING
- TASK IIIA: Analytical and Experimental Investigation of Velocity and Temperature Fields
- TASK IV: THEORETICAL DETERMINATION OF LOCAL TEMPERATURE FIELDS IN LMFBR FUEL ROD BUNDLES

TASK I: BUNDLE GEOMETRY (WRAPPED AND BARE RODS)

I.A. Assessment of Available Data

A major review of bundle pressure drop data was made prior to formulating models for subchannel pressure drop. This review and the resulting correlations are reported under Task I.D.3.

Two literature reviews were made and invited papers written in collaboration with others on "Experimental Techniques" and "Hydraulic Characteristics" of LMFBR Wire-Wrapped Rod Bundles. These papers are to appear in a special issue of Nuclear Engineering and Design and are listed on page ix of this report.

I.B. Experimental Bundle Water Investigations

I.B.1 61 Pin Fuel Bundles - Turbulent Flow

(Paul Symolon)

No additional progress has been made on the 61 pin bundle in the turbulent flow regime. A topical report will be issued later in the next quarter presenting the data and analysis.

I.B.2 61 Pin Fuel Bundles - Laminar and Transition Flow

(Paul Symolon)

No additional progress has been made on the 61 pin bundle in the laminar and transition flow regime. A topical report will be issued next quarter presenting the data and analysis.

I.B.3. 217 Pin Fuel Bundle

(Paul Symolon)

Flow split measurements have continued on the 217 pin bundle. The data taken to date is presented in Figure 1. The value calculated from the turbulent flowsplit correlation is drawn in. It is observed that the data approaches this value asymptotically. Reasonable agreement is obtained for $Re > 6,000$, and for $Re > 10,000$. The agreement is quite close. For $Re < 6,000$, the data diverges from the turbulent correlation value as expected, since this data is in the transition regime. The transition data is following the trend predicted by the laminar flowsplit correlation of Chiu and Hawley.

It has been observed that the edge flowsplit data exhibits a variation around the bundle periphery due to the effect of the wire wrap orientation. This orientation is shown in Figure 2. The six faces of the bundle (labeled by face I.D. numbers 1-6) correspond to six different wire wrap orientations. Figure 3 shows the channel flow rates measured for the $Re = 8400$ run around the bundle periphery. Each group of 8 subchannels corresponding to a particular face should have the same velocity, or equivalently, the same flow rate. The mean flow for each face is drawn in, along with an error bar representing a 68% confidence interval (plus or minus one standard deviation about the mean). A definite trend in the variation is seen to exist.

The data of Figure 1 averages together any variation of X_2 with bundle periphery. If we consider the existence of 6 types of edge subchannels, 6 flowsplit parameters can be calculated for the edge region. Represent these by

$$X_{2i} = \frac{V_{2i}}{V_b} \quad i = 1, 2, \dots, 6$$

where i corresponds to a particular face I.D. number. These flowsplit parameters were calculated using all the data taken in the fully turbulent regime ($Re > 6,000$) and are presented in Figure 4, along with an error bar representing a standard deviation about the mean. A trend similar to that of Figure 3 (in which data at one Reynolds number only was plotted) is exhibited. Note that a high velocity is obtained when the wire is in the subchannel (Face 1 and 2), and a low velocity is obtained when the wire is out of the subchannel (Face 4 and 5). Face 3 and Face 6, corresponding to half of a wire in the subchannel, gives an intermediate velocity, both nearly the same.

In the following quarter, flowsplit measurements will continue into the laminar flow regime, and the mixing experiment will finally be performed.

FIGURE 1

Flow Split Parameters vs Reynolds Number

□ X_1 Data

○ X_2 Data

△ Mean of X_2 Data for $Re > 6,000$ with 95% Confidence Interval (Mean = 1.042)

Mean of X_1 Data for $Re > 6,000$ with 95% Confidence Interval (Mean = .989)

FIGURE 2

Wire Wrap Orientation and
Face ID Numbers

I.B.4. 61 Pin Blanket Bundles

Work on these bundles was completed and reported in a series of topical reports by C. Chiu.

I.B.5. Shaved Wire 61 Pin Blanket Bundle

(S. F. Wang)

No work was done on this task during this quarter. The available data will be reported and analyzed in the spring, 1980.

I.B.6. 37 Pin ($P/D = 1.15$) Bundle

(Kevin Burns)

Initially, the flowmeter used to measure large flow rates through the 37-pin test section was suspected to be in error due to some kind of internal blockage. This suspicion was verified by an experiment in which the flowmeter was checked against a measurement of volumetric change per unit time in the source tank, which showed the flowmeter to be in error by as much as 30-40%. The cost of a new flowmeter was quoted as less than that of cleaning and recalibrating the old one, hence a new flowmeter was ordered and has since been delivered. During the time when the flowmeter was on order, large flow rates were measured by measuring the pressure difference across a flow orifice plate and using a correlation between this pressure difference the flow rate.

In anticipation of the need for flow split measurements, a preliminary experiment was designed and performed in order to determine the radial pressure distribution at the bundle exit. The purpose of this experiment was to evaluate the practicality of various flow split techniques, and specifically to determine whether the presently used, but tedious isokinetic technique is really necessary. The experimental apparatus consisted of a static pressure tap located approximately 5/8" above the bundle exit. This distance corresponds to the location of the pressure taps on the flow collector used in flow split experiments. The apparatus was found to be too crude, in that the resolution of small pressure differences was not adequate enough to make any meaningful observations with respect to flow split measuring techniques. Thus, more precise instrumentation is needed if knowledge of the detailed characteristics of the bundle exit pressure distribution is desired.

Finally, reproducibility of the pressure drop data of a previous experimenter on the same test section is presently being checked. Results of measurements to date suggest that differences in experimental procedures can cause rather disparate results for different experimenters. Thus, an effort is being made to determine the optimal method and to perform the pressure drop experiments accordingly.

I.B.7. Experimental Method for Flow Split
 Determination

James Hawley

Work has resumed on a report detailing the experimental method used in determining the flow split parameters. This report will be finished within the next month.

I.B.8 Experimental Method for Pressure Drop
 Determination

(Kevin Burns)

The procedure for subchannel pressure drop experiments will be documented in a Pressure Drop Laboratory Manual. This manual will describe the accepted technique (see Task I.B.6), based on the investigation currently being done into the validity of various procedures. The manual will also include an analysis of the errors involved in experimental data collection and data reduction. The document is expected to be completed by January 31, 1980.

I.C. Experimental Bundle Peripheral Velocity
Measurements (Laser Anemometer)

No work is planned on this Task in the near
future.

I.D. Analytical Model Development - Bundles

I.D.1 Steady State SUPERENERGY

(Kerry Basehore)

This effort has been directed at producing a second revision of the code SUPERENERGY tailored for whole case application. The work has been continued by Kerry Basehore at BNL in part under subcontract from HEDL for FFTF application. The documentation summarizing this work is under preparation and will be issued next quarter.

I.D.2. Transient TRANSENERGY

(Stuart Glazer and Tom Greene)

Transient versions of the energy code called TRANSENERGY S and M, for single and multiple bundle analysis, have been written and tested. The documentation summarizing this work is under preparation and will be issued in the spring, 1980.

I.D.3 Models for Forced Convection Analysis
 (Pressure Drop, Flow Split and Mixing)

(James T. Hawley)

Correlations have been obtained for sub-channel and bundle friction factors and flowsplit parameters for laminar, transition and turbulent longitudinal flows in wire wrap spaced hexagonal arrays. These results were obtained from pressure drop models of flow in individual subchannels. All the details and results of these models and correlations are contained in the paper "Subchannel and Bundle Friction Factors and Flowsplit Parameters for Laminar, Transition and Turbulent Longitudinal Flows through Wire Wrap Spaced Hexagonal Rod Arrays," which has been submitted for presentation at the ANS Topical Meeting on Thermal/Hydraulics, Saratoga, October 1980. In what follows, a brief summary of this paper will be given.

