LMFBR In-Core Thermal-Hydraulics: The State of the Art and U.S. Research and Development Needs

E. U. Khan

April 1980

**Prepared for the U.S. Department of Energy
under Contract DE-AC06-76RLO 1830**

**Pacific Northwest Laboratory
Operated for the U.S. Department of Energy
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PACIFIC NORTHWEST LABORATORY
operated by
BATTELLE
for the
UNITED STATES DEPARTMENT OF ENERGY
Under Contract EY-76-C-06-1830

Printed in the United States of America
Available from
National Technical Information Service
United States Department of Commerce
5285 Port Royal Road
Springfield, Virginia 22151

Price: Printed Copy \$____*; Microfiche \$3.00

*Pages	NTIS Selling Price
001-025	\$4.00
026-050	\$4.50
051-075	\$5.25
076-100	\$6.00
101-125	\$6.50
126-150	\$7.25
151-175	\$8.00
176-200	\$9.00
201-225	\$9.25
226-250	\$9.50
251-275	\$10.75
276-300	\$11.00

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Prepared for the Fuels and Materials TMC and
the U.S. Department of Energy
under Contract DE-AC06-76RLO-1830

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Richland, Washington 99352

ACKNOWLEDGMENTS

The author wishes to express his appreciation to Dr. Robert J. Neuhold of the Department of Energy for his direction and support of the state-of-the-art study. The author also acknowledges the encouragement and support of Dr. D. S. Trent of Battelle. Thanks are due to Mr. Larry Huisingh, who helped to put together the references; to Drs. R. A. Markley, John Kim, and R. Singer for their contributions to the section on U.S. in-core generic thermal-hydraulic needs; to the author's colleagues in the Fluid and Thermal Engineering Section of Battelle, with whom he had useful discussions; and to Steve Matsumoto for editing assistance. The author also appreciates the help of Professor N. E. Todreas for the use of his class notes and the work of Dr. D. S. Rowe in Appendix C. Last but not least, thanks are due to Ms. Gail Poole, Mrs. Peggy Snyder, Ms. Cheryl Bartley, Mrs. Shirley Flink and Mrs. Paulette Goodenough for helping to bring the report to its present form.

ABSTRACT

At the request of the Reactor Research and Technology Division of the Department of Energy, the Pacific Northwest Laboratory (PNL) has prepared this report to summarize U.S. research and development needs in Liquid Metal Fast Breeder Reactor (LMFBR) in-core thermal-hydraulics. The report makes a detailed critical review of the literature relevant to predicting coolant flow and temperature fields in LMFBR core assemblies for nominal and non-nominal rod bundle geometries and reactor operating conditions. The review covers existing thermal-hydraulic models, computational methods, and experimental data useful for the design of an LMFBR core. The literature search made for this review included publications listed by Nuclear Science Abstracts and Energy Data Base as well as papers presented at key nuclear conferences. Based on this extensive review, the report discusses the accuracy with which the models predict flow and temperature fields in rod assemblies, identifying areas where analytical, experimental, and model development needs exist.

- The subchannel method in conjunction with existing experimental data is a reliable method for predicting temperature fields in LMFBR assemblies. A few improvements in models are needed, and experimental data are required to improve the models.
- Significant improvements in distributed-parameter (DP) and hybrid analytical methods are needed for LMFBR applications. Local measurements of velocity turbulent intensity, shear stress, and cladding circumferential temperature distributions are needed to improve the DP and hybrid methods.
- Experimental verification of global "whole-core" methods are needed at low-flow, mixed- and natural-convection, steady-state, and transient conditions of reactor operation. Nodalization studies are needed for global methods to determine that thermal hydraulic phenomena are adequately modeled.

