A Porous Body Model For Predicting Temperature Distributions
In Wire Wrapped Fuel and Blanket Assemblies of a LMFBR

by

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Abstract

Existing methods of thermal analysis of a wire wrapped rod bundle of a Liquid Metal Fast Breeder Reactor are based on the principle of subchannel analysis. A model for thermal transport in wire wrapped rod bundles is developed here. The model is similar in principle to the one which has long been successfully used in chemical engineering for heat and mass transfer in fixed beds of packed solids. By dividing the bundle into two predominant regions and applying the model of a porous body to a LMFBR assembly a simple procedure for calculating temperature distributions in LMFBR fuel and blanket assemblies has evolved. The results obtained from this analysis were found to predict available data with as good a precision as do the more complex analyses.

Correlations for the two empirical constants (of this study) were obtained as functions of geometric parameters based on an extensive analysis of existing data.

The LMFBR fuel assemblies operate in forced convection (negligible natural convection) under steady state conditions whereas the blanket assemblies may operate in forced or mixed convection (combined forced and free convection). Two different formulations of equations, corresponding to these two convection regimes, were developed using the same basic model. The calculation procedure for assemblies in forced convection (called ENERGY I) is considerably simpler than that (ENERGY II, ENERGY III) in mixed convection, where buoyancy effects become important. Therefore it is desirable to use ENERGY I for forced convection (although ENERGY II, III can also be used in forced convection, the computational times are fifteen fold greater). In order to determine when buoyancy effects become important a new criterion is developed. Given the power, the power skew, the operating and geometric characteristics of the bundle, the critical modified Grashof Number Gr_c^* predicts when buoyancy effects become important.

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NOMENCLATURE

- A = Area
 A_T = Total Area of Hexagonal Duct
 A_C = Cross-flow Area per Unit Length
 C = Ratio of Superficial Swirl Velocity to
 Bundle Average Velocity $\left(= \frac{U_s}{\bar{U}} \right)$
 C_p = specific heat
 C_1 = ratio of swirl velocity in gap between rod
 and wall and bundle average actual axial
 velocity ($C_1 = v_\theta / \bar{V}$)
 d_e = hydraulic diameter of an actual interior
 channel
 d_w = wire wrap diameter
 d = rod diameter
 f = friction factor
 F_D = effective axial frictional drag ($= \frac{Mf}{d_e}$)
 $\bar{G}$ = average superficial mass velocity in bundle
 Gr = Grashof number
 Gr^N = (no mixing) modified Grashof number
 Gr_C^N = critical (no mixing) modified Grashof number
 Gr^* = (mixing) Grashof Number
 Gr_C^* = (mixing) critical Grashof number
 h = wire wrap lead
 h^* = dimensionless enthalpy used only in
 Equation (4) for ENERGY-I formulation

H = enthalpy in ENERGY III formulation

H^* = cross-flow effective enthalpy

K = thermal-conductivity

L = length of bundle

M = Novendsterns¹⁶ friction multiplier

N = No. of nodes in bundle at one axial level

n = coordinate in Region II perpendicular
to wall

n^* = dimensionless coordinate

Pr = Prandtl number

P = pressure

Q = heat generation per unit (total) volume
at the point x, y, z

Q^* = dimensionless heat generation $\left(= \frac{Q}{\bar{Q}} \right)$

$\bar{Q}$ = average heat generation per unit (total)
volume

$\hat{Q}_p$ = Bundle peak to average power ratio

Re = Reynolds number $\left(= \frac{Gde}{\mu} \right)$

s = coordinate in Region II parallel to duct
wall

s^* = dimensionless coordinate

S_c = Schmidt Number

T = coolant temperature at the point x, y, z
in the bundle

$\bar{T}$ = average coolant temperature in the cross-section of the bundle

T_M = average exit coolant temperature

T_{IN} = inlet coolant temperature

$(U_I)_z$ = superficial axial velocity in Region I

$(U_{II})_z$ = superficial axial velocity in Region II

$\bar{U}$ = bundle average superficial velocity

U^* = effective axial velocity associated with cross-flow

U_s = superficial swirl velocity in Region II

V_θ = actual swirl velocity between rod and duct wall

V = actual axial velocity in the bundle

$\bar{V}$ = bundle average actual axial velocity

W_{Tij} = Diversion cross-flow

W_{Sij} = Sweep cross-flow

$\bar{W}$ = bundle average axial flow

GREEK SYMBOLS

ϵ = effective enhanced eddy diffusivity

ϵ^* = dimensionless eddy diffusivity $\left(= \frac{\epsilon}{\bar{U}de} \right)$

α = thermal diffusivity

α^* = dimensionless thermal diffusivity $\left(= \frac{\alpha}{\bar{U}de} \right)$

κ = constant for 2-D transverse

heat flow in porous media

ρ = density

μ = viscosity

γ = kinematic viscosity

χ = defined in Equation 37

λ_{a1} = axial porosity in Region I

λ_{a2} = axial porosity in Region II

$\overline{\lambda}_a$ = bundle average axial porosity

λ_{L1} = lateral porosity in Region I

λ_{L2} = lateral porosity in Region II

η_T = node spacing in Region I

η_T^* = dimensionless node spacing in Region I
 $\left(= \frac{\eta_T}{de} \right)$

η_s = node spacing in 's' direction in Region II

η_s^* = dimensionless η_s

η_n = node spacing in 'n' direction in Region II

η_n^* = dimensionless η_n

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1.1 Introduction

In Liquid Metal Fast Breeder Reactors (LMFBR) the fuel and blanket assemblies are subjected to severe radial heat flux gradients and a very wide range of flow conditions. Core designers have evolved^{(1-9)*} a number of calculation methods for predicting temperature distribution in these assemblies in order to determine cladding hot spots, assembly housing bowing and deformation, and temperature and pressure fields as input to prediction of the location of incipient boiling and calculation of subsequent voiding.

The maximum burnup (Mwd/T) for current LMFBR designs is very sensitive to the peak coolant temperature in the hot channel, which in turn could depend, to a large extent, on interchannel mixing rates. In addition, the fuel assembly housing bowing is also of special concern. Bowing of the hexagonal housing is caused by differential thermal expansion. It is greatly enhanced by radiation induced stainless steel swelling in a fast spectrum and the temperature dependence of this induced swelling. Thus realistic predictions of assembly temperature distribution could

have important effects on the flow housing design considerations.

A typical LMFBR fuel assembly consists of 217 fuel rods arranged in an hexagonal array with uniform spacing maintained between the rods by wire wraps. A typical blanket assembly consists of 61 fuel rods of considerably different geometrical dimensions but also spaced by one wire wrap around each rod. There are two major hydrodynamic effects of the presence of wire wraps on the fuel assembly design. Its presence increases the pressure drop and thus the pumping power required, but has the virtue of enhancing flow mixing in the bundle and thereby further reducing temperature gradients if they exist. The fuel assemblies always operate at a high Reynolds number ($50-70 \times 10^3$) during the steady state operation of a LMFBR while the Reynolds number in the blanket assemblies may be in the laminar, transition or turbulent flow regime. Thus fuel assemblies usually operate in the region where free convection is negligible as compared to forced convection, whereas the blanket assemblies may operate both in the region where free convection is of the same magnitude as forced convection and also in the region where free convection may be much smaller than forced convection. In the present report the term mixed convection is used to specify the

regime where natural and forced convection effects are both important and the term forced convection implies that natural convection effects are negligibly small.

The various modes of energy redistribution within fuel and blanket assemblies are discussed in Appendix I. Flow sweeping is the predominant mode of energy exchange in fuel assemblies. However for blanket assemblies operating at low Reynolds number, energy transport by thermal conduction can become important. Figure 1 shows that as the coolant flows vertically upwards it is swept by the wire wraps between any two rods in a manner such that it alternates in direction. The cross flow rate is presumed maximum where the wires cross the gap between the rods and it reverses direction between adjacent spacer locations. At the bundle walls the flow is always directed in one direction parallel to the housing wall; this is called swirl flow.

Flow sweeping and swirl flow can affect the peak coolant temperature and the housing wall temperature distribution significantly. Most experimental effort has been to quantitatively determine these quantities and their dependence upon geometrical and flow parameter. The major emphasis has been in determining flow sweeping and swirl flow (or an equivalent mixing coefficient) perhaps because it is the least understood and the most

important means of momentum and energy transport. Once an estimate of flow sweeping or an equivalent mixing coefficient is made from experiments, it can be directly put into the computer programs (Table I) which have been developed based on various methods of rod bundle thermal analysis.

1.2 Existing Analytical Methods

The existing methods of wire wrapped rod bundle thermal analysis (Ref. 1-9) have been classified in Table I on the basis of how the rod assembly is subdivided in order to describe the flow and temperature fields. Most existing calculation procedures (Ref. 2-9) are based on the method where the reactor assembly is divided into subchannels (the space between 3 adjacent rods in a LMFBR is generally used to define a subchannel). A different method is introduced in Ref. 1 where the rod array in a LMFBR fuel assembly is divided into two predominant regions. Each of these regions is treated as a continuum by viewing the wire wrapped bundle as a porous body. The two region model approach was independently developed and used in Ref. 2 but a subchannel analysis calculation procedure was maintained. The present report clarifies and extends the method developed in Ref. 1 (Ref. 1 is a topical report and describes the initial phase of the work done as a part of this thesis).

Refs. 3-7 are subchannel analysis methods where the energy and continuity equations are solved level by level axially to determine the temperature field. The marching procedure followed by these computer programs reduces computational times significantly. However considerable amount of bookkeeping is required since the flow swept into and from each subchannel, by the wire wraps, is recorded and used in the equations at each level. The flow distribution into various subchannels is based on hydraulic diameter flow split which must be experimentally verified. Refs. 8 and 9 are iterative methods where the momentum energy and continuity equations are written for each subchannel along with cross-flow coupling terms and these are solved iteratively. The computational times for these iterative methods of solution is considerably greater than those required by the marching procedures. The methods of references 3-7 require a knowledge of four empirical constants (one constant each for turbulent exchange, diversion cross-flow and for flow sweeping in central and wall channels). In addition the initial and boundary conditions must be prescribed.

Both the iterative and marching methods of sub-channel analysis computer programs suffer from two major shortcomings. First each of the four correlation constants

cannot be separately and accurately determined. Second, the various modes of mixing are considered separately in these codes and linearly added together resulting in predictions of energy exchange which may be in error. Thus the usefulness of the sophisticated numerical and computational techniques may be offset by the errors and approximations they utilize. At present there appears to be the need of a relatively simplified method for predicting, within engineering limits of accuracy, the coolant temperature distribution within LMFBR rod bundles. The present thesis describes the development of a relatively simple model which is similar in principle to the one which has long been successfully used for heat and mass transfer in fixed beds of packed solids (Refs. 10-13). By dividing the LMFBR bundle into two predominant regions and applying the model of a porous body a simple procedure for calculating temperature distributions in LMFBR fuel and blanket assemblies has evolved. The results obtained from the present method were found to predict available data with as good a precision as do the more complex analysis.

1.3 Existing—Experiments

The existing experiments have been conducted with the following goals:

(a) to determine the bundle flow split i.e. the average axial flow in the three types of subchannels, namely central, wall and corner subchannels. These are input into the marching codes.

(b) to determine a sweep-flow coefficient or an equivalent mixing coefficient for direct input into computer programs.

(c) to measure swirl-flow parallel to the duct wall.

Tables 2 and 3 summarize the available world data, classified in Table 2 according to the type of experiment (see below) and in Table 3 according to geometry.

(a) The bundle flow split has been measured by isokinetic sampling techniques in Ref. 12 and by time-of-flight measurements using a thermal pulse in Ref. 15.

Both set of data show that the velocity is almost uniform across the bundle. The measurements do not agree with the theoretical method developed by Novendstern.¹⁶

Fig. 2 (taken from Ref. 17) shows this difference between theory based on hydraulic diameter flow split and various experimental data. Except for the 7-pin data (Ref. 18) the theory predicts a larger velocity in the wall channels than experimentally measured. Also shown in Fig. 2 are data obtained from Ref. 19 on a 217-pin bundle and from Ref. 20 on a 61 pin bundle.

The A.I. data¹⁹ was a result of time-of-flight measurements using a salt tracer pulse whereas the M.I.T. experiments²⁰ used a Laser Doppler Velocimeter. It is obvious that as the bundle size increases, the discrepancy between predicted flow split and the measured data also increases.

(b) The three types of experiments that are most frequently used for determining wire-wrap mixing in a rod bundle can be classified according to their boundary conditions.

1) Continuous injection of an electrolyte salt solution at a point in the bundle. Electrical conductivity measurements at various points in the rod bundle can give an estimate of the amount of salt transported by mixing. From this a mixing coefficient can be derived.

2) Continuous local injection of heated coolant. Temperature measurement in the three dimensional bundle matrix will yield a mixing coefficient with appropriate data analysis.

3) Employment of one, several, or a complete array of heated pins. By measuring temperature distributions in the bundle, one can derive a mixing coefficient.

Heated pin experiments have been performed by

Hoffman and Bumgartner², Fontana et al. Baumann and Hoffman²⁶ and Ishibashi²⁷. Refs. 2 and 26 report data on single and two heated pins in a 61 rod bundle cooled by liquid sodium and operating at high Reynolds numbers typical of fuel assemblies. The data of Fontana et al.²¹ are taken in a 19-pin bundle with liquid sodium as coolant. The bundle is subjected to power skews varying from flat (uniform) power profile to as much as 300 % across flats. A very wide range of Reynolds numbers are covered varying from 1100 to about 70000. The low Reynolds number data is very valuable since buoyancy effects may begin to play an important role. A complete description of the range of data is given in Chapter 3. Ishibashi²⁷ has obtained several sets of data on a 91 pin bundle with a geometry typical of fuel assemblies and also on a 19 pin blanket assembly.

Skok²² and Hines et al.²⁴ report hot water injection data in a 7 pin and a 127 pin bundle respectively. Skok²² covers a wide range of bundle geometry (Tables 2, 3) typical of fuel and blanket assemblies. Hines et al. only report one complete set of data on a geometry which closely represents the fuel assembly.

Salt injection experiments have been conducted by Okamoto et al.²³, Collingham et al.²⁵ and Lorenz et al.²⁸.

In addition M.I.T.²⁹ experiments should shortly yield salt injection - conductivity probe measurement data both on geometries typical of fuel and blanket assemblies of a LMFBR.

(c) Swirl flow parallel to the duct wall is a consequence of the wire-wrap flow sweeping and can be theoretically calculated by a flow continuity balance at any axial level. However a considerable amount of effort (14, 15, 20, 28) has been devoted to measuring it independently by salt¹⁴ and hot water¹⁵ injection techniques and by use of the Laser Doppler Velocimeter^(20,28). Thus code predictions for swirl flow can be confirmed by these independent measurements. Appendix 1 defines C_1 , the measure of swirl velocity used in subchannel analysis computer programs. Analysis of various sets of data for determining C_1 have been made. Fig. 3 shows that the swirl velocity varies with bundle size for bundles of similar geometry. It is yet to be resolved whether this inconsistency in C_1 is due to the relative inaccuracy in data or is in fact due to a bundle size effect. Until this is resolved care must be exercised in extrapolating data taken on smaller bundles to the larger prototype bundle.

Chapter 2

2.1 PROPOSED MODEL

Most existing calculation procedures²⁻⁹ are based on subchannel analysis.

The present model differs from subchannel analysis methods in two distinct ways. First, the rod array in a LMFBR fuel assembly is divided into two predominant regions; the central and wall regions. Second, both these regions are treated as a continuum by viewing the wire wrapped bundle as a porous body.

The various modes of energy transport that exist in the steady state operation of an LMFBR rod assembly have been discussed in Appendix 1. It should be noted that conduction through the individual fuel rods and through contacts between individual rods (as the wire wrap passes through the gap between rods) has been neglected compared to the other modes of energy transport since they are expected to be comparatively very small.

The wire-wrapped rods are packed in an array which is enclosed by a container and as a result there is a region of flow next to the container wall which is quite different in character from the flow in the central region. In the central core the mean flow oscillates around each rod progressing in an axial direction.

This oscillation of flow, shown in Fig. 1, may be imagined to contribute an effective eddy diffusivity superimposed on the normal eddy diffusivity associated with turbulence. Thus a basis of our model is the existence of an uniform lateral effective enhanced eddy diffusivity, ϵ , for heat transfer, in the inner region of the bundle. The eddy diffusivity in the axial direction is not significantly enhanced by the presence of wire wraps. In the outer region near the wall the flow is quite different. Since the wire wrap is spiralled around each rod in the same direction, the flow progresses with both an axial component and a tangential component parallel to the wall (Fig. 4).

The proposed model treats the rod assembly as a continuous porous media and energy generation by the rods is modelled by a continuous volumetric heat source distribution. The energy transfer in the transverse direction is modelled by the effective eddy diffusivity which includes the fluid oscillations due to the presence of wire wraps and also any other means of energy transport, like natural turbulence. This present model is similar in principle to the one which has been used to study heat and mass transfer in porous media. (10-13) The packed bed is analogous to the bundle of rods in our present case. The main difference

between the two cases is that the Reynolds number at which the packed beds operate are much lower than that in the nuclear reactor fuel assembly. This difference is not fundamental, however, and is reflected merely in the magnitude and form of the effective eddy diffusivity.