The approach of the paper is to consider laminar, turbulent and transition flow individually. For laminar flow, models were developed for pressure drops of individual subchannels. The results are:

$$\Delta P_{iL} = \left(\frac{C_i}{Re_i} + \frac{C_{\alpha i}}{Re_i} \frac{A_{ri}}{A_{si}} \frac{De_i}{H} \left(1 - \frac{D_w}{2De_i} \right) \right) \frac{L_b}{De_i} \frac{\rho V_i^2}{2}$$

where

$$C_1 = (-336(P/D - 1)^2 + 340(P/D - 1) + 40) \frac{A'}{A} \frac{De_1}{De_1'}$$

$$C_2 = (-318.2(P/D-1)^2 + 290.0(P/D-1) + 37.4) \frac{A'}{A} \frac{De_2}{De_2'}$$

(primed quantities refer to bare-rod values).

$$C_{\alpha 1} = 8700 / (H/D)^{.33}$$

$$C_{\alpha 2} = 8300 / (H/D)^{.16}$$

From these models, expressions for the subchannel friction factors were identified and expressions for the flowsplit parameters were derived by equal pressure drops and continuity. Bundle friction factors were found by the equal pressure drop assumption and the preceding expressions.

For turbulent flow, an existing pressure drop model due to Chiu (TR-56) for flow in edge subchannels was extended and the resulting edge subchannel friction factor identified. The change consisted of specifying the geometrical dependence of a factor, C_{f2} , which was not specified in the original derivation (TR-56); this was done by considering the effect of flow slipping over the wire on edge subchannel velocity distributions. The resulting edge subchannel friction factor expression is:

$$f_{2T} = (1 + (1.9n \frac{V_t}{V_a} \text{ gap})^2)^{1.375} (1 + C_c^2 \frac{De_2}{H} \frac{P^2}{\pi^2 P^2 + H^2}) \frac{C_T}{Re_2^{.15}}$$

Using the expressions for flowsplit parameters of TR-56 and the equal pressure drops assumption, the interior subchannel and bundle friction factors were obtained.

For transition flow, a model was proposed which superposes any one of the three pairs of the

friction factor components obtained earlier. That is, transition flow friction factors are assumed to have the form:

$$f_i = [(f_{iL})^{\ell} + (f_{iT})^{\ell}]^{1/\ell}$$

where the subscripts L and T denote the laminar and turbulent flow conditions, and the exponent ℓ dictates the pattern of superposition of the laminar and turbulent components. Phenomenologically, the form of the above equation is dictated by the smooth and continuous transition which occurs between laminar and turbulent flows. As written, the assumption embodied in this equation is that the pattern of superposition of the components is the same for the different subchannels and the bundle. This is not precisely correct as shown in TR-42. However, since there is very little other subchannel friction factor data available, this assumption is justified. Furthermore, since the shapes of the data curves in TR-42 are not greatly different, this assumption will not introduce much uncertainty into subchannel friction factor calculations. Values for the flowsplit parameters were obtained by using the transition flow values for the subchannel friction factors in the equal pressure drops assumption.

The various constants which appear in the final expressions were calibrated against the available literature bundle friction factor data. Results of this procedure show that for friction factors, 97% of the data points fall within $\pm 16\%$ of the predicted values while for flowsplit parameters, 100% of the data points are within $\pm 10\%$ of the predicted values.

I.D.4.0 Models for Mixed Convections - Overall

(Song-Feng Wang)

Overall Task Objective

The objective of this work is to develop computational methods for analyzing steady state and transient fluid flow and heat transfer processes in sodium-cooled fast reactors under mixed convection conditions. Current emphasis is on development of analytical steady state pressure drop and heat transfer models for use in the more sophisticated computer programs. These steady state fluid flow and heat transfer models will be validated by experimental work and used as a basis for further development in transient analysis.

Model Development

Before actually doing model development, an important criterion should be determined a priori - the criterion for onset of the buoyant effects. This criterion represents the conditions where the buoyant effects become important and the flow behavior can no longer be described by forced convection. It is shown in Figure 1 as the flow regime boundary between forced and mixed convection. In Figure 1, the other important criterion for model development is the boundary between mixed and natural convection where buoyant effects are so dominant that the fluid flow and heat transfer processes can be treated as natural circulation.

Having these two criteria determined, the problem of developing a model for mixed convection can be attacked either analytically or experimentally. This is true for all three levels of geometric complexity in sodium-cooled reactors: subchannel, assembly, and reactor core.

In subchannels, the buoyant effects influence the relative flows between subchannel regions. Local flow recirculation may occur when buoyant effects become significant.

In fuel assemblies, the buoyant effects influence the relative flows between subchannels or regions in the assembly. Under certain conditions, gross downflow may occur in certain regions.

In a reactor core, the buoyant effects influence the relative flow of assemblies. Flow reversal may occur in some of the assemblies under extreme conditions.

Table 1 depicts the important parameters to be determined under mixed convection conditions. For uniform power distribution across an assembly, the coolant interactions between adjacent subchannels are negligible. The thermal hydraulic analysis of the assembly can be reduced to the analysis of a single subchannel. No interchanges between adjacent subchannels are taken into account. However, for an assembly subject to power skew, the flow redistribution due to the presence of buoyant forces

become significant and the interactions between adjacent subchannels are no longer negligible. The subchannel analysis for this case is appropriate only when the boundary conditions for velocity and temperature at subchannel interfaces are known.

In Table 1, the analytic methods are further divided into distributed parameter and lumped parameter analysis. In the first method, detailed velocity and temperature distribution are obtained either analytically or numerically. From these distributions, three important parameters can be deduced: friction factor and heat transfer coefficients which can be used for lumped parameter analysis, and the criteria for the onset of local/gross flow reversal. The drawback of this local method is it requires excessive core storage and computational time for large region problems which makes it impractical for large assembly and whole core analysis. On the other hand, the lumped parameter method can analyze large regions with reasonable core storage and computational time by lumping subregions into points. It is obvious that this homogenization in space, although it greatly simplifies the analysis, makes detailed calculations impossible. This disadvantage of lumped parameter analysis reveals the significance of results from distributed parameter analysis: by having detailed velocity and temperature distributions available, we can set the appropriate lumping size for lumped parameter analysis without jeopardizing the final results.

The parameters listed in Table 1 can be obtained for different flow and power combinations, as shown in Table 2. There are at least eight cases to be considered for subchannel and assembly analysis: upflow vs. downflow, square lattice vs. triangular lattice and interior subchannels vs. edge subchannels. In the present analysis, although we are primarily concerned with sodium-cooled reactors, we do incorporate the water system and square lattice assembly for the ease of conducting LMFBR simulation experiments and potential future application in water-cooled reactors.

The model development has been started on a subchannel basis using the distributed parameter method. The equations being solved are the continuity, momentum and energy conservation equations. All physical properties are considered to be constant except the coolant density in the body force term of the momentum equation.