- Analytical methods and experimental data are needed for analysis of distorted bundles.
- In-pile data is needed especially in the following areas: fuel models, power-to-melt, gap conductance, fission gas release, and restructuring; temperature measurements in bundles at low, medium, and high burnups; forced, mixed, and natural flow regimes.
- A detailed review and comparison of the various local methods that have been used for rod bundle thermal-hydraulic analysis should be made, and the most promising method should be chosen for further improvements. These improvements include multi-cell calculation capability, inclusion of the effect of the presence of a spacer, and improvements in modeling of turbulence transport properties.

The author of this report recommends that a detailed work plan be developed for future work with emphasis on the natural-convection cooling of the core and thermal-hydraulic analysis of distorted bundles.

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LMFBR IN-CORE THERMAL HYDRAULICS: THE STATE OF THE ART AND U.S. RESEARCH AND DEVELOPMENT NEEDS

1.0 INTRODUCTION

At the request of the Reactor Research and Technology Division of the Department of Energy, the Pacific Northwest Laboratory (PNL) has prepared this report to summarize U.S. research and development needs in Liquid Metal Fast Breeder Reactor (LMFBR) in-core thermal-hydraulics. The report makes a detailed critical review of the literature relevant to predicting coolant flow and temperature fields in LMFBR core assemblies. The review covers existing thermal-hydraulic models, computational methods, and experimental data useful for the design of an LMFBR core. The literature search made for this review included publications listed by Nuclear Science Abstracts and Energy Data Base as well as papers presented at key nuclear conferences. Based on this extensive review, the report discusses the accuracy with which the models predict flow and temperature fields in rod assemblies, identifying areas where analytical, experimental, and model development needs exist.

The economic performance of an LMFBR is closely linked to the endurance life of the reactor core fuel assemblies. The fuel assembly life is affected by peak cladding temperature and the temperature distributions in the coolant, fuel, and assembly structures. These temperature distributions should be such that burnup goals can be met without seriously impairing core life and performance. For constraining temperature levels by design, it is important that the fuel assembly temperature distributions be accurately predicted and the governing thermal-hydraulic phenomena be understood. An example of temperature-affected performance is that estimates for existing LMFBR designs indicate that pin burnups might be increased by about 500 to 600 MWd/T for each degree Fahrenheit reduction in hot spot cladding temperature.

Methods developed for predicting temperature distributions should be capable of accounting for the effects of geometrical differences and the wide range of operating conditions that exist in the fuel, blanket, control, and reflector assemblies of LMFBRs. These assemblies have significantly different geometrical characteristics that may affect fuel assembly temperature distributions and thus endurance life and performance. Important geometrical parameters include the rod diameter, rod pitch (center-to-center distance between rods), assembly size, and rod-to-duct-wall spacing. Fabrication tolerances are also an important geometrical consideration: tolerance limits can significantly affect coolant flow and temperature distributions. However, irradiation-induced distortions change the bundle geometry much more significantly with potentially more effect on core thermal-hydraulics. For this reason, distorted-geometry (non-nominal) bundles receive special attention in this report.

The prediction of assembly flow and temperature distributions must consider a wide range of coolant flow and thermal convection regimes which include laminar and turbulent flow; natural, forced, and mixed (forced + natural) convection; and steady-state and transient conditions of reactor operation. For example, a typical core thermal-hydraulic design analysis flow diagram (Figure 1.1) used by LMFBR designers illustrates the various thermal convection regimes that require accurate assembly flow and temperature predictions.

Listed below are the important thermal-hydraulic phenomena that affect LMFBR assembly flow and temperature distributions. Predictive methods should be able to account (model) for these phenomena:

- Intra-Assembly Phenomena
 - turbulent mixing (with or without spacers)
 - diversion crossflow mixing (with or without spacers; includes swirl flow and flow sweep caused by wire wrap)
 - thermal conduction

FIGURE 1.1. Core Thermal-Hydraulic Analysis Flow Diagram (Dashed lines enclose non-nominal conditions. Prepared from information from R. A. Markley and others at Westinghouse, Advanced Reactor Division)

- secondary flows
- inlet flow distribution
- buoyancy-induced flow redistribution and flow recirculation
- Interassembly Phenomena
 - heat transfer
 - buoyancy-induced interassembly flow redistribution