The model described above is applicable to any wire-wrapped assembly irrespective of whether it is operating under conditions of forced convection or in a mixed (forced/free) convection regime. However the formulation for assemblies operating in forced convection is considerably simpler than those operating in mixed convection because in the latter case the axial momentum and energy equations are coupled and a simultaneous solution of continuity, momentum and energy equations is required. Due to the considerable difference in the complexities of these two calculational procedures, they are described separately. In Section 2.2 the formulation for assemblies operating in forced convection is presented and in Section 2.3 their method of solution is discussed. Formulation for assemblies operating in mixed convection is given in Section 2.4.

Given an LMFBR bundle with known geometry, total power generation rate, power skew and total flow rate it is desirable to know which of the two calculating

procedures described above should be used. In Section 4 a criterion is developed to predict when buoyancy effects become important. If buoyant effects are important a designer should use the analysis developed in Section 2.4 for an assembly operating in mixed convection.

2.2 Formulation for Bundles in Forced Convection

In applying the model to bundles in forced convection the assembly is divided into two different regions as discussed earlier. Region I is the inner core, which has a uniform superficial axial velocity $(U_1)_z$ (i.e. uniform volumetric flow rate per unit total area). Region II of thickness t , has a uniform superficial velocity $(U_{II})_z$ in the axial direction and an uniform circumferential velocity $(U)_s$ in the circumferential direction. The heat generated in the rods is represented by a continuous volumetric heat generation, $Q(x,y,z)$, which simulates the heat generation in the real rods.

The energy transport equation for Region I (inner region) may be written as follows (Fig. 5).

$$\rho C_p (U_I)_z \frac{\partial T}{\partial z} = (\rho C_p \epsilon_I + \kappa K) \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) + Q(x,y,z) \quad (1)$$

where

$(U_I)_z$ = superficial axial velocity in Region I

$T = T(x,y,z)$, the coolant temperature

ϵ_I = effective eddy diffusivity for heat transfer in Region I

Q = heat generation per unit (total) volume at the point x,y,z

K = thermal conductivity of coolant

κ = coefficient on K

κ , the coefficient of the thermal conductivity, K , in Equation 1, is included in order to take into account the distortion of heat flow lines as they curve around the rods which, as mentioned earlier, are assumed not to transmit heat. This is discussed further in Section 2.3.

In Region II the energy transport equation is

$$\begin{aligned} \rho C_p (U)_s \frac{\partial T}{\partial s} + \rho C_p (U_{II})_z \frac{\partial T}{\partial z} = (\rho C_p \epsilon_n + \kappa K) \frac{\partial^2 T}{\partial n^2} \\ + (\rho C_p \epsilon_s + \kappa K) \frac{\partial^2 T}{\partial s^2} + Q(s, n, z) \end{aligned} \quad (2)$$

Axial conduction has been neglected. The second term on the right hand side of Equation (2) (which represents energy transport in the s -direction

[parallel to duct wall] by conduction and by eddy diffusivity) can be shown to be an order of magnitude smaller than the first term on the left hand side (which represents energy transport by convection in the s -direction). However it has been retained in this analysis for completeness.

Since the convective effects are more important than the energy transport due to mixing, in Region II, the value of ϵ parallel to the duct wall in Region II (ϵ_s) may be assumed equal to its value in Region I without any significant difference in result. The magnitude of ϵ in Region II perpendicular to the wall is also taken equal to ϵ in Region I. This was later justified by excellent match with data.

The boundary conditions coupling the two Equations (1) and (2) require the temperature and the heat flux to be continuous across the boundary between the two regions. The fluid temperature distribution at $z = 0$ must be given; and the thermal boundary condition at the container wall must be specified. In what follows, the bounding wall is assumed to be insulated and the temperature distribution at $z = 0$ is assumed uniform.

The differential Equations (1) and (2) may be written in a dimensionless form, from which useful generalized dimensionless parameters ϵ^* and C are obtained.

Region I:

$$\frac{(U_I)_z}{\bar{U}} \frac{\partial h^*}{\partial z^*} = (\epsilon^* + \kappa \alpha^*) \left(\frac{\partial^2 h^*}{\partial x^{*2}} + \frac{\partial^2 h^*}{\partial y^{*2}} \right) + Q^*(x^*, y^*, z^*) \quad (3)$$

Region II:

$$C \frac{\partial h^*}{\partial s^*} + \frac{(U_{II})_z}{\bar{U}} \frac{\partial h^*}{\partial z^*} = (\epsilon^* + \kappa \alpha^*) \frac{\partial^2 h^*}{\partial s^{*2}} + (\epsilon^* + \kappa \alpha^*) \frac{\partial^2 h^*}{\partial n^{*2}} + Q^*(s^*, n^*, z^*) \quad (4)$$

where

$C = U_S/\bar{U}$ = swirl velocity ratio in Region II

$h^* = C_p \bar{U} \{ [T(x, y, z) - T_{IN}] / (\bar{Q} \cdot d_e) \}$ Region I

$h^* = C_p \bar{U} \{ [T(s, n, z) - T_{IN}] / (\bar{Q} \cdot d_e) \}$ Region II

$\alpha^* = \alpha / (\bar{U} \cdot d_e)$

$\epsilon^* = \epsilon / (\bar{U} \cdot d_e)$

$z^* = z/d_e$

$s^* = s/d_e$

$n^* = n/d_e$

$\bar{U}$ = superficial average bundle velocity

d_e = any local characteristic dimension taken to be the hydraulic diameter of interior channels in the actual bundle.

$Q^* = Q(x, y, z)/\bar{Q}$ or $Q(s, n, z)/\bar{Q}$

$\bar{Q}$ = average heat generation per unit (total) volume

T_{IN} = uniform inlet fluid temperature

Along with the boundary conditions described earlier the Equations (3) and (4) may be solved in a differential form or in a finite difference form.

2.3 Solution of the Equations for Assemblies in Forced Convection

Equations (3) and (4) are solved by writing them in a finite difference form. A forward difference scheme is used in the axial direction. The hexagonal shape of the boundary walls suggests the selection of an equilateral triangular grid as drawn in Fig. 5, with equal lengths η^*_T between nodes. Since we are solving a set of differential equations the mesh size η^*_T can be of any convenient size. Unlike in subchannel analysis methods, ⁽²⁻⁹⁾ the spacing between nodes does not have to be taken equal to the distance between the centroids of the subchannels but it may be so taken. In region II it is possible to use more than one node in a direction perpendicular to the wall. Thus 2-dimensional temperature gradients in the wall Region II can be obtained. The details of the numerical scheme and the stability criteria are developed in Appendix 3.

To solve equations (3) and (4) the following quantities are given:

1. the bundle geometry
2. the total flow rate
3. the uniform inlet temperature at $z^* = 0$
4. the fluid properties ρ , C_p , k as functions of temperature

For a particular total flow rate, the magnitudes of $(U_I)_z/\bar{U}$ and $(U_{II})_z/\bar{U}$ are determined from the hydraulic diameter flow analysis of Novendstern¹⁶ and from experimental data (Refs. 14, 15, 20). At present there is a small difference in the analytically and experimentally determined values of these velocities (Fig. 2) as discussed in Section 1.2. The experimental data shows that the average axial velocity, over a spacer pitch, is nearly uniform in the entire bundle. The hydraulic diameter flow split shows that the wall region has a slightly higher velocity than the central region. Until more data is available, a range of values of $(\bar{U}_I)_z/\bar{U}$ and $(\bar{U}_{II})_z/\bar{U}$, spreading from the experimental to the analytical predictions has been used to determine temperature distributions. It was found that the effect of this range in flow split on exit temperature distributions of an LMFBF fuel bundle is very small¹⁷.

The unknowns that must be determined are κ , ϵ^* and C . The presence of κ in the equations may be explained as follows. In the present model the heat transferred from one point to another, at a given plane, would be overpredicted because our model assumes that there are no rods within the hexagonal duct. In actuality the presence of the rods produces a two-dimensional temperature field. The heat flow lines must curve around the rods, which results in an increase in the length of their path. This is commonly termed as 'Tortuosity'³⁰ in literature. In order to take into account this two-dimensional effect the molecular conductivity term was multiplied by a coefficient κ (obviously $\kappa \leq 1$). Although Tortuosity also affects eddy diffusivity in a similar manner as it does conductivity, we chose to include this effect in the definition of ϵ^* because it is determined empirically from data. The presence or absence of the coefficient κ would only change the magnitude of ϵ^* without changing the end result.

The value of κ can be determined for any given geometry of the fuel bundle by solving the heat conduction equation $\nabla^2 T = 0$ for the configuration shown in Fig. 6. The presence of a large number of rods, contained between the two boundaries at T_1 and T_2 ,

minimizes the effect of end conditions on the mean temperature gradient. The length of the rod array, contained between isotherms T_1 and T_2 was varied to ascertain that end effects are unimportant. For a wide range of rod diameters and rod spacings (from blanket to fuel design) it was found that κ is almost constant and is equal to approximately 1.1-1.4 times the minimum lateral porosity, λ_{L_1} , for a range of $1.07 < P/d < 1.32$, which includes all fuel and blanket geometries presently under consideration.

The parameters ϵ^* and C were determined empirically as follows. Equations (3) and (4) were programmed for computer solutions. This program is for assemblies in forced convection only and is given the name ENERGY-I. The magnitudes of ϵ^* and C were established by assuming various magnitudes and selecting those which produced exit fluid temperature distributions in agreement with available test data. For all sets of data analysed a uniform ϵ^* is assumed in the bundle.

It was observed that for all sets of data analysed, where the bundle was operating in forced convection, the enhanced eddy diffusivity term, ϵ^* , was at least an order of magnitude larger than α^* . Thus even if $\kappa = 1$, the conduction effects may be neglected as compared to the mixing effect caused by the presence

of wire wrap.

The computer program ENERGY-I requires about 20 seconds running time, on an IBM-370/168 computer, to determine the fluid temperature at all axial positions of a prototype LMFBR reactor fuel bundle. Because of the assumption on the model of visualizing the effect of the wire wrap as providing additional eddy diffusivity, the temperature distributions within the initial axial region between $z = 0$ and $z = h$ (h is the wire wrap axial pitch) are not expected to match experimental data to the same degree of accuracy as the data downstream of the first wire pitch.

2.4 Formulation for Assemblies in Mixed Convection (Both Forced and Free)

The forced convection model of Section 2.2 assumes that the axial velocities in the two regions of the bundle may differ in magnitude, but remain constant with axial position. When buoyancy effects are important the axial velocity profile would no longer be constant axially but would tend to skew towards that side of the bundle with the higher power generation (heat addition). In this case the axial velocity distribution is not constant axially in either Region I or Region II but varies with axial distance from inlet of the bundle.

The cooler fluid from regions of higher density drifts towards the hotter fluid which has a lower density. This density induced lateral flow provides an additional mechanism of energy redistribution within the assembly in the mixed convection regime. Since ENERGY-I does not take into account this buoyancy induced convection it is necessary to reformulate the equations of mass, momentum and energy, within the framework of our model, to include this effect. Now we will have the axial momentum equation coupled to the energy equation and a simultaneous solution of mass, momentum and energy equations is required to determine the velocity, pressure and thermal fields.

It is necessary to note that the basic model developed in Section 2.1 does not change for assemblies operating in mixed mode of convection. Thus the model retains its two region feature. A uniform swirl velocity in the circumferential direction is still assumed. However in Region I the eddy-diffusivity due to the presence of wires is now superimposed on a three dimensional spatially varying axial flow. In Region II, also, the axial velocity varies in a similar manner.

The exact analytical solutions are very complex to obtain as a result of which an approximate form of the equations intended for numerical solution are used. The simplifications and assumptions required to derive the approximate form of the equations, from their exact form, have been taken from the work of Chelemer et al.³¹. However, due to the presence of swirl flow in our analysis, two additional terms appear, one in the axial momentum equation and another in the energy equation. These additional terms are the axial momentum transferred by the swirl flow in the circumferential direction and the enthalpy convected by the swirl flow in the circumferential direction.

The following assumptions made by Chelemer et al.³¹ are also included in this analysis.

1. The pressure is uniform in the transverse plane at any axial level.
2. Only the axial momentum equation is considered. The enthalpy and momentum associated with the transverse flows and the magnitudes of the transverse flows due to buoyancy effects are determined in an approximate manner.

The continuity, momentum and energy equations for a node i in Region I may be written by taking a mass momentum and energy balance on an element of

area A_i and length dz associated with the node i .

The aforementioned method provides

a set of conservation equations which must be solved on a computing machine.

The continuity momentum and energy equations for a node i with an associated area A_i in Region I may be written as follows (note that since the rods are not present any suitable spacing n_T between nodes may be chosen; thus the node size is arbitrary).

Region I

Continuity

$$\frac{\partial}{\partial z} (\rho_i U_i A_i) = (\Sigma Wc)_i \quad (5)$$

Axial Momentum

$$\begin{aligned} \frac{1}{\lambda_{a1}} \frac{\partial}{\partial z} (\rho_i U_i^2 A_i) - \frac{1}{\lambda_{a1}} (\Sigma Wc)_i U_i^* = - \lambda_{a1} A_i \frac{\partial p}{\partial z} \\ - \frac{A_i \bar{c}}{\lambda_{a1}} F_{D_i} \frac{\bar{U}_i |\bar{U}_i|}{2} - \lambda_{a1} \rho_i A_i g \end{aligned} \quad (6)$$

Energy

$$\frac{\partial}{\partial z} (\rho_i U_i A_i H_i) - (\Sigma Wc)_i H_i^* = Q_i A_i + (Ex1)_i \quad (7)$$

where

$$(Ex1)_i = [\bar{G}.de.(\epsilon^* + \kappa\alpha^*) \left(\frac{\bar{H}_1 + \bar{H}_2 + \bar{H}_3 - 3\bar{H}_i}{n_T} \right)] A_{ci}$$

State

$$H_i = H(\rho_i, P_i) \quad (8)$$

where

A_{ci} = cross flow area per unit axial length
between adjacent nodes (need not be
uniform but is so taken here)

A_i = area associated with node i (superficial
area)

de_i = hydraulic diameter associated with node i
(although rods are absent their effect
must be taken into account)

F_{Di} = effective axial frictional drag/length
($=Mf/de$)

f = friction factor

$\bar{G}$ = average mass velocity in the bundle,
remains constant in the axial direction.

H_i = enthalpy of node i

K = Thermal conductivity of coolant

κ = coefficient on K , described earlier

M = Novendstern's¹⁶ friction multiplier for
wire wrapped rod bundles

P = static pressure at any axial level

Q_i = heat generation per unit volume at node i .

U_i = superficial axial velocity at i

$\bar{U}_i$ = average superficial axial velocity for
node i between z and $z + dz$

$(\Sigma Wc)_i$ = net cross-flow into node i per unit axial
length

ϵ^* = effective dimensionless eddy diffusivity

α^* = dimensionless thermal diffusivity

η = spacing between node i and adjacent nodes
1, 2, 3 (need not be equal but is taken so
here)

1, 2, 3 = adjacent nodes to node i

λa_1 = axial porosity in Region I

$\bar{H}$ = average enthalpy between levels z and $z + dz$

U_i^* = axial coolant velocity associated with the
cross flow, estimated as discussed below

H_i^* = enthalpy associated with the cross-flow

It is necessary to note that spatial density variations are taken into account and thus large density variations can be accommodated by the method. The average enthalpies for any node between z and $z + \Delta z$ is not known a priori. However since the equations are intended for numerical solution, an iterative computational scheme is used to calculate the average enthalpy and update their values at each iteration.

Chelemer et al³¹ suggest that H_i^* and U_i^* be the weighted average values determined from the average enthalpies and average velocities (between z and $z + \Delta z$) of node i and its adjacent nodes. Thus for node i

$$H_i^* = \frac{1}{2} \sum_{j=1}^3 [|X_{ij}| (\bar{H}_i + \bar{H}_j) + X_{ij} (\bar{H}_i - \bar{H}_j)] / \sum_i |X_{ij}| \quad (9)$$

where the index j refers to each node surrounding node i and the weighting factor X_{ij} represents the net gain³² in the rate of mass flow between adjacent nodes i and j . With this definition the value of H_i^* is bounded by the smallest and greatest values of $\bar{H}$ for node i and adjacent nodes j . A similar equation for U_i^* may be written in terms of $\bar{U}_i$ and $\bar{U}_j$, where $\bar{U}_i$ and $\bar{U}_j$ are the average incremental axial velocities for node i and adjacent nodes j . The corresponding equations may be written as follows for any node i in Region II.