The general conservation equations with constant physical properties can be written as follows:

$$\text{Continuity} \quad \nabla \cdot \vec{V} = 0$$

$$\text{Momentum} \quad \rho \frac{D\vec{V}}{Dt} = -g_c \nabla p + \mu \nabla^2 \vec{V} + \vec{B}$$

$$\text{Energy} \quad \rho C_p \frac{DT}{Dt} = k_w \nabla^2 T + W_i + \mu \Phi$$

where W_i = rate of energy generation per unit volume

$\mu\Phi$ = rate of energy increase per unit volume
by viscous dissipation

$\vec{B}$ = body force on element per unit volume

In cylindrical coordinates

$\vec{v} \equiv$ velocity field = (v_r, v_θ, v_z)

$\vec{B} \equiv (B_r, B_\theta, B_z)$

$$\nabla \equiv \frac{1}{r} \frac{\partial}{\partial r} r + \frac{1}{r} \frac{\partial}{\partial \theta} + \frac{\partial}{\partial z}$$

$$\nabla^2 = \frac{\partial}{\partial r} \left(\frac{1}{r} \frac{\partial}{\partial r} r \right) + \frac{1}{r^2} \frac{\partial^2}{\partial \theta^2} + \frac{\partial^2}{\partial z^2}$$

$\frac{D}{Dt}$ = substantial derivative

$$= \frac{\partial}{\partial t} + v_r \frac{\partial}{\partial r} + \frac{v_\theta}{r} \frac{\partial}{\partial \theta} + v_z \frac{\partial}{\partial z}$$

When the gravitational force is the only force acting on the system,

$$\vec{B} \equiv B_z = -g\rho \frac{\partial h}{\partial z}$$

where h is the elevation of the system and z is the direction of flow. For upflow, z and h are in the same direction, $B_z = -g\rho$. For downflow, z and h are in opposite directions, $B_z = g\rho$. Here, the density is represented as $\rho = \rho_0 (1 - \beta(T - T_0))$.

Let ρ_o , μ_o , k_{wo} and C_{po} represent the characteristic reference magnitude of the corresponding property. Let U , D and $(T_1 - T_o)$ represent some convenient characteristic velocity, length, and temperature difference in the system (Usually, average velocity and hydraulic diameter are used for U and D). We then define the following dimensionless variables.

$$\vec{v}^* \equiv \frac{\vec{v}}{U} = (v_r^*, v_\theta^*, v_z^*) = \text{dimensionless velocity}$$

$$p^* \equiv \frac{P}{\rho_o U^2} = \text{dimensionless pressure}$$

$$t^* \equiv \frac{tU}{D_h} = \text{dimensionless time}$$

$$T^* \equiv \frac{T - T_o}{T_1 - T_o} = \text{dimensionless temperature}$$

$$\rho^* \equiv \frac{\rho}{\rho_o} = \text{dimensionless density}$$

$$\mu^* \equiv \frac{\mu}{\mu_o} = \text{dimensionless viscosity}$$

$$C_p^* \equiv \frac{C_p}{C_{po}} = \text{dimensionless specific heat}$$

$$r^*, z^* \equiv \frac{r}{D_h}, \frac{z}{D_h} = \text{dimensionless coordinates}$$

Writing the three conservation equations in terms of these dimensionless variables we obtain, on rearrangement

$$\nabla^* \cdot \vec{v}^* = 0$$

$$\frac{D^* v^*}{Dt^*} = -g_c \nabla^* p^* + \frac{1}{Re} \nabla^{*2} \vec{v}^* \pm \left(\frac{Gr}{Re^2} - \frac{1}{Fr} \right)$$

$$\frac{D^* T^*}{Dt^*} = \frac{1}{RePr} \nabla^{*2} T^* + \frac{D}{(T_1 - T_o) \rho_o C_{po} U} W_i + \frac{Ec}{Re} \phi^*$$

where in the last term of momentum equation, the minus sign applies for upflow and the plus sign applies for downflow. Note $\rho^* = \mu^* = C_p^* = k_w^* = 1$ for constant properties and there are five dimensionless groups appearing in these three equations.

$$Re \equiv \text{Reynolds number} = \frac{\text{inertial forces}}{\text{viscous forces}} = \frac{\rho_o U^2 / D_h}{\mu_o U / D_h^2} = \frac{\rho_o D_h U}{\mu_o}$$

$$Pr \equiv \frac{\text{mass eddy diffusivity}}{\text{thermal eddy diffusivity}} = \frac{\mu_o C_{po}}{k_{wo}} \quad (\text{Prandtl number})$$

$$Gr \equiv \text{Grashof number} = \frac{(\text{buoyancy forces})(\text{inertial forces})}{(\text{viscous forces})^2} =$$

$$\frac{g \beta \rho_o^2 (T - T_o) D_h^3}{\mu_o^2}$$

$$Fr \equiv \text{Froude number} = \frac{\text{inertial forces}}{\text{gravity forces}} = \frac{\rho_o U^2 / D_h}{\rho_o g} = \frac{U^2}{g D_h}$$

$$E_c \equiv \text{Eckert number} = \frac{\text{viscous energy}}{\text{convective energy}} = \frac{U^2}{(T_1 - T_o) C_p}$$

As we neglect heat source and viscous dissipation, these dimensionless conservation equations become

$$\nabla^* \cdot \vec{v}^* = 0$$

$$\frac{D^* \vec{v}^*}{Dt^*} = -g_c \nabla^* p^* + \frac{1}{Re} \nabla^{*2} \vec{v}^* \pm \left(\frac{Gr}{Re^2} - \frac{1}{Fr} \right)$$

$$\frac{D^* T^*}{Dt^*} = \frac{1}{RePr} \nabla^{*2} T^*$$

These are the equations used in subchannel and assembly analyses.

L A M I N A R	T R A N S I T I O N	T U R B U L E N T
FORCED		CONVECTION
MIXED		CONVECTION
NATURAL		CONVECTION

FIGURE 1. FLOW REGIME BOUNDARIES TO BE DETERMINED

TABLE 1. PARAMETERS TO BE DETERMINED UNDER MIXED CONVECTION CONDITIONS

		SUBCHANNEL	ASSEMBLY	CORE
THEORY	DIST. PARM.	A) POINTWISE VELOCITY AND TEMP, DISTRIBUTIONS B) FRICTION FACTOR AND HEAT TRANSFER COEFF. C) CRITERIA OF ONSET OF LOCAL FLOW RECIRCULATION - CONDITIONS - LOCATION	A) POINTWISE VELOCITY AND TEMP, DISTRIBUTIONS B) FRICTION FACTOR AND HEAT TRANSFER COEFF. C) CRITERIA OF ONSET OF LOCAL/GROSS FLOW REVERSAL - CONDITION - LOCATION	---
	LUMP PARM.	---	A) SUBCHANNEL AVERAGE VELOCITY AND TEMP B) CRITERIA OF ONSET OF GROSS FLOW REVERSAL - CONDITIONS - LOCATION	A) ASSEMBLY AVERAGE VELOCITY AND TEMP B) CRITERIA OF ONSET OF GROSS FLOW REVERSAL - CONDITIONS - LOCATION
EXPERIMENTS		VERIFY A), B), AND C)	VERIFY A), B), AND C)	VERIFY A) AND B)

TABLE 2.1 FLOW CONDITIONS TO BE CONSIDERED UNDER MIXED CONVECTION CONDITIONS				
		SUBCHANNEL	ASSEMBLY	CORE
THEORY	DIST. PARM.	A) ADDING VS OPPOSING B) FULLY-DEVELOPED VS DEVELOPING C) $\triangle$ LATTICE VS $\square$ LATTICE D) Na SYSTEM VS H ₂ O SYSTEM E) INTERNAL VS EDGE	A), B), C), D) E) INTERCONNECTING PARALLEL SUBCHANNELS	---
	LUMP PARM.		A), B), C), D) E) INTERCONNECTING PARALLEL SUBCHANNELS	A), B), C), D) E) INTERCONNECTING PARALLEL CHANNELS F) ISOLATED PARALLEL CHANNELS
EXPERIMENT		A), B), C), D) E) INTERCONNECTING PARALLEL SUBCHAN SUBCHANNELS	A), B), C), D), E)	A), B), C), D), F)

TABLE 2.2 POWER DISTRIBUTIONS TO BE CONSIDERED UNDER M. C. CONDITIONS				
		SUBCHANNEL	ASSEMBLY	CORE
THEORY	DIST PARM	A) UNIFORM POWER - CONST. Q'' AT CLAD SURFACE - ZERO Q'' AT ISOTACH - ZERO/CONST. Q'' AT DUCT WALL	A) POWER SKEW - CONST. $Q''/Q''(\theta)$ AT CLAD SURFACE - ZERO/CONST. Q'' AT ISOTACH - ZERO/CONST. Q'' AT DUCT WALL	---
	LUMP PARM	---	A) POWER SKEW (NEED SHAPE FACTOR FROM DIST PARM ANALYSIS)	A) POWER SKEW - Q'' ACROSS BUNDLE
EXPERIMENT				