The presence of a spacer in the assembly affects the relative importance of these mechanisms for energy redistribution within an assembly. At low-flow conditions (which exist during part load operation, during a decay heat removal transient, or during a natural circulation transient) a number of other thermal-hydraulic phenomena become important. These include interassembly heat transfer, interassembly flow redistribution due to buoyancy, and intra-assembly flow redistribution due to buoyancy. Under the extreme conditions of large power gradients and very low flows, recirculation zones might form within assemblies, which can affect the various inter and intra assembly thermal-hydraulic phenomena mentioned earlier.

Listed below are additional parameters that affect assembly flow and coolant, fuel, and structure temperature distribution during steady-state and transient conditions of reactor operation.

- Fuel Rod Modeling Parameters
 - gap conductance
 - restructuring of fuel rod
 - fission gas release
 - material thermodynamic properties
 - fuel rod geometrical characteristics
- Constitutive Equational Parameters
 - friction factor
 - friction loss coefficients for orifices, grids, etc.
 - heat transfer coefficients
- Core-Related Parameters
 - inlet assembly flow as affected by orificing

All these geometrical parameters, reactor operation regimes, and thermal-hydraulic phenomena affect LMFBR core flow and temperature distributions. The purposes of this report are:

- to identify and discuss the relative importance of the significant parameters and thermal-hydraulic phenomena affecting flow and temperature fields in LMFBR core assemblies
- to describe the physical modeling of the thermal-hydraulic phenomena
- to critically review existing models and analytical methods
- to describe experiments that have been conducted to evaluate the models and methods
- to discuss the accuracy with which the developed models and methods predict flow and temperature fields under different geometrical and operating conditions
- to identify the areas where future model development, analytical and experimental needs exist.

SCOPE OF PRESENT STUDY

Since the field of LMFBR in-core thermal-hydraulics is extremely broad, the focus here is on application-oriented areas which are of fundamental value and of immediate concern to the LMFBR designers. The scope of this study does not include flow-induced vibrations, flow impedance phenomena, flow blockages, in-core instrumentation, flow cavitation, and safety-related phenomena such as boiling and condensation. Neither does the study cover upper and lower plena effects on core thermal-hydraulics, fluid hammer effects, control rod scram dynamics, flotation, etc. A detailed discussion of the various methods of including the uncertainty effects for predicting hot spot cladding temperatures is not included. A general discussion of the aforementioned subjects are covered by Tang et al. (1978, Book).^(a)

(a) References are divided into several categories. The references and further explanation can be found in Appendix A.

In the present state-of-the-art study, emphasis is placed on the hydrodynamics and thermal hydraulics of LMFBR fuel and blanket assemblies in both the steady-state and transient conditions, in forced, mixed, and natural convection regimes, and under laminar, transition, and turbulent flow conditions. Analytical methods and computer programs (codes), are discussed along with experimental techniques used to validate codes and to improve our understanding of the various thermal-hydraulic phenomena. Most importantly, the current state-of-the-art is assessed relative to several solution procedures which have been used for understanding and predicting rod bundle flow and temperature. Thermal-hydraulics needs specific to both homogeneous and heterogeneous reactor core designs are addressed. The thermodynamics of wire-wrapped, grid-spaced, and bare-rod bundles are also considered. The major emphasis is on wire-wrapped assemblies typical of the Fast Flux Test Facility, the Clinch River Breeder Reactor, and the Conceptual Design Study. The European and Japanese work in LMFBR in-core thermal-hydraulics is discussed in detail.

2.0 SUMMARY AND CONCLUSIONS

Investigators have developed a number of analytical methods for predicting LMFBR core thermal-hydraulics. Because of core geometrical complexities, various physical approximations are required to develop tractable analytical models that can be readily solved on high-speed computers. These approximations reduce model complexity and computational resource needs, but they also reduce model generality. A larger degree of empiricism is necessary, and as the degree of empiricism increases, the range of application of an analytical method diminishes for different geometric and operating conditions.