Continuity

$$\frac{\partial}{\partial z} (\rho_i A_i U_i) = (\Sigma Wc)_i \quad (10)$$

Axial Momentum

$$\begin{aligned}
& \frac{1}{\lambda_{a2}} \frac{\partial}{\partial z} (\rho_i U_i^2 A_i) - \frac{1}{\lambda_{a2}} (\Sigma Wc)_i U_i^* + \frac{1}{\lambda_{a2}} \bar{G}.C.t. (\bar{U}_1 \bar{U}_2) \\
& = - \lambda_{a2} \frac{\partial p}{\partial z} A_i - \frac{A_i \bar{\rho}_i}{\lambda_{a2}} \frac{D_i}{2} \bar{U}_i |\bar{U}_i| - \lambda_{a2} \rho_i A_i g_c \quad (11)
\end{aligned}$$

Energy

$$\begin{aligned}
& \frac{\partial}{\partial z} (\rho_i h_i U_i A_i) - (\Sigma Wc)_i H_i^* + (\bar{G}.C.t) (\bar{H}_1 - \bar{H}_2) \\
& = Q_i A_i + (Exl)_i
\end{aligned}$$

$(Exl)_i$ = defined in equation (7a)

t = thickness of Region II. The thickness of Region II in our analysis was taken equal to $(dw + \frac{d}{2})$. The chosen thickness of Region II could deviate from this value without affecting results because the empirically determined constant C would change accordingly to yield a constant value of the product $C.t$.

where $C \equiv (U)_s / \bar{U}$ (swirl velocity ratio)

$\bar{U}$ = bundle average axial velocity

λ_{a2} = axial porosity in Region II

$\bar{H}_2, \bar{P}_2$ = average enthalpy and density between z and $z + dz$. Subscript 2 refers to node 2

adjacent to node i from which swirl flow enters node i

d_w = diameter of wire

The boundary conditions coupling the two sets of equations for region (1) and (2) require the temperature and heat flux to be continuous, as well as the density, pressure between regions (1) and (2). The fluid temperature distribution must be given at $z = 0$; and the thermal boundary condition at the container wall must be specified. It is assumed here that the inlet temperature at $z = 0$ is uniform and the duct wall is insulated. It is also assumed that ϵ^* is uniform in the entire bundle. The effect of axial conduction can be included in the term Ex_1 by connecting each node i to the axial nodes below and above it. The effect of axial conduction was initially included in the analysis. However for several runs at low Reynolds number (~ 1000) and high power generation (average temperature rise $\sim 150^\circ/\text{FT.}$) it was found that axial conduction only affects temperature distributions to a very small extent ($< 0.5\%$ of axial temperature rise). For this reason the terms coupling the node i to its neighbouring nodes above and below it have been excluded from equations (7) and (12).

The method of solution of the equations for assemblies in combined forced and free convection is given in Appendix 3. The computer program so developed was given the name ENERGY III. The unknown in the computer program are ϵ^* and C and these must be determined by analysis of free-forced convection data (Chapter 3). The ENERGY-III computer program has been used (Chapter 5) to predict the temperature distribution in a full size 61 pin blanket assembly of an LMFBR. Two different node sizes were used. When the distance between nodes n_T was equal to the centroidal distance between (126 nodes) subchannels, the computer running time was several hours. However by increasing the node size, so that only 42 nodes were used at any axial plane of the bundle, the computer running time was reduced to 9 minutes on the computer IBM 370/M168 yet good results were obtained.

The ENERGY II computer program performs similar calculations to the ENERGY III computer program except that the convective terms in the momentum equations, (left hand side of Eqs. (6) and (11)) are neglected. A large number of computer runs with a very wide range of operating conditions have shown little difference between the temperature distributions predicted by ENERGY II and ENERGY III. The computer

runs made covered both the forced and mixed convection flow regimes. Obviously the convective terms in the axial momentum equation of ENERGY III do not effect temperature distribution significantly.

A listing of the ENERGY I, II and III codes is given in Appendix 5. Table 4 shows the computer running times for determining temperature distributions in bundles of various sizes using ENERGY I, II and III.

CHAPTER 3

3.1 Evaluation of Empirical Constants of the Model by Data Analysis

The experiments described earlier in section 1.3 are used to determine ϵ^* and C the two empirical constants in the ENERGY series of codes. Apart from laser velocity measurements the three types of experiments, namely, electrolyte salt injection, hot water injection experiment, and heated pins, are used to determine ϵ^* and C . Based on physical insight and inspection of the fundamental energy balance equation we can determine that ϵ^* and C are dependent on the following parameters,

$$\epsilon^* = f(h/d, p/d, Re, Pr)$$

$$C = f(h/d, p/d, Re, Pr)$$

Since these experiments have been performed for assemblies of different geometries and at various Reynolds numbers it is possible to determine these functional relationships for ϵ^* and C .

The heated pin experiments provide a boundary condition which represents the actual LMFBR fuel assembly heat transfer boundary condition. The parameters ϵ^* and C in ENERGY are varied until coolant temperatures predicted by the code match the experimentally measured temperatures. It was found that ϵ^* and C can be found independently by matching the central region and wall region temperature maps respectively. Apparently, swirl flow effects, fortunately, do not appear to penetrate significantly within the fuel assembly. Thus ϵ^* can be determined from the data in the inner region and C from the data on the wall region.

In the analysis of the data it is assumed that the flow is fully developed in the test section. This is generally true because the experiment designer provides an entrance region to the test section where the flow becomes fully developed. Even if the flow is not fully developed at the inlet to the test section it is expected to be fully developed within a wire wrap pitch. This has been shown in Ref. 14 for a typical LMFBR bundle with a severe inlet velocity skew. The ENERGY calibration of ϵ^* and C should be unaffected due to entrance effects because only the exit temperature or concentration distribution is used for calibration purposes.

In order to show that both the salt injection and hot water injection experiments would also yield a mixing

coefficient which can directly be used in the ENERGY code for LMFBR assembly temperature calculations, it is necessary to realize that ϵ^* predominantly represents the wire sweep flow mixing effect. Thus the mixing coefficients determined from equations (13) and (14) below need not be modified (as is normally done in literature when mass eddy diffusivities are converted to heat eddy diffusivities or vice versa) in any manner for use in ENERGY.

The equations governing salt diffusion in salt injection experiments and heat diffusion in hot water injection experiments along with injection boundary conditions are given below (note: we may only concern ourselves with the steady state equation),

$$\text{Hot Water Injection: } \frac{DT^*}{Dt} = \frac{1}{Re} \left(\frac{\epsilon}{\gamma} + \frac{1}{Pr} \right) \nabla^2 T^* \quad (15)$$

$$\text{when } z = 0, \quad T^* = 1 \quad \text{where} \quad T^* = \frac{T - T_{COLD}}{T_{HOT} - T_{COLD}}$$

Water is generally used as the working fluid. One finds $\frac{\epsilon_H^*}{\gamma} \gg \frac{1}{Pr}$.

$$\text{Salt Injection: } \frac{DC^*}{Dt} = \frac{1}{Re} \left(\frac{\epsilon_D}{\gamma} + \frac{1}{Sc} \right) \nabla^2 C^* \quad (16)$$

when $z = 0$, $C^* = 1$ where $C^* = \frac{C}{C_0}$ and $C_0 =$ inlet concentration.

For all tracers utilized in wire wrap studies independent of the particular tracer $\frac{\epsilon_D^*}{\gamma} \gg \frac{1}{Sc}$. For reasons discussed before one expects $\epsilon_D^* = \epsilon_H^*$.

3.2 Analysis of Forced Convection Data by ENERGY-I

The forced convection data was analysed by using the ENERGY-I code. The ENERGY-I code requires, as input, the flow split between regions I and II. As discussed in section 1.3 and shown in Fig. 2 there is a discrepancy between the predicted¹⁶ and experimentally observed flow splits. The available data^(14, 15, 18, 19, 20) on flow split was obtained only for the wall channels of a wire wrapped bundle.

The flow split can be determined by (a) completely neglecting geometrical tolerance, and (b) allowing for geometrical tolerance. In reality, geometrical tolerance effect exists and must be taken into account. Following the method given by Hanson in Ref. 20 (appendix) if T is the diametrical tolerance across the flats of a bundle and a fraction $(1 - F)$ in the wall region. Then the pitch, p , of the wire wrapped bundle must be increased

by Δp due to the "looseness" cause by geometrical tolerance, where

$$\Delta p = TF/(\sqrt{3} \cdot N_{\text{RINGS}})$$

The gap between the wire and the wall is given by

$$\Delta R = \frac{T}{2} (1 - F)$$

The two extreme cases for tolerance distribution are

$$\text{Case I: } F = 0 \quad \Delta p = 0 \quad \Delta R = T/2$$

$$\text{Case II: } F = 1 \quad \Delta p = T/(\sqrt{3} \cdot N_{\text{RINGS}}) \quad \Delta R = 0$$

The actual tolerance will lie somewhere in between the two extreme cases mentioned above.

The effect of these two extreme cases on flow split in the wall channel are shown in Fig. 2 for the MIT 61-pin bundle which has a diametrical tolerance of 0.0195 in. in a duct with a distance across the hexagonal

flats of 2.560 in.

In most bundles it is expected that Case II will exist where all the tolerance is accommodated within the bundle. Observation of the (Ref. 20) M.I.T. bundle agrees with this suggestion as the pins were observed to contact the duct faces when the bundle was assembled. This was also observed in the AI 217 pin bundle (Ref. 33). Wherever there is a doubt as to which of the two limiting cases exists within a bundle, we have shown the two limits of flow splits for wall channel corresponding to the two cases.

It is interesting to note that for the 7-pin bundle it was possible to predict both the wall channel and inner channel flow splits accurately. As the bundle size increased there appears to be a divergence between theoretical predictions and experimentally observed data on a similar geometry. Allowing the experimental predictions to have an associated error bar, it still appears that the theoretical flow split predicts higher axial velocities in the wall channels (and lower axial velocities in central and corner channels) than experimentally observed. If the divergence in the theoretical and experimental curves is in fact real then a correction factor on hydraulic diameter alone would not bring the two curves together. A bundle size effect exists in that

case. If a bundle size effect exists it could be only due to a change in the basic flow field in going from a 7-pin to a 217-pin bundle.

The method of Novendstern¹⁶ for determining flow division in the subchannels of a LMFBR bundle is briefly described below and it is shown how one can use this to determine the flow split between regions I and II of our model.

Novendstern¹⁶ considers a bundle with NN1 interior channels, NN2 wall channels and NN3 corner channels. Since the pressure drop in all subchannels is the same, the velocity in each subchannel can be determined using the friction factor relationship. Thus for channels of type 1 (interior), type 2 (wall) and type 3 (corner) one can write

$$\frac{f_1}{de_1} \frac{\rho v_1^2}{2} = \frac{f_2}{de_2} \frac{\rho v_2^2}{2} = \frac{f_3}{de_3} \frac{\rho v_3^2}{2} \quad (17)$$

Assume f the friction factor is given by $f = \frac{C}{Re^m}$, and substituting into the equation (17)

$$\frac{v_1^{2-m}}{de_1^{1+m}} = \frac{v_2^{2-m}}{de_2^{1+m}} = \frac{v_3^{2-m}}{de_3^{1+m}} \quad (18)$$

The continuity equation is,

$$NN_1 V_1 A_1 + NN_2 V_2 A_2 + NN_3 V_3 A_3 = \bar{V} A_T \quad (19)$$

then from (18) and (19) eliminate V_2 and V_3 to get

$$\frac{V_1}{\bar{V}} = \frac{A_T}{[NN_1 A_1 + NN_2 A_2 \left(\frac{de_1}{de_2}\right)^\alpha + NN_3 A_3 \left(\frac{de_3}{de_2}\right)^\alpha]} \quad (20)$$

Similar equations for $\frac{V_2}{\bar{V}}$ and $\frac{V_3}{\bar{V}}$ can be

easily derived. Here

$$\alpha = \frac{1 + m}{2 - m}$$

The V 's are the actual fluid velocities.

In the ENERGY series of codes the flow splits

$$\frac{U_1}{\bar{U}} \quad \text{and} \quad \frac{U_2}{\bar{U}}$$

at the inlet must be specified by the user. In order to determine these, one may use Novendstern's method for subchannels as follows. For Region I the flow split is the same as for Novendstern's central channels, i.e.

$$\frac{U_1}{\bar{U}} = \frac{V_1}{\bar{V}} \cdot \frac{\lambda_{a_1}}{\bar{\lambda}_a}$$

if the boundary of Region I is taken to pass through the centers of rods adjacent to the wall. For region II the flow split $\frac{U_2}{\bar{U}}$ may be calculated from equation 21.

$$\frac{U_2}{\bar{U}} = \left[\left(\frac{V_2}{\bar{V}}\right) F_2 + \left(\frac{V_3}{\bar{V}}\right) F_3 \right] \frac{\lambda_{a2}}{\bar{\lambda}_a} \quad (21)$$

where

$$F_2 = \frac{NN_2 \cdot A_2}{NN_2 A_2 + NN_3 A_3} \quad F_3 = \frac{NN_3 \cdot A_3}{NN_2 A_2 + NN_3 A_3}$$

λ_{a2} = Average axial porosity of region II

λ_a^- = Average axial bundle porosity

Until the question of the bundle size effect is fully resolved one needs to take into account the flow split effect discussed earlier while analyzing other data and in predicting temperatures for the full size bundle at power conditions. The ENERGY-I analysis of data has taken into account the uncertainty in flow split in calibrating the two constants ϵ^* and C . The effect of this flow split uncertainty on the constants is discussed explicitly only for those cases where it was found to have a measurable effect.

Tables 2 and 3 show the forced convection data analysed. Table 2 lists the data based on bundle size and experimental techniques and Table 3 based on the significant geometrical characteristics. The data covers a wide range of geometry useful for thermal-hydraulic optimisation studies.

Since Ref. 17 reports the detailed analysis results for each set of data in Tables 2 and 3, only the detailed

analysis of two sets of data will be repeated here.

The two sets of data analysis reported here are the ORNL 19-pin FFMIIA heated pin experiments and the GE 127 pin hot water injection experiments. The two data sets were chosen because they represent two different types of experiments on bundles of considerable difference in size. Moreover, the ORNL experiments are very valuable because data for a very wide range of operating conditions was obtained, including data in the mixed convection regime. The results of calibrating ϵ^* and C for all sets of data will be reported here.

3.2.1 Analysis of ORNL - 19PIN FFMIIA Bundle (Ref. 21)

Analysis of the portion of the ORNL data which is in forced convection will be reported here. The analysis of ORNL data in mixed convection is reported in the section 3.3.

The Oak Ridge National Lab. FFM-IIA data was taken in a 19 pin bundle with the geometry similar to the FFTF bundle. Fig. 7 shows a cross-section of the wire-wrapped bundle. Liquid sodium was used to cool the electrically heated pins. The heated length of the rods is 21 in. The coolant enters the bundle approx. 6 in. below the plane where the rods are heated in order to provide a fully developed flow field at the entrance to

the heated section. Thermocouples are located in the rod walls and also in the wire wraps. An exit rake is located 3 in. above the exit of heated bundle and is instrumented with thermocouples which directly measure coolant temperatures at the center of various subchannels. The axial power profile is uniform. The hexagonal duct wall is surrounded by guard heaters to minimize heat losses.

Many data sets are reported for various power skews (Fig. 7) and for Reynolds numbers from about 1000 to 70,000.

The analysis of the ORNL data was performed first for the case of all heated pins at various power skews. The ENERGY-I equations were solved with the following additional assumptions:

a) k , ρ , C_p , μ were evaluated at the average temperature over the flow cross-section and taken as uniform at each axial position; they varied and were re-evaluated at successive axial positions.

b) A uniform value of ϵ^* was assumed throughout the bundle.

For each of the test runs the best ϵ^* and C was determined by choosing the magnitudes which gave a calculated exit fluid temperature distribution that agreed with the measured value. Fig. 8, 9, 10, 11 show

the data and the calculated temperature profile. It was found that C only affects the edge row temperatures. Therefore ϵ^* was evaluated directly by using the data within the bundle. Then C is varied until the best agreement with data is obtained in the outer region. The magnitude of ϵ^* is shown in Fig. 12. It was found that both ϵ^* and C are independent of Reynolds number for $Re > 15000$. The optimum values of ϵ^* and C were found to be 0.017 ± 0.005 and 0.085 ± 0.015 respectively from various sets of data where all pins are heated. Thermocouple 20 was not shown in Figs. 8-11.

It is interesting to note that the same ϵ^* matches data for various power skews. This result was anticipated and confirms our physical interpretation of ϵ^* as a parameter dictated by bundle geometry.

The major problem encountered was in analyzing the single heated pin data. As discussed in Ref. 1 (see Fig. 13) computer programs like THI-3D and COTEC have also not been able to predict this data satisfactorily. The single pin tests, however, raise several interesting questions, which cannot be fully resolved without additional experimental data. The large asymmetry in radial temperature profile for a centrally heated pin can be attributed to either an internal swirl flow (i.e. due to only 2 rows of pins the wall circumferential flow is

able to penetrate almost to the center of the bundle) or due to bowing of the heated pin. The single pin tests were conducted at 8-10 kw/ft. Distortion of the FFM bundle under power and flow has been observed via X-ray photography and image enhancement technique. The distortion appeared to be power dependent.²¹ These radiograph tests were made above 8 kw/ft, at which power level the spacing between the wall and the rod appeared to vary in the axial direction. On one side of the bundle the rod to wall gap increased and on the opposite side it decreased. This change of wall to rod gap in the circumferential direction should not affect C since the latter is a space average quantity. However, if individual heater pins distorted in the axial direction at these power levels, then asymmetry in radial temperature distribution for a centrally heated pin should be expected.