I.D.4.1 Models for Mixed Convection - Subchannels
(Song-Feng Wang)

The analytical solutions of subchannel velocity and temperature distributions in bare rod bundles under mixed convection conditions have been obtained for the following eight cases:

	Upflow	Downflow
Triangular Lattice	1) interior subchannel 2) edge subchannel	5) interior 6) edge
Square Lattice	3) interior 4) edge	7) interior 8) edge

In these calculations the following assumptions are made:

- 1) the heat flux at the inner surface of clad is constant.
(Thermal two region problem)
- 2) the coolant is thermally and hydrodynamically fully-developed laminar flow
- 3) all physical properties are considered to be constant except the coolant density in the body force term
- 4) no heat transfer and shear forces at the isotach
- 5) no heat transfer across the duct wall
- 6) no viscous dissipation, pressure work, and axial conduction

- 7) no heat source in clad and coolant
- 8) the coolant field is at steady-state

From the calculated velocity and temperature distributions, we can further deduce the friction factors, heat transfer coefficients, criteria for the onset of flow recirculation and the criteria for transition from laminar to turbulent flow.

The calculated results indicate that, as buoyant effects become important, the friction factor and heat transfer coefficient increase from their isothermal values for upflow and decrease for downflow. The increments/decrements of these two parameters are more pronounced for large P/D bundles than for small P/D bundles. The criteria of the onset of flow recirculation are represented by the critical Rayleigh number, $(Ra)_c$. For large P/D , $(Ra)_c$ is small, i.e., flow recirculation occurs at low Ra . The criteria for transition from laminar to turbulent flow are represented by critical Reynolds number, $(Re)_c$. This $(Re)_c$ also decreases for increasing P/D , i.e., flow becomes turbulent at low Re for large P/D bundle.

A detailed description of the calculational procedures and the results are given in a paper to be presented at BNL February 1980. The paper will be included in the next progress report.

Future Work

It was realized, during the calculation, that the heat transfer between assemblies is important under mixed convection conditions. Therefore, one of our assumptions is invalid: no heat transfer across duct wall. The effects of this heat transfer on the resultant velocity and temperature fields are under investigation and the results will be included in the BNL paper.

I.D. 4.2 Models for Mixed Convection - Assembly
(Paul Symolon)

In this quarter, some preliminary calculations were made to roughly estimate the temperatures and velocities which will be established in a test section simulating interassembly heat transfer in the mixed and natural convection flow regime.

To simplify the analysis, attention was focused on two neighboring channels, which are isolated from the rest of their environment, and exchange heat only with each other. Each channel is assumed to have a single lumped temperature at any axial level. The development of the model will briefly be outlined.

The geometry of the test section considered is given in Figure 1. The rod in channel 1 has a specified power q' , and for channel 2 the rod operates at zero power. The channel pressure drop has two components, frictional losses and hydrostatic pressure loss.

$$\Delta P = \Delta P_f + \Delta P_{HYDRO}$$

where

$$\Delta P_f = f \frac{L}{D} \frac{\rho V^2}{2}, \quad f = \frac{C_f}{Re}$$

$$= \frac{\mu C_f L}{2 D^2} V$$

$$\Delta P_{HYDRO} = \rho \int_0^L P dz$$

Equating pressure drops for each channel gives:

$$\begin{aligned}
 V_1 - V_2 &= \frac{2D^2g}{\mu c_p L} \int_0^L (\rho_2 - \rho_1) dz \\
 \rho_2 - \rho_1 &= \rho_0 \beta (T_1 - T_2) \\
 V_1 - V_2 &= \frac{2D^2g\rho_0\beta}{\mu c_p L} \int_0^L (T_1 - T_2) dz \quad (1)
 \end{aligned}$$

The heat transfer is modeled by an overall heat transfer coefficient:

$$q = UA(T_1 - T_2)$$

where

$$U = \left[\frac{1}{h_1} + \frac{\Delta x}{K} + \frac{1}{h_2} \right]^{-1}$$

A laminar flow correlation was used for h_1 . An energy balance on the two control volumes of Figure 2 gives:

$$\dot{m}_1 c \frac{dT_1}{dz} = -UP(T_1 - T_2) + q'$$

$$\dot{m}_2 c \frac{dT_2}{dz} = +UP(T_1 - T_2)$$

Solving these equations with boundary conditions $T_1 = T_2 = T_0$ at $z = 0$ yields

$$T_1 - T_2 = \frac{Q}{K_1 + K_2} \left[1 - e^{-(K_1 + K_2)z} \right] \quad (2)$$

where: $Q = \frac{q'}{\dot{m}_i c}$, $K_i = \frac{U P}{\dot{m}_i c}$ $i = 1, 2$

For an adiabatic wall ($U = 0$) the solution is:

$$T_1 - T_2 = Q Z \quad (3)$$

For the case of adiabatic walls, Eq. 3 into Eq. 1 gives, in dimensionless form:

$$\begin{aligned} \frac{V_1}{\bar{V}} &= \frac{1}{2} \left[1 + \left\{ 1 + \frac{Ra}{Re, Pr} \frac{Z}{c_s} \frac{L}{D} \frac{V_1}{\bar{V}} \right\}^{1/2} \right] \\ \frac{V_2}{\bar{V}} &= 2 - \frac{V_1}{\bar{V}} \end{aligned} \quad (4)$$

where $\bar{V} = \frac{1}{2}(V_1 + V_2)$

For the case accounting for heat transfer; Eq. 2 into Eq. 1 gives:

$$\frac{V_1 - V_2}{\bar{V}} = \frac{Ra}{Pr Re} \frac{Z}{c_s} \frac{L}{D} \left\{ \frac{1/L}{K_1 + K_2} \left(1 + \frac{1/L}{K_1 + K_2} \left(e^{-(K_1 + K_2)L} - 1 \right) \right) \right\} \quad (5)$$

Once velocities are known, temperatures are obtained from Equations 2 and 3. For the adiabatic case the velocities can be calculated readily from Equation 5, however must be solved by iteration since the right hand side also depends on velocity.

Figure 3 gives the results of a calculation with $q' = 2400 \frac{\text{BTU}}{\text{Ft Hr}}$ (close to the maximum power for the planned experiment). The channel axial temperature rises are given as a function of the average Reynolds number. For higher Reynolds numbers the hot channel Δt is about 25°F, and the cold channel Δt is 0°F. The adiabatic and heat transfer models give the same results. As the Reynolds number goes down, the difference between the two models becomes larger. At a Reynolds number of about 200 the analysis of the heat transfer model yields a negative V_2 . Once this occurs the model is no longer valid, since the boundary condition imposed on channel 2 is then incorrect. However, this point can be considered an "onset of flow reversal" since we have passed the point at which $V_2 = 0$.

Figure 4 shows the prediction of "onset of flow reversal" for the two models. For the adiabatic case we simply put $V_2 = 0$ in Equation 4, and solve for q' as a function of $\bar{V}$. For the heat transfer calculation trial and error must be used. As expected, as flow is decreased, or as power is increased flow reversal occurs first for the adiabatic case. The heat transfer model is expected to give a conservative result since a laminar forced convection correlation was used for h . Heat transfer due to free convection should become important at such low Reynolds numbers.

ΔT for the cold channel will be measurable for high rod powers, and low Reynolds numbers. In terms of the experimental design, this implies small hydraulic diameters should be used, with small fluid velocities.