This report identifies five methods of rod bundle thermal hydraulic analysis, each relating to the approximations made in modeling the physical thermal-hydraulic phenomena and to the thermal resolutions needed for the solution of a particular problem. These methods are the one-dimensional approach, global analysis, subchannel analysis, distributed-parameter (DP) analysis, and hybrid analysis. The one-dimensional approach treats an entire assembly as a single calculation node. When used with hot channel factors, this methods leads to very conservative estimates of hot channel coolant and hot spot cladding temperatures. Consequently, the one-dimensional approach is no longer used. Each of the other four methods has its own range of application and is most useful in a particular flow and thermal regime:

- Global analysis lumps a number of subchannels into a single calculation node. The principle of global analysis is commonly used in calculations involving the whole LMFBR core since there is an upper limit to the number of calculation nodes that can be conveniently used with present day computers.
- Subchannel analysis is commonly used for the thermal-hydraulic analysis of a single fuel assembly. A bulk average value characterizes the hydrodynamic and thermal coolant conditions in each subchannel. Thermal-hydraulic interactions between subchannels are allowed. (An alternate method for thermal-hydraulic analysis of an assembly is

the porous body model. The limitations, range of application, and degree of empiricism of subchannel analysis and the porous body model (Appendix C) are very similar. Most comments on subchannel analysis are equally applicable to the porous body model.)

- Distributed-parameter (DP) analysis predicts local coolant and cladding temperature distributions on a scale much finer than a subchannel. Because of the fine nodalization required for the DP method, the number of equations to be solved is much larger than the number of equations in other methods. This requires increased storage and computational resources.
- The hybrid analysis method is a compromise between subchannel analysis and DP analysis; i.e., the subchannel is divided into a large number of cells, but the basic assumptions of the subchannel analysis are retained.

Table 2.1 is a list of recent LMFBF analytical methods for rod bundle thermal-hydraulic analysis and Figure 2.1 shows the evolution of these methods. Some of the capabilities and limitations of the basic analytical methods are discussed below.

Since global analysis methods lump a large number of nodes into a single calculation node, it is necessary to ensure that the local thermal-hydraulic phenomena are adequately modeled. This requires extensive comparison with data and/or more detailed methods of analysis.

The most widely used method of rod bundle thermal-hydraulic analysis is the subchannel analysis method. With experimental support, subchannel analysis can predict the peak coolant temperature in a typical LMFBF rod assembly within approximately $\pm 7\%$ of the assembly mixed-mean (axial) temperature rise. The assembly transverse temperature distribution and assembly average axial pressure drops (using existing correlations) can be satisfactorily predicted. However, improvements are needed in predicting duct wall circumferential temperature distribution in wire-wrapped assemblies, cladding circumferential temperature distribution, flow redistribution behind grid spacers, and channel coolant temperatures in regions of large temperature gradients such as duct

TABLE 2.1. List of Recent LMFBR Analytical Methods for
Rod-Bundle Thermal-Hydraulic Analysis

<u>Method or Code Name</u>	<u>Author and Reference</u>	
<u>Subchannel Analysis and Porous Body Models</u>		
COBRA-IV	Wheeler et al.	1976, N4 ^(a)
COMMIX	Domanus et al.	1977, N4
COTEC	Novendstern	1972, N4
ENERGY I, II, III	Khan et al.	1975, N4
FORCMIX	Graves	1972, N4
FULMIX I, II	Magee/Wei	1977, N4
ORRIBLE	Wantland	1974, N4
SIMPLE	Jones et al.	1977, N4
<u>Distributed-Parameter Method</u>		
	Bender	1969, N1
	Cook	1969, N1
	Oberjohn	1970, N1
	Carajilescov et al.	1976, N1
	Bartzis et al.	1977, N1
	Chuang et al.	1977, N3
	Kim et al.	1977, N3
	Yeung et al.	1978, N3
<u>Whole-Core Global Method</u>		
(steady state and transient)		
COBRA-Whole Core	George et al.	1979, N4
CORTAN	Khan et al.	1979, 04
<u>Hybrid Method</u>		
VELASCO-3D	Eifler and Nijssing	1979, 03