Only for large internal swirl flow can such severe asymmetries in temperature exist. If large internal swirl exists then the inherent assumptions in ENERGY should prevent it from matching the radial temperature profiles for the 20% and 300% power skew cases. For this reason it is believed that if internal swirl exists at all it is small in magnitude and it does not affect temperature distribution at the exit of the bundle. It

is our interpretation that the severe radial temperature asymmetry for the centrally heated pin run is primarily due to rod bowing. Other ORNL single heated pin runs were also analyzed. The runs with single heated pins at the wall were also used to determine C. The predictions for these runs matched with data much better than for the centrally heated pin case.

The accuracy of the coolant thermocouple was assumed to be $\pm 4^{\circ}\text{F}$. The thermocouples located in the rod walls and in the wire wrap gave a local temperature which was difficult to use to compare with ENERGY predictions of subchannel averaged temperatures.

3.2.2 Analysis of GE-127 Pin Hot Water Injection Tests (Ref. 24)

The 127 pin bundle was 72 in. long and contained a hexagonal flow duct. The diameter of the pins is 0.25 in., p/d and h/d are 1.28 and 24 respectively. The Reynolds number of the test was about 55000. Hot water is injected and mixing rates were inferred from temperature measurements downstream. Fig. 14 shows the main test loop. The loop temperature was maintained at 100°F . Approximately 2% of the main flow was diverted from the loop, just upstream from the main pump, heated in a 900,000 Btu/h boiler to 161.4°F , increased in pressure

up to 125 psi with a small 2 hp jet pump and reinjected into the center of the bundle. Fig. 15 shows the cross-section of dummy fuel bundle. Lateral dispersion of hot water was inhibited for a length of one inch above the injection tube by a "dam" in the form of a filler rod welded to each of six rods surrounding the central injection tube. The flow rate in the injection flow channel is approximately made equal to the flow rates in the 6 central channels. Care was exercised in the adjustment of injection and main flow rates during experimentation so as to minimize any localized lateral dispersion due to mismatch in flow velocities.

Lattice temperature measurements downstream from the hot water injection point were measured at every 1 in. axial distance from 6 in. to 65 in. Fig. 16 shows the lattice temperature probe assembly. This probe assembly was designed to fit within the tube which simulated fuel rods and could measure the temperature of the moving fluid in any one of the six channels surrounding the instrumented rod without perturbing the flow. The probe assembly could be moved along the 72 in. length of the dummy fuel rod and could be rotated within any particular tube so as to orient the thermocouples preferentially in any direction. In order to confirm the directional temperature sensitivity of the

probe assembly two dimensional heat transfer calculations were done on the probe assembly, dummy fuel rod and water environment for several assumed circumferential temperature distributions. One severe distribution is shown in Fig. 17. In this case the thermocouple response was calculated to be 95°F showing considerable directional sensitivity.

It is obvious from location of the injection position (Fig. 15) that estimation of swirl flow in the wall regions cannot be made from this setup. Therefore the effort was concentrated on determining for the particular geometry and flow conditions.

In order to determine ϵ^* for salt and hot water injection experiments, the following procedure was adopted. The temperature rise for various channels is determined from the ENERGY code by varying the input parameter ϵ^* . Fig. 18 shows a plot of temperature rise vs. ϵ^* for several channels, at the 12 in. axial level and at the 36 in. level. On the same figure the recorded data is plotted showing a $\pm 1^{\circ}\text{F}$ spread due to assumed thermocouple accuracy. It is interesting to note that the slope of the curve in Fig. 18 for the injection channel at 12 in. is large but for other channels the slope is not large at 12 in. At 36 in. the slope of all the curves are small. The only meaningful

estimate of the range of ϵ^* can be obtained from the injection channel data at 12 in. For other channels, when a $\pm 1^\circ\text{F}$ spread on data is superimposed on the plotted computer curves, it is seen that almost any value of ϵ^* should predict a temperature rise within the data range. Thus usefulness of the data 6 to 14 in. beyond injection is limited. The value of ϵ^* so determined as found to lie between 0.025 and 0.03. The procedure described above is only the first step in data analysis. The next step, a refinement on Step I, is described below.

Since at the injection point there is a perturbation of the main flow stream due to injection of salt or hot water, it is quite conceivable that these perturbation effects may last several inches downstream of injection point. The data is then analyzed by taking the inlet plan in the computer program to be several inches downstream of the injection point in the experiment. For this data the inlet plane was assumed to be 12 in. downstream of injection point and data at this 12 in. location was assumed as the initial channel condition. Fig. 19 shows the data for the injection channel and also the predictions of ENERGY for 5 different cases. Set I shows the predictions for the case where the inlet to ENERGY is at the same plane as the injection point

For other curves the 12 in. level in the experiment is the starting point for the calculations. It is seen that curves I and II do not match although ϵ^* equals 0.031 for both cases. This was found to be due to lack of energy balance in the data between inlet and 12 in. level. The total energy at the 12 in. level, as calculated by taking the sum of the product of flow and enthalpy over all channels, was found to be greater than the total energy in the injection plane. The data at 12 in. is probably more reliable (as discussed below) and free of any inlet perturbation effects. The inlet energy balance was corrected to match that at the 12 in. level. On superimposing a $\pm 1^\circ\text{F}$. error band on the data ϵ^* was found to be in the range 0.02 to 0.035 or an approximate average of 0.0275 ± 0.008 .

The data after the 12 in. axial level is probably more reliable because of the following reason. It was found that the radial temperature distribution across the bundle had a peak which rotated circumferentially as the fluid travelled up the bundle. Perhaps an internal swirl had generated due to slightly asymmetric injection into the six inner channels. This internal swirl appears to die off at about 12 - 18 in. when the radially peak temperature stops rotating. Thus as one proceeds axially downstream the reliability of the

data increases but its sensitivity to ϵ^* decreases (Fig. 18). Therefore the 12 in. level data was chosen as a reliable set of data to start calculations and yet maintain reasonable sensitivity to ϵ^* variations.

Analysis of all available mixing data on wire-wrapped bundles in forced convection is reported in Ref. 17. These include several data sets specifically directed towards obtaining an estimation of the swirl flow in the wall region. A few of the general conclusions drawn are as follows. For bundles operating under forced convection, the mixing coefficient ϵ^* does not appear to be a function of Reynolds number. Additionally no dependence of C on Reynolds number was found. However analysis of swirl data shows a dependence of C on bundle size. No firm explanations are yet available for this. This bundle size effect could be attributed¹⁷ to experimental errors; however more experimentation is required before one can conclude if the bundle size effect does exist or if experimental errors are giving misleading results. Some suggestions for further experiments needed to resolve this are given in Chapter 6.

Fig. 20 shows that the variation of ϵ^* with $1/(h/d)$ is linear in the range of interest. As the wire-wrap pitch is increased ($d/h \rightarrow 0$ or $\frac{h}{d} \rightarrow \infty$) the ϵ^* vs. d/h

plot is expected to curve and tend to an asymptotic value for the case of no wire-wrap. In that case only the natural turbulence effect would contribute to ϵ^* . It can also be seen in Fig. 20 that for a given d/h , or $(1/(h/d))$ ϵ^* is maximum for a p/d of about 1.25.

Variation of C with d/h is shown in Fig. 21. It was found that the effect of p/d on C was not significant enough to be discernable and hence only the effect of d/h is shown in Fig. 21. As d/h increases, it was found that the scatter in data also increases. The only available data for C for large values of d/h are the SKOK and M.I.T.²⁰ data.

For practical use the recommended value of ϵ^* is the value read from Fig. 20 with approximately a $\pm 30\%$ (of ϵ^*) error bar. At $d/h \sim 0.02$, $p/d \sim 1.24$, the range shown in Fig. 20 is 0.025 to 0.048. Wherever the range is not shown the above mentioned range $\pm 30\%$ of ϵ^* may be used. For C the range shown in Fig. 21 must be used.

The analysis of the SKOK data of Ref. 22 is described elsewhere¹⁷. Since only data over one axial wire wrap pitch is given in Ref. 22, our confidence level

in analyzing it is not good. The values of C obtained from the data of Ref. 22 required many assumptions. It appeared that C so obtained was underestimated. Thus the SKOK data is shown as a lower limit on C . The M.I.T. results on the other hand appear to overpredict C . This is because the transverse velocity was measured in only one position which was centrally located in the gap between rod and wall. The average gap velocity could be as much as 20% lower than the point velocity measured if the flow field is turbulent and as much as 50% lower if the local flow field is laminar. Superimposed upon this is an error bar of $\pm 15\%$ in point measurement of velocity using an LDA. The value of C lies between the limits shown in Fig. 21.

A correlation was obtained for

$$\epsilon^* = f(h/d, p/d) \text{ and } C = f(h/d)$$

based on the results of data analysis shown in Figs. 20 and 21. These correlations must be used only for bundles in forced convection and with the following additional restrictions:

(a) all wire-wraps of the same diameter wrapped in the same sense from the same start position.

$$(b) \quad 14 \leq h/d \leq 52$$

$$(c) \quad 1.14 \leq p/d \leq 1.32$$

The correlations for ϵ^* and C are as follows:

$$\epsilon^* = [0.85 - 12.8 \{|p/d - 1.25|\}^{1.424}] (d/h) \quad (22)$$

Data correlated within $\pm 20\%$ at a 95% confidence level.

$$C = 0.09 + 2(d/h - 0.03) \quad (23)$$

Data correlated within $\pm 35\%$ at a 95% confidence level.

Fig 22 shows how well the correlation matches the points obtained by data analysis. An advantage of obtaining these correlations is that the functional relationships for ϵ^* and C can be directly built into the ENERGY-I code.

The value of ϵ^* and C obtained from the correlation may be different from their values obtained by data analysis (Fig. 22). Therefore it would be meaningful to take the nominal values of ϵ^* and C from the correlation along with their expected deviation from nominal, and using this range of empirical constants, determine the extreme range of predictions for the original data set. Figs. 23-28 show one typical run from each data set.

Fig. 21 shows several data points above the line representing the maximum permissible limit of C . The tangent of the wire wrap angle is taken to represent the upper limit to C because of the following reasons. In our model C is taken to be uniform along the circumference of the duct at any axial level. Since the wire wrap provides the driving force for the swirl flow it is difficult to imagine the average circumferential swirl flow ratio exceeding the tangent of the wire wrap angle. A possible explanation is given below for those data points that lie above the line representing the upper limit to C .

Each set of data has an associated error bar. Superimposed upon this is an additional, although relatively smaller,

error bar due to the assumptions made in reducing and analysing the data to obtain C . Only a few of the error bars are shown in Fig. 21 to maintain clarity. The error bars extend well below the upper limit to C showing that the data is valid within experimental limits of accuracy. The correlation for swirl flow, however, lies below the upper limit of C in the entire design range of d/h values for fuel and blanket assemblies. Since the circumferential temperature distributions are relatively insensitive to C it is recommended that the correlation for C be used in design calculations until more data is available.

3.3 Analysis of Mixed Convection Data by Energy II and III

In the mixed convection regime buoyancy effects become important especially if the bundle is subjected to high power skews. Consequently the density differences within the bundle would lead to a velocity distribution which is not uniform in either region at any axial level. The ENERGY-I code which assumes uniform velocities in each region, therefore, cannot be used to determine temperature distribution for bundles in mixed convection. In addition to the various means of energy transport described in Appendix I, there is another mechanism for bundles in mixed convection. This is the net drift of

flow from regions of higher density to regions of lower density. Thus cooler fluid would be induced, by the buoyancy effect, to be transported to the hotter regions thereby reducing temperature gradients within the bundle. It is obvious that since the ENERGY-I code does not account for the buoyancy induced convection in the transverse direction, it should predict much steeper temperature gradients across the assembly than the ENERGY II and ENERGY III codes. Alternatively, the mixing coefficients ϵ^* and C might have to be increased unrealistically in order to get the same temperature distribution as obtained for a bundle in mixed convection.

Both the computer programs ENERGY II and ENERGY III were used to analyse the ORNL FFMIIA data²¹ between Reynolds numbers 1100 to about 4500.

These Reynolds numbers are based on the hydraulic diameter of Region I, as the characteristic dimension. As discussed in Chapter 4, Reynolds number alone is not sufficient to determine whether buoyancy effects are important or not. Power skews, mixing coefficient ϵ^* , bundle geometry, bundle power are some of the other important parameters which determine if buoyancy plays an important role.

Table 5 shows the main features of ORNL runs in which the buoyancy effects are expected to become important. The two unknowns in our method are ϵ^* and C and these must be determined by analysis of this data. One would expect that ϵ^* and C at low Reynolds numbers should have the same values as at high Reynolds numbers, since the basic phenomenon of flow sweeping modeled by ϵ^* and C does not change. Moreover thermal conduction effects may become important enough to obscure the effect of ϵ^* , in which case a wide range of ϵ^* would predict the temperature distribution. This is indeed found to be true.

Although the following analysis of ORNL data (Table 5) was done with ENERGY-III, the results obtained by ENERGY-II are very similar for a wide range of power skews and flow rates.

Figs. 29-36 show the exit temperature distribution for the ORNL data of Table 5. Values of ϵ^* and C used were obtained from figures 20 and 21 for the geometrical conditions of the ORNL bundle. Reynolds number less than 1500 thermal conduction effects become so important that a wide range of ϵ^* can be used without changing the prediction of the exit temperature distribution. As Reynolds number is increased above 1500 one finds a gradual decrease in this ϵ^* range. Fig. 37

shows the variation of ϵ^* vs Reynolds number for the entire range of ORNL data. The vertical lines on this figure are the permissible range of variation of ϵ^* for which the maximum change in the exit temperature for any channel would not exceed $\pm 6^\circ\text{F}$. The number $\pm 6^\circ\text{F}$ was chosen to represent the estimated maximum error made in measuring the fluid temperature by a thermocouple. It is interesting to note that within the limits of measurement ϵ^* appears to be independent of Re .

Figs. 38 and 29 show the development of the velocity profiles for ORNL runs 1416 and 1101 at Reynolds number of 1100 and 2480 respectively. The velocity profiles for both runs skew towards the higher power side of the bundle. However, for run 1416 the maximum skew in velocity profile occurs at about $3/4$ of the heated length whereas for run 2480 it occurs at the exit of the bundle. This result is similar to that obtained by Tsai et al.³⁴. The magnitude of the maximum velocity skew (peak/avg.) shown in Figs. 38 and 39, is considerably less than obtained by Tsai et al. This is because these authors did not consider the effect of frictional drag, due to the presence of wire wraps and fuel rods, in their model. Frictional drag reduces the skewness in velocity profile considerably. Thus, for an assembly operating at a given power and for a given power skew and flow rate, the smaller the drag force, the larger would be the

velocity skew and hence the larger the effect of buoyancy. It is for this reason, perhaps, that Tsai et al.³⁴ find buoyancy effects to be important up to Reynolds numbers of 6000 although their power skew and flow rates are similar to the ORNL data. The present analysis of the ORNL data shows that buoyancy effects are not important above Reynolds number of about 4400. By reducing the frictional drag to a small fraction of its original value for run 1416 it was found that reverse flow begins in certain areas of the bundle. However ENERGY III as yet cannot calculate a reverse flow condition since the equations are not in a form suitable to handle reverse flow situations.

Comparison of ENERGY III prediction with the prediction made by the THI-3D code⁹ for ORNL run 1416 shows an interesting result (Fig. 30). Both codes predict the wall channel 37 temperature well. However, the temperatures for wall channels 40, 31 and 32 are under predicted by both codes. Several possibilities which may explain this were examined. The axial drag coefficient was varied by $\pm 30\%$ without any significant change in wall temperatures although the temperatures in region I were affected considerably. The change in inlet flow distribution had almost no effect at all on exit temperature distribution. Finally, it is possible that

residual bowing from previous runs might have been present in the bundle and effected the experimental temperature distributions. This is a possibility since bundle bowing was observed in the ORNL bundle by radiograph tests²¹ during previous runs.

Chapter 4

4.1 Criterion to Determine when Buoyancy Effects Become Important

The computational methods used for reactor assembly thermal-hydraulic calculations are considerably more complex if buoyancy effects are important. As discussed in Chapter 2, the ENERGY-I code can be used when free convection effects are unimportant. However when free convection effects become important the ENERGY II and ENERGY III codes must be used. The ENERGY II and ENERGY III codes can always be used for forced convection calculations but the computational time for these codes, for a 61-pin blanket assembly is at least fifteen fold greater than for ENERGY-I. Therefore, ENERGY-I should be used for forced convection calculations. Since blanket assemblies may operate in either forced or mixed convection flow regimes, a designer must know which computational method should be used. A criterion is developed here which permits one to determine when buoyancy effects become important.