The mechanism which causes flow reversal can be explained in terms of the superposition of the frictional and hydrostatic pressure drop components into the total channel pressure drop. The imposed pressure drop across each channel must be the same. The hydrostatic component depends on the channel axial temperature distribution, and will be larger for the cold channel. If the imposed ΔP is larger than the channel hydrostatic ΔP , upflow will occur. If they are equal no flow will occur. If hydrostatic ΔP is larger than imposed ΔP , downflow will occur. This is illustrated in Figure 5. The hydrostatic components are shown for each channel. If the imposed ΔP is from level 1 to level 2, each channel will have upflow. As ΔP is decreased to ΔP_{13} the hot channel will have upflow, and the cold channel no flow. If ΔP is decreased to ΔP_{14} , the hot channel will have upflow, and the cold channel downflow. If ΔP is decreased further to ΔP_{15} , both channels will have downflow. The model given here is only valid up for imposed $\Delta P \geq \Delta P_{13}$, as the boundary conditions are otherwise invalid.

From these preliminary calculations, it appears that significant interassembly heat transfer will occur only for low Reynolds numbers ($Re \lesssim 700$) and high rod powers ($q' \approx 2,000 \frac{\text{BTU}}{\text{Ft hr}}$). Also, the axial

FIGURE 1 SIDE VIEW TOP VIEW

$$D = .5''$$

$$P/D = 1.25$$

FIGURE 2

CONTROL VOLUME

FIGURE 4
PREDICTION OF ONSET OF FLOW
REVERSAL FOR TWO MODELS.

FIGURE 5

Superposition of Pressure Drop Components

I.D.5 Shape Factors for Transverse Conduction in
Rod Bundles

(T. E. Greene)

Many of the computer codes designed to model rod bundle thermal-hydraulics are based on the subchannel analysis method. Subchannel analysis simplifies the modeling of rod bundles by defining discrete control volumes, or subchannels, each of which is characterized by a single temperature, velocity and density. This simplification is made at the expense of losing the detailed velocity and temperature distributions within the subchannel. As a result, empirical relations are required to describe the heat transfer and viscous shear at the subchannel boundaries. In particular, one of the heat transfer mechanisms requiring empirical information is conduction heat transfer through the coolant in a direction transverse to the primary coolant flow. This transverse heat conduction, which becomes important in the mixed convection regime, is the focus of the present work.

The most frequently used method for modeling transverse heat conduction is to assume that the temperature gradient across a subchannel boundary is simply the difference between the average temperatures of the two subchannels sharing the common boundary divided by the distance between the subchannel centroids. Of course, the actual temperature gradient at the boundary will rarely, if ever, be equal to this assumed temperature

gradient. In recognition of this fact, an empirical correction factor, or shape factor, is introduced to account for the actual shape of the temperature distribution. Figure 1 illustrates the subchannel concept and the relationship between the actual temperature gradient and the assumed temperature gradient. This illustration suggests a precise definition of the shape factor:

$$SF \equiv \frac{\eta \int_z^{z+\Delta Z} \int_0^{P-D} \frac{\partial T}{\partial n} dx dz}{(P-D) \Delta Z (T_i - T_j)}$$

where: SF = the shape factor

η = the distance between the subchannel centroids

$T_i - T_j$ = the difference between the subchannel average temperatures

$P - D$ = the length of the common boundary

$\frac{\partial T}{\partial n}$ = the actual temperature gradient across the boundary at some point (x, z) on the boundary

ΔZ = the height of the subchannel element

From the above equation we see that shape factors can vary axially and radially with each subchannel boundary.

Previous Work

A study of heat transfer literature was made to determine what results have been obtained in the past and what methods have been employed in finding shape factors for rod bundles. Several works were found which considered transverse conduction shape factors. Tables 1A and 1B summarize the characteristics of the studies made. All were similar in that they considered low Prandtl number coolants under slug flow conditions. With the exception of Fukuda's work, all shape factors were found by calculating the detailed temperature distributions within the subchannels and numerically integrating the results to find the subchannel average conditions.

Ramm et al. [5], analyzed the combined effects of heat transfer due to turbulent mixing and conduction heat transfer over a wide range of flow conditions, including laminar, transition and turbulent flow. In addition, the effect of the entrance region on the mixing parameters was considered. Although this work did not explicitly determine conduction shape factors, it is useful to the present work in that it identifies the following characteristics:

- 1) The mixing parameters are strongly dependent on the boundary conditions applied. This

indicates that the evaluation of the shape factor will have to be done in a region well-removed from any assumed boundary conditions.

2) At high Reynolds numbers the entrance lengths may be large. This will be counteracted, to some degree, by the low Prandtl numbers encountered with the liquid metals.

Parameters Influencing Shape Factors

The initial inquiry into the shape factor was concerned with identifying the properties and physical parameters which could have an effect on the shape factor. Intuitively, it is expected that any quantity effecting the temperature distribution should also effect the shape factor. Figure 2 is a summary of the quantities expected to be important in determining shape factors.

A dimensional analysis of the energy and momentum equations was done in order to find appropriate dimensionless groups for correlating the shape factor. The analysis indicated that the following dimensionless groups should be used to correlate the shape factor for a fixed geometry:

$$\text{Reynolds Number} \quad \frac{V \, De}{\nu}$$

$$\text{Prandtl Number} \quad \frac{\mu C_p}{k}$$

$$\text{Grashof Number} = \frac{g\beta(T-\bar{T})De^3}{\nu^2}$$

$$- \frac{q'''L}{k\nabla T}$$

This last group may be interpreted as a measure of the spatial power distribution within the bundle. The dimensionless analysis is included as Appendix A of this report.

Region of Shape Factor Importance

Transverse heat transfer between subchannels occurs by turbulent interchange and sweeping flow (wire wrap effect) as well as by conduction. It is expected that under some conditions, such as high Reynolds number flows and small power skews, the effect of transverse conduction will be negligible in comparison to the other processes. On the other hand, at low Reynolds numbers with large power skews, conduction may be the dominant mode of transverse heat transfer.

In order to focus on the cases of greatest significance, a preliminary analysis is being made to define the conditions under which the inclusion of the shape factor has a significant effect on the temperature distribution. As an importance criteria, the effect of shape factor will be assumed to be negligible if its inclusion results in a change of no more than 3°F in any subchannel temperature.

A problem arises when using this criteria. A priori knowledge of the shape factor is required in order to evaluate the importance. As a first approximation, it seems reasonable to assume that the shape factors will be somewhere within the range of $0.5 \leq SF \leq 2.0$. This range of shape factors can be used to estimate the operating conditions where shape factor is important. If the subsequent determination of the shape factors shows that the assumed range is seriously in error, then the region of shape factor importance can be redefined using the updated range of shape factors.

Assessment of the shape factor importance is being done with the ENERGY-III code for two geometries

LMFBR Fuel P/D = 1.24 H/D = 48 D = .23"

LMFBR Blanket P/D = 1.067 H/D = 8 D = .52"

In both cases a 37 pin bundle will be used.

Techniques for Finding Shape Factors

The most promising method for finding the conduction shape factors is the use of distributed parameter analysis. The detailed temperature and velocity profiles are calculated and the subchannel average temperatures and the heat flux across subchannel boundaries are then calculated by numerical integration. With this information shape factors can be calculated using the definition discussed earlier.

A list of features has been formulated as a criteria for evaluating the suitability of a code for calculating shape factors. An ideal code would have the following characteristics:

- 1) solve the energy and momentum equations in at least 2-D, preferably 3-D.
- 2) use either a truncated series solution or a finite difference solution on a detailed grid.
- 3) be able to model mixed convection
- 4) have at least a 2-D fuel pin model
- 5) allow bundle power skews
- 6) be able to model turbulent mixing
- 7) be able to analyze coupled cells

Table 2 summarizes the characteristics of a number of codes considered for use.

The two codes most nearly meeting the ideal code characteristics are the MIXCONV code and the BODYFIT code. While neither of these codes meets all requirements, modifications can be made. In the case of MIXCONV a 2-D fuel model and a cell coupling scheme would have to be added. BODYFIT will require the addition of a 2-D or 3-D fuel model.

BODYFIT has the advantage of giving a 3-D solution and its physical model is applicable over a wider range of conditions than is possible with MIXCONV. MIXCONV is limited to fully developed laminar flow with

buoyancy effects included. Whether or not the BODYFIT code will be used will depend on its availability and the feasibility of using the detailed model for the number of cases required.

Appendix A

Correlating Variables for Shape Factors

The steady-state energy equation (neglecting compressibility and viscous heating) may be written:

$$\rho C (\underline{V} \cdot \underline{\nabla}) T = q''' + K_{\text{eff}} \nabla^2 T$$

$$\text{where: } K_{\text{eff}} = SF \cdot K$$

Now consider two flows, 1 and 2, in similar bundles.

Define the F-factor:

$$F_{\gamma} \equiv \frac{\gamma_1}{\gamma_2}$$

where: γ is any physical quantity (T, g, k, q''' , dimensions, etc.)

The requirement that the bundles be similar requires that

$$F_x = F_y = F_z = \dots \text{ any bundle dimension F Factor } \dots = F_L$$

since both flows must satisfy the energy equation and noting that

$$\nabla_1 = F_{\nabla} \nabla_2 = \frac{\nabla_2}{F_L} \quad \nabla_1^2 = F_{\nabla}^2 \nabla_2^2 = \frac{\nabla_2^2}{F_L^2}$$

$$\begin{array}{ccccccc}
 \text{I} & & \text{II} & & \text{III} & & \text{IV} \\
 \frac{F_{\rho} F_V^2}{F_L} & = & \frac{F_P}{F_L} & = & F_{\rho} F_{\beta} F_G F_T & = & \frac{F_{\mu} F_V}{F_L^2}
 \end{array}$$

solving III and IV for F_V

$$F_V = \frac{F_{\rho} F_{\beta} F_G F_T F_L^2}{F_{\mu}}$$

combining this with the result from the energy equation
and noting $F_{pr} \equiv \frac{Pr_1}{Pr_2} = \frac{F_{\mu} F_C}{F_K}$

$$\frac{F_{\rho}^2 F_{\beta} F_G F_T F_L^3}{F_{\mu}^2} \cdot \frac{F_{\mu} F_C}{F_K} = 1$$

$$Gr \equiv \frac{\beta g (T - T_m) L^3}{\nu} \qquad Pr \equiv \frac{\mu C}{K}$$

$$\therefore (GrPr)_1 = (GrPr)_2 \qquad GrPr = \text{constant}$$

Combining I and IV

$$\frac{F_{\rho} F_V F_L}{F_{\mu}} = 1 \qquad \text{or} \qquad Re_{\nu} = \text{constant}$$

Writing the first relation found from the energy equation
as:

the following result is obtained:

$$\frac{F_{\rho} F_C F_V F_T}{F_L} (V_2 \cdot \nabla_2) T_2 = (F_q''') q_2''' + \frac{F_{SF} F_K F_T}{F_L^2} K_{eff2} \nabla_2^2 T_2$$

Multiplying this equation by $\frac{F_L^2}{F_K F_T}$

$$\frac{F_L^2 F_{\rho} F_C F_V}{F_K F_L} (V_2 \cdot \nabla_2) T_2 = \frac{F_q''' F_L^2}{F_K F_T} q_2''' + (F_{SF}) K_{eff2} \nabla_2^2 T_2$$

The required condition for the two flows to have the same shape factor is $F_{SF} = \frac{SF_1}{SF_2} = 1$. Using this condition along with the fact that flow 2 must satisfy the energy equation the following relation is obtained:

$$\frac{F_L^2 F_{\rho} F_C F_V}{F_K F_L} = \frac{F_q''' F_L^2}{F_K F_T} = F_{SF} = 1$$

The momentum equation (steady-state) is:

$$\rho (\underline{V} \cdot \nabla) \underline{V} = -\nabla p + \rho \beta (T - T_m) \underline{G} + \mu \nabla^2 \underline{V}$$

Applying the same type of analysis as was used with the energy equation above, the following results:

$$\frac{F_{\rho} F_C F_V K_L}{K_K} = \frac{F_{\rho} F_V F_L}{F_{\mu}} \cdot \frac{F_{\mu} F_C}{F_K} = 1$$

or $Rey \cdot Pr = \text{constant}$

The second relation from the energy equation analysis

$$\frac{F_{q'''}}{F_K} \frac{F_2^2}{F_T} = 1 = \frac{q''' L}{K \nabla T} = \text{constant}$$

In summary, the shape factor should be able to be correlated as

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$$\dot{q} = -(SF) K (P-D) \Delta z \left(\frac{T_i - T_j}{\eta} \right) = -K \int_z^{z+\Delta z} \int_0^{P-D} \frac{\partial T}{\partial n} dx dz$$

$$SF \equiv \frac{\eta \int_z^{z+\Delta z} \int_0^{P-D} \frac{\partial T}{\partial n} dx dz}{(P-D) \Delta z (T_i - T_j)}$$

FIGURE 1. SUBCHANNEL MODEL AND SHAPE FACTOR DEFINITION

FIGURE 2 - PARAMETERS AFFECTING SHAPE FACTOR.

RESEARCHER	COOLANT TYPE	ROD ARRAY TYPE	POWER DISTRIBUTION
FRANCE AND GINSBERG (1)	LIQUID METAL	FINITE SQUARE ARRAY $1.0 \leq P/D \leq 1.5$	UNIFORM AXIALLY AND AZIMUTHALLY WITHIN RODS. No BUNDLE SKEW
RAMM AND JOHANSSON (2)	Low PR FLUIDS $P_R: .03, .003$	INFINITE TRI- ANGULAR ARRAY $P/D: 1.10, 1.30$	UNHEATED RODS NON- UNIFORM TEMPERATURE DISTRIBUTION AT INLET
YEUNG AND WOLF (3)	SODIUM	FINITE TRIANGULAR ARRAY	UNIFORM AXIALLY AND AZIMUTHALLY WITHIN RODS. BUNDLE POWER SKEW.
FUKUDA (4)	SODIUM	INFINITE TRIANGULAR ARRAY	NONUNIFORM TEMPERATURE DISTRIBUTION

TABLE 1A. SUMMARY OF PREVIOUS WORK ON SHAPE FACTORS.
CONTINUED IN TABLE 1B.

RESEARCHER	FLOW REGIME	FLOW DISTRIBUTION	CALCULATIONAL METHOD USED
FRANCE AND GINSBERG (1)	SLUG FLOW	$\frac{U_I}{U_J} = \frac{D_{EI}}{D_{EJ}}^{5/7}$	DISTRIBUTED PARAMETER ANALYSIS
RAMM AND JOHANSSON (2)	SLUG FLOW $10^3 \leq Re \leq 2 \times 10^5$	UNIFORM	DISTRIBUTED PARAMETER ANALYSIS
YEUNG AND WOLF (3)	SLUG FLOW	$\frac{U_I}{U_J} = \frac{D_{EI}}{D_{EJ}}^{5/7}$	DISTRIBUTED PARAMETER ANALYSIS
FUKUDA (4)	SLUG FLOW	UNIFORM	ANALOG SIMULATION USING ELECTRICAL RESISTANCE PAPER

TABLE 1B. SUMMARY OF PREVIOUS WORK ON SHAPE FACTORS,
CONTINUED FROM TABLE 1A.