Previous works⁽³⁸⁻⁴⁹⁾ have used the conventional Grashof number as a criterion to determine when buoyancy effects become important. However most of

the work reported in literature on free and mixed convection in tubes and rod bundles is not directly applicable to a wire wrapped LMFBR assembly operating in a combined free and forced convection regime.

The boundary condition assumptions of constant heat flux or uniform wall temperature has been used extensively in most previous work. In an LMFBR bundle there is an axial as well as radial power skew which complicates the analysis. In addition the rods and the wire wraps increase the frictional drag and the wire wraps promote mixing. Both these effects have a bearing on whether buoyancy is important. Previous works do not consider these and as a result the criterion for predicting when natural convection effects become important in a rod bundle will be different from the conventional Grashof number.

Imagine an LMFBR wire wrapped bundle operating at a given power in steady state with the following operating characteristics:

- a) average axial temperature rise across the bundle of height L is, $(T_M - T_{IN})$.
- b) radial power skew (peak/avg) is $= \hat{Q}_p$
- c) average actual axial velocity in bundle $= V$
- d) duct flat to flat distance $= D$

- e) Average Reynolds number based on any local characteristic dimension, taken here as d_e , (the hydraulic diameter of an actual interior channel) = Re

The power skew would cause the temperature of the fluid on that side of bundle with higher heat generation to rise more rapidly than the fluid on the opposite side of the bundle. This sets up a transverse density gradient in the bundle which results in a net drift of flow towards the side of the bundle with higher heat generation. The resulting velocity profile due to this buoyancy effect is skewed. Thus superimposed on the forced convection is the aforementioned free convection.

It is expected that for a wire wrapped assembly there is a dimensionless quantity, analogous to conventional Grashof number, which is small when buoyancy effects are negligible. To search for the proper form of this dimensionless quantity the ratio of buoyant to frictional forces in the equation of motion is examined.

$$\frac{\text{Buoyant forces}}{\text{frictional forces}} = \frac{g\beta[T - \bar{T}(z)]}{F_D V^2} \quad (24)$$

where

$T = T(x,y,z)$ the temperature of fluid at

(x,y,z) in the bundle

$\bar{T}(z)$ = mean fluid temperature at height z above

inlet

$F_D = Mf/de$

M = Novendsterns¹⁶ multiplication factor

for wire wrapped bundles

$$M = \left(\frac{1.034}{(p/d)^{0.124}} + \frac{29.7(p/d)^{6.94} Re^{0.086}}{(h/d)^{2.239}} \right)^{0.885}$$

In order to predict buoyancy effects in an LMFBR bundle a conservative criterion is required. The drag or frictional forces do not change significantly along the length of the bundle, however the buoyancy forces are expected to be a maximum at the exit of the bundle. In order to evaluate the ratio of buoyant to frictional forces it is necessary to estimate the magnitude of the temperature difference (of Eq. 24) in terms of quantities given for the system. For a special case where this can be done simply is the limit where there is no mixing within the bundle. Then one can easily show that at the exit of the bundle, the maximum temperature T is related to the mean temperature T_M by the equation

$$\frac{T - T_{IN}}{T_M - T_{IN}} = \hat{Q}_p$$

or

$$\frac{T - T_M}{T_M - T_{IN}} = \hat{Q}_p - 1$$

Using the above equation with equation (24),
the ratio of

$$\frac{\text{buoyant forces}}{\text{frictional forces}} = \frac{g\beta(\hat{Q}_p - 1)(T_M - T_{IN})}{F_D V^2} = Gr^N (\text{say})$$

For the particular case, of no mixing, it is expected that Gr^N , the modified Grashof number indicates when buoyancy effects are important. When it increases we expect buoyancy effects to become more important. It is expected that we will find a critical Grashof number above which buoyancy effects are important and below which buoyancy effects are unimportant regardless of the bundle operating and geometric characteristics.

Then

$$Gr^N = \frac{g\beta(\hat{Q}_p - 1)(T_M - T_{IN})de^3}{Mf Re^2 \gamma^2} \quad (25a)$$

or

$$Gr^N = \frac{Gr}{Mf Re^2} (\hat{Q}_p - 1) \quad (25b)$$

where

$$Gr = \frac{g\beta(T_M - T_{IN})de^3}{\gamma^2}$$

As the radial peak to average power skew is increased or conversely the friction factor, Reynolds number or M reduced, the buoyancy effects should become more important. Thus Gr^N may provide a means of determining when buoyancy effects become important provided one can find a critical Grashof number for no mixing, Gr_C^N , which is a constant for all values of Gr , bundle geometry, peaking ratios.

In deriving equation (25) it was assumed that no mixing occurs within the bundle. In wire wrapped assemblies, both mixing caused by flow sweeping and thermal conduction would tend to reduce transverse temperature gradients and hence the buoyant forces, as defined in equation (24). If mixing and conduction effects are neglected the criterion developed to predict onset of mixed convection may be very conservative.

To derive a criterion which takes mixing effects into account one must write $(T - \bar{T}(z))$ in equation (24) in terms of an expression that involves the mixing terms ϵ^* and α^* . In order to derive such an expression one imagines the existence of rods within the bundle and two channels (one hot, one

colder) in the bundle at a distance η apart. These channels may or may not be equal to two subchannels. For convenience let us assume these are two subchannels since the following derivation is independent of our model. The subchannels each have a mass flow rate $\dot{m}$, an area A and a cross-flow area per unit length of A_c . The heat generation in the hotter channel is greater than that in the other channel. Denoting the two channels with the symbols 1 and 2 respectively we have the energy equations for

Channel 1

$$\dot{m} \frac{dH_1}{dz} = Q_1 A + A_c \rho (\epsilon_H + \alpha) \frac{(H_2 - H_1)}{\eta} \quad (27)$$

Here ρ is the average density at any axial level and is taken uniform for the two channels. The energy equation for Channel 2 is

$$\dot{m} \frac{dH_2}{dz} = Q_2 A + A_c \rho (\epsilon_H + \alpha) \frac{(H_1 - H_2)}{\eta} \quad (28)$$

where

$H_1(z), H_2(z)$ = enthalpy at position z

Q_1, Q_2 = local heat generation per unit
volume

$$\frac{Q_1}{\left(\frac{Q_1 + Q_2}{2}\right)} = \hat{Q}_p = \text{Peak to average skew}$$

It is assumed that the mass flow rate is $\dot{m}$ in both channels. This assumption is not necessary to make in this derivation. However, since considerable simplifications result, it was decided to do so. Then eliminating H_2 , by combining Eqs. (27) and (28), one finds

$$\frac{d^2 H_1}{dz^2} + 2\gamma \frac{dH_1}{dz} = (q_1 + q_2)\gamma \quad (29)$$

where

$$\gamma = \left(\frac{A_c}{A} \right) \left(\frac{de}{\eta} \right) (\epsilon^* + \alpha^*)$$

$$q_1 = \frac{Q_1 A}{\dot{m}}$$

$$q_2 = \frac{Q_2 A}{\dot{m}}$$

similarly

$$\frac{d^2 H_2}{dz^2} + 2\gamma \frac{dH_2}{dz} = (q_1 + q_2)\gamma \quad (30)$$

solving equation (29) and (30) with the appropriate boundary conditions it can be easily shown that

$$(H_1 - H_{IN}) = \frac{1}{4\gamma} (q_1 - q_2) (1 - e^{-2\gamma z}) + \frac{q_1 + q_2}{2} z \quad (31)$$

$$(H_2 - H_{IN}) = -\frac{1}{4\gamma} (q_1 - q_2) (1 - e^{-2\gamma z}) + \frac{q_1 + q_2}{2} z$$

$$\text{then } \bar{H}(z) - H_{IN} = (h_1(z) + H_2(z))/2 = \left(\frac{q_1 + q_2}{2} \right) z \quad (32)$$

where $\bar{H}(z)$ = enthalpy of average channel

Assuming constant specific heats, dividing equations (31) and (32) and subtracting unity from each side one gets,

$$\frac{(T_1 - \bar{T}(z))}{(\bar{T}(z) - T_{IN})} = \frac{1}{2\gamma z} (\hat{Q}_p - 1) (1 - e^{-2\gamma z}) \quad (33)$$

At the exit of the bundle, i.e. at $z = L$

$$\bar{T}(z) \equiv T_M.$$

Substituting equation (33) for $z = L$ into equation (24) we have another form of Gr_M which we denote by Gr^* .

$$Gr^* = \frac{Gr}{M f Re^2} \cdot \frac{1}{2\gamma L} (\hat{Q}_p - 1) (1 - e^{-2\gamma L}) \quad (34)$$

Note that by putting $\gamma = 0$ (no mixing) and using L'Hospitals rule one finds

$$Gr^* = Gr^N = \frac{Gr}{M f Re^2} (\hat{Q}_p - 1) \text{ which is equation (25).}$$

The distance, η , between the hot and ~~channel~~^{COLD} channel is arbitrarily chosen to be $D/2$, where D = duct flat to flat distance. It is also assumed that the cross-flow area per unit length $A_c = d(p/d - 1)$ where d = rod diameter and p/d = pitch to diameter ratio.

Then

$$\gamma = \frac{d(p/d - 1)}{A} \cdot \frac{de}{\eta} (\epsilon^* + \alpha^*)$$

defining

$$de = \frac{4A}{dw}, \quad dw = \text{wetted perimeter of each subchannel} = \pi d/2 \quad (\text{assumed})$$

$$\gamma = \frac{d(p/d - 1)}{A} \frac{4A}{\frac{\pi d}{2} \frac{D}{2}} (\epsilon^* + \alpha^*)$$

whence

$$\gamma = \frac{(p/d - 1)}{\pi} \frac{16}{D} (\epsilon^* + \alpha^*) \quad (35)$$

It is necessary to note that the derivation of equations (34) and (35) involves many assumptions that can only be justified if one can find a critical value of Gr^*_c which can predict inception of mixed convection for a very wide range of geometries (h/d , p/d), power skew, total power and bundle lengths.

For a wide range of LMFBR bundles from blanket to fuel assemblies, it can be shown that $1 \gg e^{-2\gamma L}$. We are thus left with a relatively simple form of equation (34).

$$Gr^* = \frac{Gr}{M f Re^2} \frac{(\hat{Q}_p - 1)}{2} \frac{1}{\gamma L} \quad (36)$$

or

$$Gr^* = \frac{Gr}{f Re^2} \chi \quad (37)$$

where

$$\chi = \frac{1}{M} \frac{\hat{Q}_p - 1}{2} \frac{1}{\gamma L} \quad (38)$$

In order to determine the critical values of Gr^N and Gr^* , which can predict inception of mixed convection the following procedure was used.

Ideally one would determine a critical Gr^N or Gr^* by performing experiments for a wide range of parameters Gr , χ and bundle geometries and determine the values of Gr^N and Gr^* at the point of inception of mixed convection for each set of parameters. If this critical value of Gr^N and/or Gr^* are independent of the parameters, the criterion so developed for predicting inception of buoyancy is valid.

Only one set of experimental data for fuel bundles in mixed convection is available. No such data for blanket assemblies is available. The ENERGY II and III codes predict the mixed convection data of Ref. 21 well (using the same values of ϵ^* and C determined from forced convection data). In addition the ENERGY II, III predictions for the blanket assembly of an LMFBR (Section 5.2) matches well with the predictions of codes like THI-3D³⁶. Thus some measure of confidence exists in the predictions of

temperature distributions by ENERGY II and III codes in the mixed convection regime.

Due to lack of mixed convection data it was decided to make computer experiments on several bundle geometries to determine critical values of Gr^N and Gr^* for inception of mixed convection regime.

For a given assembly (i.e. known Gr , χ etc.) ENERGY I computer program was run for a wide range of Reynolds numbers. For the same Reynolds numbers ENERGY II program was run and the temperature distributions compared with the corresponding ENERGY I runs. For lower Reynolds numbers it was assumed that mixed convection effects become important at that Reynolds number, at which the ENERGY I and ENERGY II predictions differ by $k_1\%$ of the average axial bundle temperature rise ($T_M - T_{IN}$). Two values of k_1 were chosen; $k_1 = 3\%$ and $k_1 = 5\%$. Corresponding to these two values of k_1 one obtains two Reynolds numbers and two critical values of Gr^N and Gr^* , denoted by Gr_C^N and Gr_C^* .

The aforementioned procedure was repeated for three different bundles with different values of χ (Table 6). For each bundle the computer experiments yield a critical range of values of Gr_C^N and Gr_C^* , based on the 3% and 5% criteria.

Fig. 40 shows the variation of χ , for the three different bundles, with Reynolds numbers. It is the parameter χ which is responsible for taking into account the effect of mixing and geometry in the definition of Gr^* . Figs. 41 and 42 show the variation of Gr_C^N and Gr_C^* with χ for the three bundles.

The interesting result is that Gr_C^* remains almost constant for all values of χ but Gr_C^N varies significantly. The fact that Gr_C^* is almost uniform with χ is encouraging and confirms the validity of our assumptions in deriving Gr_C^* .

In order to determine, for any given bundle with known operating conditions, when buoyancy effects become important one could use $Gr_C^* = 0.025$ from Fig. 42 and using equation 37 develop a plot of Re vs Gr , which is the lower curve in Fig. 43. Since Gr is known one can determine the Re below which buoyancy effects are important. If the no-mixing critical Grashof number Gr_C^N number was used instead of Gr_C^* one finds, that a conservative estimate of the Reynolds number is obtained for which buoyancy effects become important. When buoyancy effects become important the designer should use either the ENERGY II or ENERGY III code.

The values of $k_1 = 3\%$ and 5% were chosen arbitrarily. These limits must be left to the discretion of the designer who must determine these based on desired accuracy.

Although Table 6 only shows results for a few bundles, the large range of geometrical parameters used gives some confidence as to the validity of the criterion Gr_C^* . However more computer runs over a wide range of Gr , χ and $\hat{Q}_p$ are required before the criterion can be fully confirmed. In addition experiments on various bundle geometries and various operating conditions are required to confirm the validity of the new criterion.

CHAPTER 5

5.1 Application of ENERGY-I Calculations to a 217-pin FFTF Bundle

The calibration of the ENERGY-I, II, III codes is described in Chapter 3. The values of the two empirical constants ϵ^* and C were determined from several sets of data on geometry similar to the FFTF dimensions. The value of ϵ^* obtained was in a range 0.01 (HEDL) to 0.02 (O.R.N.L.) and C was in a range from 0.05 (HEDL) to 0.085 (O.R.N.L., MIT). Moreover there is a considerable uncertainty in flow split which must be taken into account. In order to see the effect of this wide range in the coefficients, on our ability to predict the coolant temperature distributions in a prototype FFTF 217-pin bundle a parametric study was run using the following base parameters.

1. The mean temperature rise across the bundle of 300°F.
2. Axial power skew (peak/avg.) of 1.23 and
3. Radial power (linear) skew (max/min) of 1.50.
4. Inlet temperature of the coolant of 600°F.

The maximum duct wall temperature difference at the exit of the core was investigated for various size bundles. The base case conditions were maintained for bundles of all sizes. Fig. 44 shows that whereas mixing with nominal swirl (finite ϵ^* and C) reduces $(\Delta T_{\max})_{\text{duct}}$ for small bundles (as compared to no mixing and no swirl) it increases it for bundles containing more than 91 pins. Thus while an increased value of ϵ^* is conducive to reducing the maximum $(\Delta T_{\max})_{\text{ax}}$ temperature within the bundle, it increases the duct wall temperature difference. The swirl flow (finite C) on the other hand has been found to reduce the $(\Delta T_{\max})_{\text{duct}}$ without any appreciable effect on $(\Delta T_{\max})_{\text{axial}}$ since $(\Delta T_{\max})_{\text{axial}}$ occurs inside region I which is unaffected by C.

Fig. 45 shows the maximum dimensionless axial temperature rise in the $[\frac{(\Delta T_{\max})_{\text{ax}}}{\Delta T}]$ bundle and the maximum dimensionless duct wall temperature difference $[\frac{(\Delta T_{\max})_{\text{duct}}}{\Delta T}]$ at the core exit as a function of ϵ^* and C. Fig. 45 was plotted for the theoretical flow split case (see Fig. 2). By superimposing the ϵ^* range from 0.01 to 0.02 on figure 45 one obtains the following range of predictions:

$$1.286 \leq \left(\frac{\Delta T_{\max}}{\Delta T} \right)_{\text{axial}} \leq 1.307 \quad (39)$$

$$0.273 \leq \left(\frac{\Delta T_{\max}}{\Delta T} \right)_{\text{duct}} \leq 0.301$$

A range similar to equation 39 above can be obtained for the case where uniform velocity is maintained within the bundle (Figure 46). The following range of predictions results:

$$1.250 \leq \left(\frac{\Delta T_{\max}}{\Delta T} \right)_{\text{axial}} \leq 1.27 \quad (40)$$

$$0.291 \leq \left(\frac{\Delta T_{\max}}{\Delta T} \right)_{\text{duct}} \leq 0.318$$

Based on the limits specified by equations 39 and 40 the following limit on predictions of the maximum temperature within the bundle, $T_{\max}$, and $(\Delta T_{\max})_{\text{duct}}$ is formed.

$$977^{\circ}\text{F} \leq T_{\max} \leq 993^{\circ}\text{F} \quad (41)$$

$$82^{\circ} \leq (\Delta T_{\max})_{\text{duct core exit}} \leq 95^{\circ}\text{F}$$

The limits on $T_{\max}$ are clearly shown in Fig. 47 where temperatures along a cross-section A-B of the bundle exit (exit of core) are shown. As ϵ^* is

decreased, the maximum exit temperature increases and occurs at a position closer to the duct wall.