CODE	EQUATIONS SOLVED	MIXED CONVECTION CAPABILITY	TURBULENCE MODEL	MULTI-CELL ANALYSIS	SOLUTION TECHNIQUE	BUNDLE POWER SKEW	FUEL MODEL
MIXCONV (KIM, WANG)	ENERGY MOMENTUM (2-D)	YES	N/A	No	SERIES	N/A	CLAD ONLY 1-D
LOCAL (YEUNG)	ENERGY SLUG FLOW (2-D)	No	No	YES	SERIES	YES	1-D
RODBUN (RAMM)	ENERGY SLUG FLOW	No	YES	YES*	SERIES	YES	YES
3-D FLOW (WONG)	ENERGY SLUG FLOW	No	No	No	SERIES	N/A	No
BODYFIT (SHA)	ENERGY MOMENTUM (3-D)	YES	No	YES**	FINITE DIFFERENCE IN TRANS- FORMED CO- ORDINATES	YES	1-D

* - 2 TRIANGULAR INTERIOR CELLS

** - MODELS ENTIRE BUNDLE

TABLE 2. DISTRIBUTED PARAMETER CODE CHARACTERISTICS

II. SUBCHANNEL GEOMETRY (BARE RODS)

Work in this area was completed with the reports of:

J. Kelly "Turbulent Interchange in Triangular Array
Bare Rod Bundles" (COO-2245-45TR) July 1977, and

J. Bartzis "Hydrodynamic Behavior of a Bare Rod Bundle"
(COO-2245-48TR) June 1977.

III.1 Three-Dimensional Steady Isothermal Flows (Robert W. Sawdye)

Due to limited funds, the experimental portion of this work was given greater priority and the computational efforts have been delayed until after significant measurements have been made.

Experimental Design

The necessary optical and electrical components are being acquired to complete the three-dimensional laser doppler anemometer (LDA). Two velocity components will be measured with the operational system utilized by D. Boyle for the transient measurements described in his thesis (soon to be published as a report under this project). The third velocity component will be measured with equipment being purchased together with other currently available apparatus which was acquired for previous work on this project.

As was discussed in the immediately previous progress report, the test cell containing the turbulent flow field of interest will translate with respect to the stationary LDA equipment in moving from one measurement station to another. The velocities are measured at the point of intersection for all of the laser beams. To maintain the position of this measuring point fixed in space, these light beams must each travel through the same amounts of air, water and plexiglas for all movements of the test cell. To achieve this, the test section will be immersed in a bath which will surround the flow cell with water and will be large enough to allow for the necessary translations to measure throughout the cell interior. The experimental apparatus is shown in schematic form in Figure 1, which contains the two lasers and their associated optics, the three photodetectors, the test cell and the water bath. These are all shown in a horizontal cross-section view of the plane of measurement. The flow cell will be attached by an overhanging beam to a translating table (along with the tubing that provides the flow in the cell), and the movement of the table can be precisely controlled by screw adjustments.

The three photodetectors will be connected, two at any one time, to the two frequency trackers which follow the velocities (u, v, w) and their fluctuations (u', v', w'). Each tracker is connected to an RMS unit which has outputs to determine the time averaged or mean velocity ($\bar{u}, \bar{v}, \bar{w}$) and level of turbulence kinetic energy (TKE) for that velocity component (u'^2, v'^2, w'^2). Both trackers together are connected to a turbulence processor unit which can provide the correlation of the two velocity fluctuations, or in effect, one of the Reynolds stress components ($\overline{u'v'}, \overline{u'w'}, \overline{v'w'}$). Since there are three velocity components being measured at each of the points of interest within the steady flow field, the measuring procedure will involve switching between the three combinations of two components to obtain each component of the Reynolds stress.

The cubic test cell interior will measure 20 cm on a side, due mainly to translation limitations imposed by the table on which it will be mounted (a small milling table). This size constraint implies the possibility of providing more than 1000 measuring points if a 2 cm space interval is used for the three dimensions. It may be possible to assume the existence of some symmetry in the various flow fields to be investigated, and thereby to considerably reduce the amount of time devoted to their measurement. It is intended that an integral number of space intervals will be used to accommodate the inlets and outlets, and the inlets will be designed to provide a uniform velocity profile. The maximum inlet Reynolds number that can be achieved with this apparatus is about 10^5 ; therefore, fully turbulent flow fields will be developed.

Figure. 1 Plan View of Laser Doppler Apparatus for Measuring Three-Dimensional Flow Fields

III.2 Two-Dimensional Nonisothermal Turbulent Flow

(Soon H. Chang)

The main effort during this period has concerned new research plans for advanced physical and numerical modeling of nonisothermal turbulent flow.

The Research Areas

There are three major areas of this work:

- 1) advancement of the physical models
- 2) advancement of the current numerical method
- 3) verification of the resulting code with experimental results: (ANL scale model results are available for experimental validation).

The Proposed Research

Advancement of the physical models

- 1) Buoyancy terms will be incorporated in the two turbulence transport equations
- 2) Multi-equation turbulence modeling will be tested with both three-equation modeling and four-equation modeling^{2,3}
- 3) The relation between the eddy diffusivity of heat and the eddy diffusivity of momentum will be more accurately modeled.

Advancement of the current numerical method.

- 1) The pressure iteration portion of the solution scheme, which can be the most time-consuming part of the calculation will be investigated for reducing time and increasing accuracy.
- 2) Appropriate boundary conditions will be investigated for more accurate calculations.
- 3) Tensor viscosity method⁴ will be investigated for more accurate treatment of numerical diffusion

Tenative Time Schedule

Dec. 1 1979 - Jan. 15th, 1980	Review of Problem
Jan. 16th, 1980 - March 31, 1980	Testing the present code with experimental results
April 1, 1980 - July 31, 1980	Review of Turbulence Modeling - Propose and Implement Improvements
August 1, 1980 - Oct. 31, 1980	Review Numerical Methods - Propose and Implement Improvements
Nov. 1, 1980 - Dec. 31, 1980	Further Modification of Computer Code
Jan 1, 1981 - March 31, 1981	Verification of Advanced Code with Experimental Results
April 1, 1981 - May 30, 1981	Analyze Results

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IV. Theoretical Determination of Local Temperature Fields
in LMFBR Fuel Rod Bundles (Chung-Nin Channy Wong)
Wire-Wrapped Rod Bundle Heat Transfer Analysis

I. Introduction

During the month of September, a review of the method which was used to analyze the steady-state heat transfer problem of the wire-wrapped rod bundle was completed. It was shown earlier in refs. 1 and 2 that the original intention was to solve the steady-state laminar and turbulent flow problems by employing the steady-state approach. This approach does not include any temporal term in the governing equations. Hence, any of the unknown variables will not be a function of time. In addition, the authors assume that the working fluid is incompressible, single phase and has constant properties. These assumptions permit the authors to uncouple the momentum equations and the energy equation. As a result, different sets of equations are to be solved. One set consists of three momentum equations plus the continuity equation with three velocities (i.e., radial, traverse, and axial velocities) and pressure as the unknown variables. The second set has the energy equation only and the unknown parameter is either the temperature or the internal energy.

II. Steady-State Approach vs. Transient Approach

The equations of motion for a three dimensional steady-state fluid flow problem are nonlinear elliptical equations. In the Cartesian Coordinate System, they can be expressed as the following:

$$U_j \frac{\partial U_i}{\partial X_j} = -\frac{1}{\rho} \frac{\partial P}{\partial X_i} + \nu \frac{\partial^2 U_i}{\partial X_j^2} \quad \text{for } i = 1, 2, 3$$

The Newton-Rafason method was proposed to solve these nonlinear equations [Ref. 1, 2]. However, it is well-known that if the initial guess does not lie within the radius of convergence, Newton's method does not guarantee that it converges to the right answer. In the previous reports [Ref. 1, 2], the authors used the previously calculated value of the variables in the adjacent upstream mesh cell as the initial guesses. This estimation works fine provided that the size of the mesh cell is small enough such that the initial guess is within the limit of convergence. Hence, an enormous number of mesh cells are expected to be necessary to solve a simple fluid flow problem with a moderate Reynolds number. Even though the rate of convergence for Newton's method is second degree convergence, a substantial amount of time is needed to evaluate the Jacobian matrix before each iteration. This requires tremendous storage space and

large computational time. This approach is not practical, especially for the turbulent flow problem.

The other way to obtain the steady-state solution is to work on the problem as a transient problem. Let the transient solution proceed with time until it reaches the equilibrium state. Even though the present major goal is to solve the steady-state problem, the transient problem is also an interesting problem and will properly be the next major topic in our research. Therefore, some effort has been spent to search for the best solution technique for the transient problem.

The transient equations of motion have a parabolic form. They can be written as following:

$$\frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial X_j} = - \frac{1}{\rho} \frac{P}{X_i} + \gamma \frac{\partial^2 U_i}{\partial X_j \partial X_j} \quad \text{for } i = 1, 2, 3$$

If the transient approach is used, one can formulate the finite difference equations of motion by expressing the nonlinear convection term fully explicitly. In this manner the nonlinearity problem can be easily suppressed. The only restriction is that the size of the time step cannot be greater than the Courant number (i.e., $\Delta t \leq \Delta X_i / |V_i|$). This implies that the fluid cannot convect larger than one mesh cell within one time-step. Then, if this criterion is satisfied, the

the solution will converge. For the steady-state solution, let the transient solution proceed with time until it reaches the final equilibrium state. Unfortunately, there isn't any good way to predict the time required to achieve the steady-state solution. It depends highly upon the problem itself. This uncertainty makes it hard to choose between the transient approach and the steady-state approach. Finally, the authors decided to go ahead with the transient approach because it offers an easy way to linearize the problem and an approach which can solve both the steady-state and the transient solutions.

III. Method of Solution

The solution technique adopted in our analysis is quite similar to the Generalized Implicit Continuous Eulerian (ICE) technique. In the equations of motion, the convection term and the diffusion term are treated explicitly while the pressure gradient term is treated implicitly. For any time-step, the momentum equations can be written as the following forms:

$$\rho \frac{U_i^{n+1} - U_i^n}{\Delta t} = - \text{convection}^n + \text{diffusion}^n - \text{pressure gradient}^{n+1}$$

where n , the time step = $1, 2, \dots, N$

Expressing the pressure gradient implicitly implies that the pressure wave will propagate with an infinite speed. This approximation appears to be reasonably good provided that the Mach number of the flow problem is very small. In all of our cases, the fluid is incompressible; thus, the sonic speed is tremendously high. Even though our problem is in the turbulent flow regime, the Mach number is still comparatively small. So treating the pressure gradient implicitly is a reasonably good approach.

In the momentum equations, the velocities (i.e., radial, traverse and axial velocities) at the new time level are expressed in terms of the new pressure and the old velocities. That is, for any velocity,

$$U_i^{n+1} = U_i^n + \Delta t \cdot \{-[\text{convection}]^n + [\text{diffusion}]^n - [\text{pressure gradient}]^{n+1}\}$$

where $i = 1, 2, 3$

Substituting these velocities into the continuity equation, an equation for the pressure at the advanced time will be obtained. This equation has an elliptical form, quite similar to the Poisson equation. It can be written as following:

$$\begin{aligned}
& \frac{2}{\Delta r^2} + \frac{2}{r_i^2 \Delta \theta^2} + \frac{2}{\Delta z^2} P_{ijk}^{n+1} - \frac{1}{\Delta r^2} P_{i+1,j,k}^{n+1} \\
& - \frac{1}{\Delta r^2} P_{i-1,j,k}^{n+1} - \frac{1}{r_i^2 \Delta \theta^2} P_{i,j+1,h}^{n+1} - \frac{1}{r_i^2 \Delta \theta^2} P_{i,j-1,h}^{n+1} \\
& - \frac{1}{\Delta z^2} P_{i,j,k+1}^{n+1} - \frac{1}{\Delta z^2} P_{i,j,k-1}^{n+1} \\
& = B_{ijk}
\end{aligned}$$

$$\text{where } B_{ijk} = B(v_r^n, v_\theta^n, v_z^n, r, \theta, z, t)$$

$$\text{for } i = 1, I$$

$$j = 1, J$$

$$k = 1, K$$

The discrete pressure equations can be considered as a set of linear equations with the discrete pressures, P_{ijk} as the unknowns. If expressing these equations in the matrix form, it can be written as:

$$[A]\{x\} = \{B\}$$

Notice the elements in the coefficient matrix $[A]$. The matrix, A , is only spatial dependent or geometrical dependent. All the velocities at the old time and other temporally

dependent variables appear in the vector $\{B\}$. If the direct method is applied to solve this set of the linear equations, one only has to invert the matrix once. For any other time-step except the first one, the coefficient matrix $[A]$ won't change, so neither does the inverse of the matrix, i.e., $[A]^{-1}$. Therefore, only backward substitution is needed to calculate the new pressure at any time. This saves a lot of computational time and cost.

The direct method instead of any iterative method will be used here to solve the linear system. Our solution technique is basically designed to solve any problem which is oriented in the polar coordinate system. In this system, the coefficient matrix $[A]$ has the following properties: it is real, weakly diagonally dominant and has complex eigenvalues [Ref. 3]. Most of the iterative techniques used to solve a system of linear equations won't be efficient here [Ref. 3]. In our calculation, a direct method known as LU-decomposition is used. For any time-step, the pressure at the advanced time level can be obtained by backward substitution. Once the pressure is known, the velocities at the advanced time level can be calculated easily.

IV. Sample Problem

A computer code called 3DFlow has been written. The purpose of this code is to analyze the three

dimensional fluid flow problem using the method as stated in section III. To verify this code, a simple circular pipe flow problem has been set up and is being solved.

(a) The geometry of the pipe

Shape = Circular

Diameter of the pipe = 1 inch

Length of the pipe = 10 inches

(b) The properties of the fluid

Fluid = Water

Density, $\rho = 62.3 \text{ lbm/ft.}^3$

Viscosity, $\mu = 2.3194 \times 10^{-4} \text{ lbm/ft. sec.}$

Kinematic Viscosity, $\nu = 4.2.26 \times 10^{-6} \text{ ft.}^2/\text{sec.}$

(c) Flow Condition

Initial condition: no flow at time < 0

Boundary condition: for any time ≥ 0

1. At the center, $r = 0$

$$U_r = 0$$

$$\frac{\partial U_r}{\partial r} = 0$$

$$\frac{\partial U_z}{\partial r} = 0$$

2. At the wall, $r = 0.5$ in.

No slip

$$U_r = 0$$

$$V_\theta = 0$$

$$U_z = 0$$

3. At the inlet, $z = 0$ in.

$$U_r = 0$$

$$V_\theta = 0$$

$$U_z = 1.011 \times 10^{-2} \text{ ft./sec.}$$

(d) The Result

After the first time-step (approximately equal to 0.5 sec.), a linear pressure distribution along the flow is observed. This big pressure gradient with a constant value is needed to initialize the flow. The velocities along the pipe at this time step are uniformly distributed as in the inlet condition. Once the fluid starts moving, the value of the pressure gradient drops off sharply at the second time-step. This phenomenon happens because when the fluid starts moving with a constant speed, there is no need of additional pressure to accelerate the flow. The only pressure drop is due to the wall friction which is comparatively small for the low Reynolds number flow.

Within the first seven time-steps, the solution behaves as expected. One can easily observe that

the viscous effect starts coming in from the wall and diffuses slowly towards the center of the duct. This phenomenon, obviously, changes the axial velocity profile. But at the eighth time-step, a small perturbation of the axial velocity occurs. It propagates and amplifies with time. As a result, the system becomes very unstable, and finally the solution diverges. One way to resolve this problem is to impose some damper to diminish this perturbation. The outcome will be discussed in the next report.

V. Discussion

When the problem of the amplification of the small perturbation has been resolved, our next effort will be concentrated on solving the fluid flow problem of the internal cell in the wire-wrapped rod bundle.

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