A large range of ϵ^* and C do not significantly effect $T_{\max}$ and $(\Delta T_{\max})_{\text{duct}}$. $T_{\max}$ lies between 977°F and 993°F and $(\Delta T_{\max})_{\text{duct}}$ at the core exit lies between 82°F and 95°F. It is interesting to note that most of the predictions by other codes lie very close to this range (Table 7 and Fig. 48).

It is necessary to indicate the above range on ΔT_{axial} ($\Delta T_{\text{axial}} = 285^\circ\text{F}$) may be deceptively small especially since hot channel factors have not been included in the calculations. With the inclusion of H.C.F.'s the temperature range within which we can predict $T_{\max}$ with great degree of confidence is expected to increase.

Fig. 49 shows the variation of $(\Delta T_{\max})_{\text{duct}}$ with the axial distance for two values of C . The axial distance is measured from the inlet of the core (active-fuel) and extends to the exit of the fuel assembly. It is seen that the effect of lowering C from 0.085 to 0.05 is to raise the spatial average value of $(\Delta T_{\max})_{\text{duct}}$ from 36 in. to 84 in. by approximately 10°F. Duct structural designers must specify if this difference in $(\Delta T_{\max})_{\text{duct}}$ is significant.

The peak temperature in the duct wall at the 84 in elevation occurs at the midpoint of face C (see Fig. 47) for $C = 0.085$. However if $C = 0.05$ the peak temperature occurs on face A very close to the corner between faces A and C. The rotation of the peak temperature in the duct wall appears to be relatively insensitive to C . This should be useful for core restraint design purposes.

5.2 Application of ENERGY I, II, III Calculations to a 61-pin Blanket Assembly in Forced Convection and in Mixed Convection

A blanket assembly in an LMFBR could operate in either forced convection (negligible natural convection) or in mixed convection (combined forced and free) during its lifetime in the reactor. In this section first calculations are presented for a typical blanket assembly of an LMFBR in forced convection using ENERGY-I. Next, by reducing Reynolds number to a very low value (but keeping the same power to flow ratio so as to maintain the same mean temperature rise across the bundle) the assembly is made to operate in the combined forced/free convection regime ($Gr^* > Gr_c^*$). ENERGY II and ENERGY III calculations are presented for the

blanket assembly in mixed convection.

For a typical blanket assembly operating in forced convection the characteristic geometric and operating conditions are shown in Table 8 from Ref. 36. The p/d and h/d ratios are 1.076 and 7.7 respectively. Moreover the wire wrap in the wall channels are almost half the diameter of those in the central channels. The axial peak to average power skew is 2.06. The radial power skew³⁶ is 3:1 across flats and 6:1 across corners.

One of the difficulties in making ENERGY calculations for the blanket geometry is that both ϵ^* and C must be determined by extrapolation from fuel geometries. Calibration of ENERGY for blanket geometries has not been made because no data exists. For the purpose of this analysis ϵ^* was determined to be 0.019 by linearly extrapolating values obtained for fuel geometries as shown on Figure 20. The value of C was obtained by adjustment of the value obtained by a similar linear extrapolation of Figure 21. The adjustment was based on the assumption that the swirl mass flow in a blanket assembly is the same for half or full size edge pin wire thickness since this flow is predominantly generated from the bundle interior. Since the width of Region II, $d/2 + dw$, is

fairly insensitive to d_w due to the large pin diameter, d , this adjustment was relatively small and the final C value determined was 0.3. It is possible that our assumption on swirl mass flow for a bundle with half size edge pins yields an overprediction.

Fig. 50 shows a comparison of THI-3D³⁵ and ENERGY-I predictions at the exit of the blanket assembly. Except for the wall regions, the temperature distributions matches well.

Fig. 51 and 52 show a comparison of ENERGY II and ENERGY III temperature distribution calculations for the same blanket assembly, operating in the mixed convection regime. The Reynolds number was reduced to 850 while the same power to flow ratio was maintained. The node size used in the ENERGY II calculations was much smaller (126 nodes per axial level) than the node size used in the ENERGY III calculations (42 nodes per axial level). Yet the temperature distributions for the central and wall regions are found to be close. Results of Chapter 2 for fuel assembly geometry in mixed convection showed that computational savings could be made achieved without significant loss of accuracy by dropping convective terms while maintaining node size (ENERGY III $\rightarrow$ ENERGY II). These results show

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additionally that tradeoffs between inclusion of convective terms and changing node size (within a factor of 3) can be made without change in accuracy (i.e. ENERGY III with large nodes versus ENERGY II with small nodes). Since ENERGY III is inherently the more accurate code, these results are encouraging since they indicate that the characteristic of the porous body model allowing increased node size can be exploited to yield reduced computational time.

Chapter 6

6.1 Proposed Experiments

Based on the data analysis (Chapter 3), discussions (Appendix 2) and application of the code ENERGY to a 217 pin prototype bundle at full power (Chapter 5) we have come to the conclusion that further experimentation is required if reduction of the range 977-993° on coolant T_{max} and 82-95° on $(\Delta T_{max})_{duct}$ is desired for 217-pin assemblies. The range of values of ϵ^* and C that correspond to this uncertainty in T_{max} and $(\Delta T_{max})_{duct}$ are 0.012 to 0.02 for ϵ^* and 0.05 and 0.085 for C. The nominal values of ϵ^* and C which we believe are appropriate are 0.017 and 0.067. The nominal value of ϵ^* is weighted slightly more towards its value determined from the ORNL data (as compared to its value obtained from HEDL data) because of our greater confidence in the ORNL heated pin experiment.

The nominal range of C has been determined by giving equal weight to the values obtained from ORNL and HEDL experiments, unlike what was done for ϵ^* . The reason for this is as follows. The value of ϵ^* determined from HEDL data depends upon the absolute magnitude of the concentrations measured by the

conductivity probes. For reasons discussed in Ref. 17 it is seen that ϵ^* determined would have large error bars associated with it. However C can be determined, to a good degree of approximation, by the distance traveled by salt along the circumference of the hexagonal duct. Measurement of the movement of the salt front is obtained by noting the response of the probe to concentrations above background and hence determination of the front location is not sensitive to absolute magnitude of probe reading.

Since each degree F reduction in $T_{\max}$ could mean an increase in max. burnup by up to 500 Mwd/T¹⁷ it is imperative that a better understanding of the flow field and bundle size effects be obtained from more carefully designed experiments. Two different experiments will be briefly described. The first experiment is similar to the heated pin experiments described in Ref. 2. The second experiment has not been used before for LMFBR mixing studies and is recommended. It may even be possible to combine the two types of experiment in an optimum fashion into a single setup. Before describing the experiment design let us again briefly examine the need for these experiments.

- (1) Each °F reduction in $T_{\max}$ could mean up to 500 Mwd/T increase in burnup. Decrease in $(\Delta T_{\max})_{\text{duct}}$ also means an increase in burnup.
- (2) Tables 2 and 3 show that certain data gaps exist before a thermal-hydraulic optimization study on LMFBR fuel assembly design can be made. Very little reliable data exists for swirl flow and ϵ^* at low values of h/d ($h/d < 48$). The only complete set of data available below $h/d < 24$ is SKOK²² hot water injection data on 7-pin bundles. More data is required on larger bundles for low h/d and p/d . Sufficient data also does not exist to determine if rod diameter as a variable itself is of importance.
- (3) Although several sets of experiments have been performed on FFTF bundle geometry, there still remains a great deal of speculation with regard to the flow field in bundles of various sizes. For example, is the bundle size effect observed real or is it due to faulty

instrumentation and errors in measurement? Is the flow split a function of bundle size? A knowledge of the correct flow split could significantly reduce errors in predicting temperature distributions. If internal swirl flow exists for smaller bundles and reduces toward zero as the bundle size increased to a 217-pin bundle, then the internal swirl flow must affect the flow field in these smaller bundles. Therefore experimental data observed on bundles smaller than the full size bundle could lead to erroneous results.

None of the experiments utilizing heated pins measure fluid temperatures in the axial direction. Point measurements of temperatures by the wire and wall thermocouples cannot be easily used by existing sub-channel analysis codes. The experiments proposed will permit measurement of axial fluid temperature distribution with as much accuracy as the exit fluid temperature distribution by varying the heated length of the pin but keeping the same exit position where the exit rake thermocouples measure the fluid temperature.

6.2 Experiment I

A 217-pin experiment is proposed with one or two heated pins strategically located in the manner suggested in Ref. 2. A 217-pin bundle, and location of the thermocouples in the subchannels is shown in Figure 53. These thermocouples are placed in an exit rake similar in design to the ORNL exit rake. Water is used as the coolant. The use of water as coolant is justified for fuel assemblies where buoyancy effects are unimportant at high Reynolds number and also (since $\epsilon \gg \alpha$) thermal conduction is unimportant both for sodium and water. Initially only rod 1 is heated. Fig. 54 shows the temperature rise as a function of the radial distance for three different Reynolds numbers. The rod power for the experiment in Fig. 54 is 10Kw/ft. Since ϵ^* is not expected to be a function of Reynolds number, the same value of ϵ^* ($\epsilon^* = 0.017$) was used to obtain curves I, II and III. Fig. 55 shows the temperature distribution vs radial distance when the rod power is changed to 5 and 15 kw/ft but Reynolds number is maintained at 10000. One would ideally like to use a rod with the maximum power rating, but rod bowing at high linear power rating (i.e. 8 kw/ft) has been observed²¹ in a 19 pin

experiment. Here the power to flow ratio is much smaller than in Ref. 21 and it is expected that bowing and related problems should diminish considerably in severity. In order to cover a wide Reynolds number range ($5 \times 10^3 - 15 \times 10^3$) and yet produce accurate data with the lowest power rating of the rod, it is recommended that the 10 kw/ft rod be used for the experiments. As the h/d is reduced the mixing rate ϵ^* increases and the maximum temperatures in Fig. 54 would reduce. The thermocouples can be designed to measure temperatures to within $\pm 1/2^\circ\text{F}$. Since only one thermocouple (located at the subchannel centroid) is used in each subchannel in these experiments, it is expected that the temperature measured by it may be slightly different from the subchannel average temperature. However, the temperature gradients are small in the transverse direction and the difference in the thermocouple reading and the subchannel average temperature is expected to be small. In order to take such errors into account for determining a realistic estimate of the reduction in the range of ϵ^* and C from the proposed experiment, a large error bar of $\pm 1^\circ\text{F}$ is applied to the thermocouple precision. Fig. 56 shows the calculated variation of the temperature rise recorded by thermocouple L vs ϵ^* .

On superimposing a $\pm 1^\circ\text{F}$ error bar on these curves about the temperature corresponding to nominal ϵ^* of 0.017 one finds the precision with which ϵ^* can be determined by this experiment. As the power to flow ratio decreases (in going from curve I to III in Fig. 56) the precision with which ϵ^* can be determined decreases. For this reason curve III in Fig. 56 shows a much larger ϵ^* range than curves I and II. Therefore for the 10 kw/ft experiment only curves I and II (at Reynolds number of 5000 and 10000) should be used to determine ϵ^* . A third set of data, if desired, could be obtained at a Reynolds number of 7500. The range within which experiment I can determine ϵ^* (assuming a nominal value of ϵ^* of 0.017) is 0.016 to .018.

Fig. 57 shows the radial temperature distribution for two rods heated, one at the center and one at the wall (Fig. 53). Both rods have the same power. It is seen that the swirl flow affects the temperature gradients significantly only near the wall. The wall thermocouples F_1 and F_2 show a large effect of variation of C . This is also seen in Figs. 58 and 59 where the circumferential temperature profile is plotted.

As the swirl velocity ratio C is varied, calculations show that the temperature profile in the circumferential direction changes significantly.

Fig. 60 shows the calculated sensitivity of the temperature of thermocouple F_2 to variations in C . It is obvious that C can be determined to a high degree of accuracy. An independent set of measurements using the Laser Doppler Velocimeter (LDV) can be used to verify the value of C so obtained. Fig. 60 shows that if the nominal value of C is 0.067 the range within which C can be determined from experiments is 0.0645 to 0.071. Assuming the Laser Doppler Anemometer can confirm this range of C and can also be used to determine the flow split accurately one can show using Figs. 45 or Fig. 46 that the maximum core exit temperature, T_{max} , can be predicted to within $\pm 2^\circ\text{F}$ and the $(\Delta T_{max})_{duct}$ can be predicted to within $\pm 2.5^\circ\text{F}$. Even if these limits were arbitrarily doubled, the precision with which the temperature distribution can be determined from the proposed experiments would be far better than the present accuracy.

The range of values of ϵ^* and C determined from Experiment I (for the 217 pin bundle) can be compared with values obtained for these parameters by analysis of data in smaller bundles of similar geometry. Two such bundles are the ORNL²¹ 19-pin bundle and the ANL^{14,15} 91-pin bundle. If the range of ϵ^* and C does not change in going from a larger

to a smaller bundle, a bundle size effect does not exist. In this case, data obtained on smaller than full size fuel and blanket assemblies can be used to obtain ϵ^* and C. These can be used with confidence to predict the full size prototype bundle temperature distributions.

However, if a significant variation in ϵ^* and C is found between the 217-pin bundle and the smaller bundles, then it is suggested that Experiment I be repeated on a smaller bundle of the same geometric characteristic to determine if the difference in ϵ^* and C is due to bundle size effect or if it is due to an experimental error by previous investigators.

If the bundle size effect, in fact exists, then data on smaller bundles should not be used for prototype predictions.

6.3 Experiment II

This experiment is proposed as a supplement to Experiment I to determine ϵ^* independent of an assumed value of flow split. Specifically while calibrating computer programs like COTEC⁴ and ENERGY-I a certain velocity profile is assumed in the bundle. Codes like THI-3D⁹ calculate this

velocity profile. While some small amount of data is available on flow splits, there is not a single set of reliable data that can give the cross-sectional distribution of subchannel average velocities in the bundle. If the codes⁴ assume a 3-dimensional velocity profile or if THI-3D⁹ calculates a velocity profile different from the 'correct' velocity profile, there will be errors made in calibration of mixing coefficients. Moreover the velocity profile may vary axially due to changes in coolant properties with temperature. Thus substituting an assumed velocity field into the energy equation and then determining a ϵ^* (or a coefficient for sweep flow as was done for THI-3D when using ORNL data (Ref. 36)) by matching temperature fields is not strictly correct.

Figs. 61 and 62 show the cross-section of the experimental setup. A single heated rod is located at the center of the bundle which is surrounded by a 'jacket'. The fluid flowing within the bundle is not used as a heat sink. Instead the fluid in the jacket is used as a coolant. Since the fluid within the bundle is not a heat sink, a steady temperature profile distribution will be established after a developing length which would be as long as

8 - 9 ft. It would be possible to reduce considerably the axial distance required to achieve steady temperature profiles provided preheated fluid with a similar temperature distribution as the steady state distribution could be supplied at the inlet of the bundle. Obviously one does not know the steady temperature profile 'a priori'. So to start the experiment, preheat the inlet coolant to give a radial inlet temperature distribution (in practice several concentric zones) which is similar to a calculated steady temperature profile. Then by a trial and error procedure it should be fairly straightforward to preheat the inlet fluid in a manner such that steady temperature profile is established within 12 to 18 in. The details, such as flow rates within the bundle and the coolant flow rate outside the bundle, can be calculated easily and are not given. The thermocouple locations are also not shown in the figure.

Since the fluid within the bundles is not a heat sink the velocity and temperature profiles remain constant axially. For such a system the energy equation becomes

$$\frac{\partial}{\partial r} [r(\rho C_p \epsilon_H(r) + k)] \frac{\partial T}{\partial r} = 0$$

At high Re for water (also for sodium) as coolant in the presence of wire wrap $\rho C_p \epsilon_H \gg k$, then

$$\frac{\partial}{\partial r} [r \rho C_p \epsilon_H \frac{\partial T}{\partial r}] = 0$$

If the total heat transferred radially across a vertical surface at position r in a length L is Q (Btu/hr) then using this as a boundary condition

$$\epsilon_H(r) = - \frac{Q}{2\pi r L (\rho C_p) \left(\frac{\partial T}{\partial r} \right)_{r=r}}$$

whence

$$\epsilon_H^*(r) = - \frac{Q / (\bar{V} d e)}{2\pi r (\rho C_p L) \left(\frac{\partial T}{\partial r} \right)_{r=r}}$$

$\bar{V}$ = bundle average velocity

This type of experiment was originally designed by Yagi and Kunii (Ref. 37) for studying heat transfer near wall surface in packed beds. If Q is known then by measuring the temperature gradient $\left(\frac{\partial T}{\partial r} \right)_{r=r}$ at the radial position ' r ' one can calculate the radial distribution of $\epsilon^*(r)$. Thus, if the mixing coefficient near the duct wall decreased, it should be reflected by a change of slope of the T vs r curve. With such a system, therefore, the velocity

data need not enter the calculation of ϵ^* and consequently the possible error in determining ϵ^* is reduced.

It is necessary to note that a one-dimensional energy transport is assumed in the formulation of equation. Consideration must be given to the fact that a swirl velocity exists near the wall.

However, for the centrally heated rod it is difficult to imagine the existence of a circumferential temperature gradient. The Karlsruhe experiments² do not report a temperature gradient in the circumferential direction for the case when the central pin was heated. These tests were carried out very carefully, at low power rating of the centrally heated pin. The ORNL 19-pin experiments report a circumferential temperature gradient for the centrally heated pin. It is believed that this is due to rod bowing since the pin was operated at 10 kw/ft. and bowing was observed at power ratings above 8 kw/ft. Even if a very small fraction of the total energy is transported circumferentially (which we believe will not occur after 12-24 in.) the results should not be affected to any large extent.

The sequence in which the two experiments should be conducted is as follows:

- (1) Perform the single centrally heated pin portion of Experiment I and determine if circumferential temperature gradients exist.
- (2) Perform the two heated pin portion of Experiment I - get swirl flow data.
(For this case uniform ϵ^* is assumed throughout the bundle.)
- (3) For the pin-array of Experiment I - get swirl flow data from Laser Doppler Velocimeter.
- (4) If in Step (1) it is found that for the centrally heated pin circumferential gradients exist, Experiment II cannot yield $\epsilon^*(r)$. If however circumferential gradients are found to be very small or non-existent then conduct Experiment II to determine $\epsilon^*(r)$. The bundle array of Experiment I could be modified by adding a circumferential cooling jacket and a new inlet region.

- (5) It is expected that ϵ^* is not a function of radius. However, if it is found to be a function of the radial position in Step (4), Step (2) must be repeated and in calibration of C the swirl velocity ratio, the correct local ϵ^* should be used.

7.1 Limitations and Extensions of Present Work

The present work is mainly concerned with the development of a new model for predicting temperature distributions in wire wrapped LMFBR assemblies. The model is similar to the porous body model that has been previously used at low Re for heat and mass transfer calculations in packed beds. The model has been successful in predicting nuclear reactor assembly temperature distributions for a large range of flow and thermal parameters.

The method requires a knowledge of two empirical constants ϵ^* and C. These have been determined by calibration with existing data in fuel assemblies over a wide range of geometrical parameters. However, our confidence in predicting temperatures in assemblies whose geometric characteristics are outside the range of data analysed is not good. This is a shortcoming of the method developed. The shortcoming of the method is compensated by its small computational times, which is of great asset to the designer. While the present method was developed for wire wrapped LMFBR bundles it has broader applicability to

- (a) any single phase flow through a rod array with grid spacers, i.e. PWR sub assembly in single

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phase flow (ENERGY I) and

(b) in two phase flow (ENERGY II) through rod arrays.

Of course an axial distribution of ϵ^* must be introduced and empirically determined from data for these cases.

Moreover this method could be applied with proper calibration to reactor situations where fuel is in a packed bed array either by design (pebble bed reactor) or as a result of a hypothetical accident.

Confirmation of the results obtained from ENERGY II and ENERGY III codes is required when data on mixed convection in LMFBR bundles is available. These codes have been found to predict the temperature distributions in the ORNL FFMIIA bundle well, in the mixed convection regime.

The criterion developed for predicting inception of forced/free convection regime must be experimentally checked for a wide range of parameters.

As mentioned in Chapter I one of the prime concerns in LMFBR fuel assembly design is the duct bowing due to thermal expansion and its enhancement due to swelling in a fast spectrum. Swelling in turn, is very sensitive to the duct temperature distribution. Blanket assemblies are subjected to very severe radial flux skews as a result of which considerable amount of interassembly heat transfer occurs. This interassembly heat transfer could significantly effect the duct wall temperature distribution.

Therefore many blanket assemblies must be thermally coupled to each other (possibly at least seven adjacent assemblies should be analysed at a time) to determine their temperature distribution.

The present subchannel analysis methods would require excessive time and computer storage to solve for the temperature distributions in 7 adjacent assemblies simultaneously. However the porous body model in ENERGY, permits using larger nodes without a significant loss in accuracy (Table 4 and Figs. 51, 52). Thus small storage combined with a low computational time would make the ENERGY codes ideal starting points for developing methods for multiassembly calculations. For the same reasons it is suggested that ENERGY I, II and III be modified to predict temperature distributions during a flow or power transient situation.

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Table 1

<u>Name</u>	<u>Author</u>	<u>Organization</u>	<u>Category</u>	<u>Type</u>
ENERGY	Khan	M. I. T.	1	2-Regions
MISTRAL	Baumann	Karlshruhe	2	2-Regions
CΦBRA IIA	Wheeler et al	Battelle	3	
CΦTEC	Novendstern	WARD	4	Marching Codes Include Directional Sweep Flow Correlation Axial Flow Specified
FΦRCMX	Graves	A. I.		
FULMX	Magee	G. E.		
ΦRRIBLE	Wantland	O. R. N. L.		
CΦBRA III	Rowe	Battelle	5	Boundary Value Problem Solved, Boundary Condi- tions Must Be Known Input Flow Districution Obtained From Momentum Equations
THI-3D	Sha	ANL		

TABLE 2

REF. \ NO. OF PINS	7	19	37	61	91	127	217
ANL - RAS ANL - CT ANL - CT	①(b)				⑥(a) ⑨(b)		
ORNL		④(c)					
MIT				⑤(d)			
BATTELLE	②(d)						
JAPANESE		⑫(c)		⑥(b) ⑫(c)			
HEDL							⑩(b)
GE						③(a)	
KARLSRUHE				⑦(c)			
CADARACHE	③(a)						

LEGEND ① 1,2,---11 NUMBER ASSIGNED TO EACH EXPERIMENT FOR USE IN TABLE 3

② TYPE OF EXPERIMENT (a) HOT WATER INJECTION (b) SALT INJECTION

(c) HEATED PIN (d) LASER

TABLE 3 DATA ANALYSED

$\begin{array}{c} h/d \\ p/d \end{array}$	14	17	19	21	24	26.5	28	34	43	48	50	52	55
1.065			3*										
1.09				12									
1.1													
1.14	3			3			3						
1.20									6 12	1 8			
1.24										9		4 11	
1.25					5					5			
1.28					10								2
1.315		7						7			7		
1.355						3							

* NUMBERS REFER TO TABLE - 2

Table 4

CODE	BUNDLE SIZE		
	217 (36" Long)	61 Pins (64" Long)	
Nodes Per Level	438	42 Nodes	126 Nodes
ENERGY I	20 secs.	x	< 4 sec.
ENERGY II	x	< 1 min.	4 min.
ENERGY III	x	8 min.	> 2 hrs.

Table 5

ORNL RUN NO.	REYNOLDS NO. (Based on hydraulic diameter of interior channels) .	POWER SKEW (Direction of skew in Fig. 7)
1432	1000	Flat/Power
1416	1100	3.4:1 (B)
1524	1150	2.8:1 (E)
1101	2480	3:5:1 (B)
1324	2580	Flat
1056	3620	3:1 (E)
959	3740	3:1 (B)
1602	4550	Flat

Table 6

Bundle No.	Bundle Geometry					Power Skew	Heated Length	M	$\frac{2 \gamma L}{(R_c^{.5000})}$	$G_{r-5} \times 10^{-5}$	$X=f(R_{ec})$ (Fig. 40)		R_{ec}		G_{rc}^n		G_{rc}^*	
	p/d	h/d	d in.	D in.	No. of Pins						ϕ_p	in.				k=3%	5%	3%
1	1.25	52	0.25	1.45	19	1.7	21	1.04	4.58	1.7910	.186	.147	4400	3000	0.2	0.3	.047	.072
2	1.25	52	0.25	1.45	19	1.7	58	1.04	12.2	1.8758	.032	.019	1400	800	2.2	3.4	.067	.073
3	1.076	7.7	0.52	4.45	61	1.7	64	1.88	1.54	1.7300	.275	.225	5000	3200	0.068	0.15	.051	.091
1	1.25	52	0.25	1.45	19	1.2	21	1.04	4.58	1.791	.044	.031	2800	1750	.14	.21	.023	.038

Table 7 14

Max. Coolant Temperatures	
SIMPLE 28	1000 °F
THI-3D 10	992 °F
CφTEC11	988 °F
ENERGY	985 °F $\pm$ 8 °F
FφRCMX 20	972 °F

Table 8³⁶

Geometric and Operating Characteristics of
a 61-Pin Blanket Assembly

Description	
Cladding o. d. (in.)	0.520
Cladding i. d. (in.)	0.490
Pellet o. d. (in.)	0.485
Pellet density (% of TD)	95.6
Wire-wrap o. d. (in.)	0.037 ^a
Wire-wrap pitch (in.)	4.0
Rod pitch (in.)	0.560
Across hex flat (in.)	4.450
Hex corner radius (in.)	0.16
Flow rate (lb/h)	29,412
Total power [kW(th)]	700.3
Heated length (cm)	162.56
Average heat flux [Btu/(h ft ²)]	53,963
Inlet temperature (°F)	724
Outlet temperature (°F)	992
Rod-bundle average velocity (ft/sec)	5.30
Rod-bundle inlet pressure (psia)	29.4

^aCorner rods have 0.020-in. wires; edge rods have 0.020-in. wires over 120-deg facing duct.

Table 9

	FUEL				BLANKET			
	Forced Convection		Mixed Convection		Forced Convection		Mixed Convection	
	Data Available	Data Required	Data Available	Data Required	Data Available	Data Required	Data Available	Data Required
No. of Pins	7-217	217	19	61-217	7-19	61	x	19-61
p/d	1. 14-1. 32	1. 2-1. 3	1. 25	1. 2-1. 3	1. 065-1. 09	1. 06-1. 14	x	1. 06-1. 14
h/d	14-52	20-52	52	20-52	20	7-14	x	7-14

FIG. 1

FIG. 2

FIG. 3

FIG. 4

FIG.5

NUMERICAL SOLUTION OF $\nabla^2 T = 0$ TO DETERMINE κ
2 DIMENSIONAL

FIG. 6

x ≡ LOCATION OF EXIT THERMOCOUPLE

TEST PARAMETERS: REF. 21

1. FLUID - LIQUID Na
2. FLOW RATE - UP TO 56 GPM ($Re = 1,000$ TO $70,000$)
3. POWER SKEW
 - a. ALL RODS UNIFORMLY HEATED
 - b. EACH ROD HEATED INDIVIDUALLY
 - c. B 20 % HIGHER THAN E
 - d. E 20 % HIGHER THAN B
 - e. B 300% HIGHER THAN E
 - f. E 300% HIGHER THAN B
4. $d = 0.23$ IN.
 $h/d = 52$
 $p/d = 1.24$

FIG. 7

FIG. 8

FIG. 9

FIG. 10

FIG. 11

FIG. 12

FIG. 13

FIG. 14

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FIG. 61

FIG. 62

FIG. 63

APPENDIX 1Definition of Mixing Mechanisms as used in literature

Energy redistribution in a wire-wrapped fuel assembly takes place by the following mechanisms:

- (a) Thermal Conduction - characterized by the thermal diffusivity, α .
- (b) Turbulent Exchange on a Molecular Level (including flow scattering) without a net transfer of fluid. The dimensionless group characterizing turbulent exchange is, $W_{Tij}/\bar{W}$, where W_{Tij} is the turbulent exchange rate and $\bar{W}$ is the average bundle flow rate.
- (c) Cross-Flow - any convection of fluid due to a radial pressure gradient can be classified as cross-flow. Cross-flow can be subdivided into two categories: (1) Diversion Cross-Flow: That fraction of the total cross-flow between any two sub-channels that occurs due to a pressure gradient set up by virtue of the dissimilarities in local hydraulic characteristics (either geometrically, hydrodynamically or thermally induced). For example, as the wire lead to a diameter increases to very large

values ($\frac{h}{d} \rightarrow \infty$) the hydraulic diameter of channels i and j would be locally different at various axial levels causing flow to redistribute itself by diversion cross-flow. Thus the varying axial and transverse flow resistance in the presence or absence of wire wrap can cause diversion cross-flow.

Diversion cross-flow may be characterized by the term, $W_{Dij}/\sqrt{W}$

Flow-Sweep: For wire-wrapped rod bundles of $p/d > 1.18$ the axial flow has no direct vertical path through the bundle but at regular intervals crosses the wire and is swept from one channel to another due to the favorable pressure gradient set up by the wire. This sweeping effect of the wire would extend a small distance below and above the location where the wire crosses the gap. Thus the axial momentum of flow in a subchannel is periodically changed by the presence of the wire. The fraction of axial momentum carried by the sweep flow in the transverse direction can be considerably greater than that carried by diversion cross-flow, depending upon the wire lead to diameter ratio. For rod bundles with $p/d < 1.18$ a part of the fluid can flow vertically upwards in a bundle and part of it will be periodically swept into other channels. The flow field for such a

bundle would be even more complex than for a bundle with $p/d > 1.18$.

Flow sweep may be characterized by the term, $W_{Sij}/\bar{W}$ (or W_{Sij}/W_1).

Swirl Flow: The sweep flow in the wall channels has characteristics which are different from that in the central channels. Whereas the sweep flow between two subchannels in the central parts of the bundle changes direction periodically, the sweep flow between two wall channels is always in the same direction. This sweep flow along the housing wall is known as the swirl flow. The swirl flow, perhaps, fluctuates about a mean value. It is characterized by the term, $C_1 = V_\theta/\bar{V}$, where V_θ is the velocity in the gap between the rod and the wall and $\bar{V}$ average axial bundle velocity.

Most experimental effort has been to quantitatively determine these four mechanisms individually if possible and their dependence upon geometrical and flow parameters. The major emphasis has been in determining flow sweeping, perhaps because it is the least understood and the most important means of momentum and energy transfer. Once these mechanisms are known, they are direct input into the subchannel analysis computer programs as they are now formulated.

APPENDIX 2Comparison of Heated Pin, Hot Water Injection and Salt Injection Experiments

Our analysis of the various types of data showed that the heated pin experiments are probably the most reliable, provided care is taken in designing the temperature sensors and the heated rods. The discussion to follow compares heated pin experiments with non-heated pin experiments (salt and hot water injection included). The Laser Doppler Velocimeter experiments are not discussed since they can at present only be used for wall channels where they can supplement the results of any of the above experiments.

The major problem with the two non-heated experiments (apart from a different boundary condition than the prototype) is that an injector must be introduced into the main flow stream. Local perturbations of the bulk flow can affect the measurements many inches downstream of the injection point. Three factors that are extremely important in obtaining reliable reproducible results with any injection device are the injector design, injector position and injection flow rate^(14, 15). Considerable effort has been devoted by Lorenz¹⁴ and Pederson¹⁵ in testing various injector designs. Their

results show an injector located at the subchannel centroid with injection velocities close to the local mainstream velocity should give reliable results for wall channels. A similar result has been obtained⁵⁰ for an injector located at the centroid of a central subchannel. The HEDL²⁵ inlet perturbation effects were found to last from 6 in. to 9 in. The flow drift of two channels (see Ref. 17) has been interpreted by investigators as an internal swirl flow. ENERGY analysis shows that this drift is an inlet perturbation effect. A similar inlet perturbation effect was found by examining, level by level, the GE-127 pin²⁴ hot water injection data. The inlet perturbation in this case is an extremely small internal swirl flow which dies off after about 12-18 inches.

The state of our understanding of the hydraulics of the wire wrapped bundle is such at present that an injection scheme tested in the wall and first row subchannels for a small bundle cannot be used for a large bundle with a great deal of confidence. Thus until a complete knowledge of the local flow fields within subchannels of a wire wrapped bundle is available, each experimenter should review references 14, 15 and 50 to determine if the objectives of his test justifies

performance of additional injector development work.

The tracer detection by wall thermocouples (for hot water injection) and wall conductivity probes is a localized point measurement not suitable for lumped subchannel analysis computer programs. The isokinetic sampling technique¹⁸, however, along with conductivity probes (or thermocouples) located near or at the center of the subchannel²¹ can yield more useful results. Although wall thermocouples can be designed²⁴ carefully, a steep temperature gradient between wall and the center of the subchannels could give misleading results. The M.I.T. experiments²⁹ use a variable injection position and fixed detection probes at the outlet of the bundle. These probes are located at the center of the subchannels. The tracer detection in non-heated experiments can, therefore, be carefully designed to give reliable results. The mass balance (in salt-injection experiments) and energy balance (in hot water injection experiments) should be as good as the energy balance in heated pin experiments²¹. To date poor mass balances²⁵ and energy balance¹⁵ for central injection have been obtained. Heated pin experiments (Refs. 2, 21) have yielded energy balance within 3-5% as compared to 13-35%²⁵ for non-heated pin experiments.

The isokinetic sampling technique does not appear to suffer from the same major drawbacks as the others but a possible source of error in this technique is discussed below. Using the isokinetic sampling technique the differential static pressure between a subchannel and another reference subchannel at the exit plane to the test section is measured under the undisturbed conditions. Then an extraction device is placed at the measuring plane and flow withdrawn until the previous $\Delta P_{\text{ref.}}$ between the subchannel and the reference subchannel is obtained. It is then assumed that the original flow split is re-established and the "proper flow" flow split is being sampled.⁶ The problem with the technique is that re-establishing the previous $\Delta P_{\text{ref.}}$ between the subchannel and reference subchannel is necessary but not sufficient to ensure that the original flow split is established. This is because the pressure difference between the subchannel in question and subchannels other than the reference may not be the same as before even though $\Delta P_{\text{ref.}}$ has been re-established. Figure 63 shows a comparison of isokinetic sampling data on nearly similar geometries in a 7-pin¹⁸ and in a 91-pin¹⁴ experiment. Although wire orientations are different for the two cases the difference in results cannot be explained by wire-orientation effect alone since wire orientation C lies in between A and B but the corresponding curve for C/C_0 does not.

It could possibly be a bundle size effect. If the latter is ruled out (at present the latter cannot be ruled out) then the reliability of the technique can be questioned. More experiments are required to test fully the reliability of the isokinetic sampling method.

The most useful portion of the data generated by salt and hot water injection experiments is produced in the axial distance 1 to 1-1/4 wire leads downstream of the injection point. Codes like C ϕ TEC⁴ and THI-3D⁹ use the first few inches (for lead of .12 in.) for sweep flow calibration. As discussed in section 3.2 of Chapter 3, as one goes downstream very small errors in tracer concentration yield a large range for ϵ^* since the calculated salt concentration vs. ϵ^* curves become flatter with distance downstream. Moreover for ENERGY purposes the injection channel data was found to give the least errors in calibration (see Fig. 18) for the reasons mentioned in Section 3.2. If inlet perturbation effects continue for a few inches downstream of the injection plane, the most useful portion of the data is lost. It is more difficult to develop data free of inlet perturbation for central injection than for wall injection due to lack of probe maneuverability (bundle must be disassembled every time) and accessibility. In heated pin experiments any rod in the bundle can be easily heated by external control. Whereas concentrations in the injection and neighboring channels continuously

decrease in the tracer injection experiment, the temperatures of the channel near the heated rod or rods in a heated pin experiment (line source) continuously increase. Heated pin experiments can be designed more easily to give accurate reliable axial temperature variations by changing the length of the pin heated and using the same exit (see Chapter 6) thermocouples. The data from Refs. 2, 21 has generally shown a greater degree of accuracy and reliability than for tracer injection experiments.

One of the most important features of tracer injection experiments (that can again be achieved by heated pin experiments in which heat is generated only over a small axial length) was found in Ref. 15. Tracer injection into an inner (No. 12) channel adjacent to the wall channel (see Fig. 33 of Ref. 15) shows that most of the fluid is transferred to an adjacent inner channel 11 by diversion cross-flow (due to the presence of wire in channel 12 its hydraulic diameter decreases) and very little to adjacent channel 3. One would have expected at least an equal amount to be transferred to both channels due to diversion cross-flow plus an additional amount to channel 3 by sweep flow as the wire crosses the gap between rods 2 and 3. It is not correct to conclude that interaction between wall and inner region is smaller than the interaction between any two interior channels based on this evidence. There is some evidence that mixing is uniform in the bundle.² However, a knowledge of these

local hydraulic interactions between subchannels can be very useful in trying to assess the relative importance of sweep and diversion cross-flow in codes that linearly add these two effects. It is believed by the present authors that no code, as yet, can predict this type of a local flow redistribution with any degree of confidence. This is one motivation for development of simpler codes like ENERGY which lump these modes of energy transfer. However as more sophistication is built into the hydraulic models of the more advanced codes like THI-3D and COBRA, this type of data will be most useful.

Only three sets of heated pin experiments are available for the fuel-assembly geometry. These are the ORNL²¹ 19 pin, Japanese²⁷ 19 and 61 pin and the Karlsruhe^{2,26} 61 pin experiments. Although several sets of useful data are available from these experiments a number of important issues still remain unclear. The Karlsruhe^{26,2} heated pin experiments show that the effect of wall swirl flow penetrates at least 3-4 channels radially inwards, almost to the center. This effect is not due to conduction because conduction effects have been found to be an order of magnitude smaller than the wire wrap mixing effects for $Re > 10000$. If internal swirl exists (which we do not believe, based on our analysis of the 217 pin bundle) for the 61 pin bundle its effect cannot be separated out due to penetration of wall effects.

However, if a 217-pin experiment were performed using water with single or two heated pins strategically located it would be possible to confirm our belief, based on analysis of HEDL experiments, that no internal swirl exists. By locating heated pins near the wall the swirl flow (C) can be determined and further confirmed by laser measurements. In addition the existence of bundle size effects can be checked. The need for such an experiment (for a large range of h/d , p/d , d) exists and is described in Chapter 6.

Appendix 3

SOLUTION OF ENERGY EQUATIONS

(a) ENERGY-I

Equations (3) and (4) may be written in a finite difference form. The hexagonal shape of the boundary walls suggests the selection of an equilateral triangular grid, as shown in Fig. 5, in the central region and a rectangular grid in the wall region. Let η_T^* be the dimensionless node size in Region I and let δz^* be the dimensionless step length in the axial direction.

In Region I expanding the dimensionless enthalpy at locations 1, 2, 3 about i and taking a forward difference in the axial direction gives the following finite difference form of Equation (3).

$$\begin{aligned}
 h_i^*(z^* + \delta z^*) = & (h_1^* + h_2^* + h_3^*) \cdot \frac{4\delta z^*}{3\eta_T^{*2}} \frac{(\epsilon^* + \kappa\alpha^*)}{VEL1} \\
 & + h_i^*(z^*) - h_i^*(z^*) \cdot \frac{4}{3} \frac{(\epsilon^* + \alpha^*)}{\eta_T^{*2}} \cdot \frac{\delta z^*}{VEL1} \\
 & + Q_i^* \delta z^* / VEL1
 \end{aligned}$$

where $VEL1 = \bar{U}/(U_I)_z$

or

$$h_1^*(z^* + \delta z^*) = \frac{h_1^* + h_2^* + h_3^* + (M - 3) h_1^*(z^*)}{M} + \frac{Q^* \delta z^*}{VEL \ 1}$$

where

$$M = (3\eta_T^{*2} VEL \ 1) / \delta z^* . 4(\epsilon^* + \kappa\alpha^*)$$

Knowing the inlet dimensionless enthalpy distribution one can march upwards in steps of δz^* until the exit of the bundle. The step size δz^* and η_T^* must be sized to make $M \geq 3$ for numerical stability.

Therefore

$$\frac{3\eta_T^{*2} VEL \ 1}{4\delta z^* (\epsilon^* + \kappa\alpha^*)} \geq 3$$

whence

$$\delta z^* \leq \frac{\eta_T^{*2} VEL \ 1}{4(\epsilon^* + \kappa\alpha^*)}$$

Thus the axial step size depends upon the flow split ratio $VEL \ 1$, the node size η_T^* and the mixing coefficient.

A similar stability criterion is derived below

for δz^* in Region II. The axial step size is determined by the smaller of the two values of δz^* .

The finite difference form of equation (4) is written for node i in Region 2 (Fig. 5) as follows

$$\begin{aligned} & \frac{C(h_2^* - h_i^*)}{\eta_s^*} + \frac{1}{\text{VEL } 2} \frac{h_i^*(z^* + \delta z^*) - h_i^*(\delta z^*)}{\delta z^*} \\ &= (\epsilon^* + \kappa\alpha^*) \frac{(h_2^* + h_3^* - 2h_i^*)}{\eta_s^{*2}} + (\epsilon^* + \kappa\alpha^*) \frac{(h_1^* - h_i^*)}{\eta_n^{*2}} \\ &+ Q^* \end{aligned}$$

whence the stability criteria obtained by keeping the coefficient of the h_i^* terms positive can be shown to be

$$\delta z^* \leq \frac{1}{\left[\frac{2(\epsilon^* + \kappa\alpha^*)}{\eta_s^{*2}} + \frac{(\epsilon^* + \kappa\alpha^*)}{\eta_n^{*2}} - \frac{|C|}{\eta_s^*} \right] \text{VEL } 2}$$

where VEL 2 = flow split ratio in Region 2

$$= \frac{\bar{U}}{(U_{II})_z}$$

(b) ENERGY III

The solution procedure for ENERGY III equations is now given. The ENERGY II code solution procedure follows automatically by neglecting the convective terms in the momentum equation and will not be discussed here.

The equations of continuity, momentum and energy for Regions I and II can be written in the finite difference form and are given by equations below³¹. By eliminating enthalpy in terms of density utilizing an equation of state and reference pressure one can write a momentum and energy equation for each node i in the (x,y) planes and an overall continuity equation (sum of the cross-flows for all nodes is zero).

For Regions I and II the equations may be written as follows for any node i

Momentum

$$(\text{COEFF1})_i \Delta \rho_i + (\text{COEFF2})_i \Delta U_i + 32.2 \Delta \rho = (\text{COEFF3})_i$$

ENERGY

$$(\text{COEFF4})_i \Delta \rho_i + (\text{COEFF5})_i \Delta U_i = (\text{EX1})_i \Delta Z + (\text{QA})_i \Delta Z$$

CONTINUITY

$$\sum_{i=1}^N w_c = 0 = \sum_{i=1}^N (\text{COEFF6})_i \Delta \rho_i + \sum_{i=1}^N (\text{COEFF7})_i \Delta U_i$$

where N = total number of nodes in Region I and II

Both for Regions I and II

$$\text{COEFF1} = \left[(U_z + \Delta U) + (U_z + \Delta U - U^*) + \left(\frac{1}{16} \right) F_D \Delta Z \right. \\ \left. \cdot (2U_z + \Delta U)^2 \right] / \lambda_a^2 + 16 \Delta Z$$

$$\text{COEFF2} = \left[\rho_z \left((2 + F_D \Delta Z / 2) * U_z + (1.0 + \frac{F_D \Delta Z}{8}) \Delta U \right. \right. \\ \left. \left. - U^* \right) / \lambda_a^2 \right]$$

$$\text{COEFF3} = (-\text{GBAR} * C * \Delta Z * A_c * (\bar{U}(I) - \bar{U}(2)) / \lambda_a) \\ - [\rho (32.2 \Delta Z + 1/2 F_D \Delta Z U_z^2 / \lambda_a^2)]$$

$$\text{COEFF4} = A(U_z + \Delta U) (R * (\rho_z + \Delta \rho) + H_z - H^*)$$

$$\text{COEFF5} = A(\rho_z) (H_z - H^*)$$

$$\text{COEFF6} = A(U_z + \Delta U)$$

$$\text{COEFF7} = A * \rho_z$$

$$\lambda_a = \lambda_{a1} \text{ in Region I}$$

$$= \lambda_{a2} \text{ in Region II}$$

$$R = \left(\frac{dH}{d\rho} \right) Z$$

Q and EX1 are defined in Chapter 2.4.

The non linear set of equations above can be written for each node i . Since there are N nodes there will be $2N$ equations and one continuity equation i.e. there are $(2N + 1)$ equations. For each node the unknowns to be determined are ρ and U . In addition one pressure P which is the same for all nodes at any axial position, z , must be determined. Therefore there are $2N + 1$ unknowns in $(2N + 1)$ equations. The equations are non-linear and are solved iteratively as follows .

Initially the coefficients are evaluated assuming $\Delta\rho$ and ΔU to be zero. The equations are solved simultaneously yielding updated values of $\Delta\rho$, ΔU and ΔP . The values of $\rho(z + \Delta z)$, $U(z + \Delta z)$ and $P(z + \Delta z)$ are determined from calculation of $\Delta\rho_i$ and ΔU_i . These are substituted back into the nonlinear coefficients in the equations and the updated equations are solved until the convergence criteria $\sum W_c < \epsilon$ is satisfied. Here ϵ was chosen to be 10^{-7} .

APPENDIX 4

Analysis of Data Using Subchannel Analysis

Version of ENERGY Code

Analysis of forced convection data reported in Ref. 17 was performed by using the subchannel analysis version of the ENERGY-I code called ENERGY-IS. The subchannel analysis form of ENERGY code maintains the basic region model discussed in Chapter 2.1 where the bundle is divided into two predominant regions. In the central region (Region I) the mixing effect is modeled by an enhanced effective eddy diffusivity ϵ_1^* and in the wall region a uniform swirl flow C_1 (see nomenclature and Appendix 1 for definition) is maintained along the duct wall. The axial velocities used in ENERGY-IS are the actual axial velocities in the voids of the bundle. Another difference between the two versions of ENERGY is the nodal network. The subchannel analysis requires each node to lie at the center of each subchannel and the area associated with each node is the actual subchannel area. The empirical constants ϵ_1^* and C_1 obtained from data analysis using ENERGY-IS can be related as follows to ϵ^* and C , obtained in Chapter 3, using the porous body model, ENERGY-I.

Let λ_a^- be the average axial porosity of the bundle, defined as follows:

$$\lambda_a^- = \frac{\text{void area}}{\text{total area}}$$

Let λ_{L_1} and λ_S be the lateral porosities in region I and region II, respectively, defined as follows:

$$\lambda_{L_1} = \frac{p - d}{p} \quad (\text{A.11})$$

$$\lambda_S = \frac{g}{t} \quad \text{where } t = g + \frac{d}{2} \quad (\text{A.12})$$

The width of region II, t , as discussed in Chapter 2.4, was taken in the analysis of data to be equal to $\frac{d}{2} + g$, where g = gap between the rod and wall and d = diameter of rod.

The energy equation in region I, for any node i , may be written as follows for the porous body version, ENERGY-I (subscript P represents the porous body model)

$$(\rho UA)_P \frac{dH}{dz} = (QA)_P + (EX1)_P \quad (\text{A.13})$$

where

$$\begin{aligned} EX1 = & GBAR_P \cdot \left(\frac{d_e}{\eta}\right) \cdot (\epsilon^* + \kappa \alpha_P^*) \cdot (A_C)_P \\ & \cdot (H_1 + H_2 + H_3 - 3H) \end{aligned} \quad (\text{A.14})$$

For the subchannel analysis version, ENERGY-IS

$$(\rho UA)_S \frac{dH}{dz} = (QA)_S + (EX1)_S \quad (\text{A.15})$$

where

$$\begin{aligned} EX1 = & GBAR_S \cdot \left(\frac{d_e}{\eta}\right) (\epsilon_1^* + SF \alpha_S^*) (AC)_S \\ & \cdot (H_1 + H_2 + H_3 - 3H) \end{aligned} \quad (\text{A.16})$$

where SF = a shape factor used in subchannel analysis version, ENERGY-IS. It can be easily seen

$$(\rho UA)_P = (\rho UA)_S \quad (A.17)$$

$$(QA)_S = (QA)_P \quad (A.18)$$

$$(GBAR)_P = \lambda_a^- (GBAR)_S \quad (A.19)$$

$$(AC)_P = \left(\frac{1}{\lambda_{L_1}} \right) (AC)_S \quad (A.20)$$

$$\alpha_P^* \left(= \frac{\alpha}{\bar{U} d_e} \right) = \frac{\alpha_S^*}{\lambda_a^-} \quad (A.21)$$

since

$$\bar{U} = \lambda_a^- \bar{V} \quad (A.22)$$

In order to obtain the same value of $\frac{dH}{dz}$ it is necessary that

$$\epsilon^* = \frac{\lambda_{L_1}}{\lambda_a^-} \epsilon_1^* \quad (A.23)$$

and also

$$\kappa = \lambda_{L_1} * SF \quad (A.24)$$

In a similar fashion one can show that

$$C = C_1 \frac{\lambda_S}{\lambda_a^-} \quad (A.25)$$

Since ϵ^* , C and κ have already been determined in Chapter 2 and 3, the values of SF, ϵ_1^* and C_1 can be calculated using equations (A.23), (A.24) and (A.25). These calculated values must be used in the subchannel analysis version, i.e., ENERGY-IS.

APPENDIX 5

The Listing of Computer Programs

1. ENERGY-I
2. ENERGY-II
3. ENERGY-III
4. ENERGY-IS

is given in the following published report:

"Manual For ENERGY I, II, III Computer Programs,"
by E. U. Khan, W. M. Rohsenow, A. A. Sonin and
N. E. Todreas, COO-2245-18TR, M.I.T., May, 1975