(a) See Appendix A: References

LOCAL HYDRO-THERMAL METHODS

SINGLE CELL, NO SPACERS,
FORCED-CONVECTION, STEADY
STATE, 2-DIMENSIONAL
(SEE TABLE 3.2 ALSO)

SUBCHANNEL ANALYSIS METHODS

FORCED/MIXED CONVECTION,
SPACERS, STEADY STATE OR
TRANSIENT

FIGURE 2.1. Evolution of U.S. Analytical Methods for LMFBR Rod Bundle Thermal-Hydraulic Analysis

walls. In forced convection, the major emphasis in subchannel analysis is on the solution of the energy equation and on the evaluation of empirical constants from existing thermal data, which are needed to match predictions with data. This approach is possible because the hydrodynamic and thermal fields are not strongly coupled in forced convection. For application where these fields are strongly coupled, such as in mixed/natural convection and for distorted bundles, more data is needed to assess the subchannel analysis method.

Distributed-parameter methods are commonly used for prediction of intra-channel flow and thermal fields. The developed DP methods are limited to single-cell calculations with the assumption of zero shear at the cell boundaries. Another common assumption is the fully developed flow field. For distorted bundles and for bundles with spacers, the full momentum equations need to be included and multicell calculations performed. More data is needed for understanding the thermal-hydraulic phenomena within subchannels, especially in the presence of spacers. This data will be useful in the development of models for the DP and hybrid analysis methods.

The hybrid method has been developed because the DP method requires large storage and computation times. However, the hybrid method needs significant improvement before use in design calculations. The shortcoming of the hybrid method is its inability to include the effects of spacers on the turbulent transport characteristics and the lack of thermal-hydraulic coupling between channels. The DP and hybrid methods need to be extended to include transients and mixed/natural convection regimes. A detailed discussion of the assumptions, approximations, and limitations of the analytical methods is presented in Section 3.0 of the report.

In order to assess, validate, and/or calibrate the analytical methods, a large body of hydrodynamic and thermal-hydraulic data has been generated for bare-rod, grid-spaced, and wire-wrapped bundles. Table 2.2 indicates the types of data that have been obtained in the rod bundle geometry. These are discussed in detail in Section 3.0 of this report. Other areas discussed in the report include the following:

- multiassembly computational methods and their application to core-restraint design
- thermal-hydraulics of geometrically distorted bundles
- turbulence models for bare-rod bundles (turbulence models that include the effect of spacers are needed)
- modeling of the thermal-hydraulics phenomena listed in Table 2.1 for bundles with different geometrical and operating characteristics and comparison of code predictions with existing data
- innovative design approaches to increase fuel burnup
- duty-cycle transients as they affect the assembly temperature distribution and assembly life
- a detailed comparison of the porous body model and subchannel analysis.

2.1 U.S. RESEARCH AND DEVELOPMENT NEEDS^(a)

The U.S. research and development needs identified for this report are separated into four categories:

1. Current Needs
2. Future Needs
3. General needs in related areas (current and future)
4. Project-Related Needs

2.1.1 Current Needs

Current needs include analytical and experimental areas where development would be of immediate benefit in understanding LMFBR thermal-hydraulics. The needs are listed as they apply to the global, subchannel, distributed-parameter, and hybrid analysis methods. Two areas of special importance for research and development are discussed separately: distorted bundles and natural convection. Fuel rod modeling is also discussed.

(a) Prepared with contributions from Drs. R. Singer (ANZ), R. Markley (WARD) and J. Kim (GE).

TABLE 2.2. Types of Hydrodynamic and Thermal-Hydraulic Data In Rod Bundles

Hydrodynamics (Laminar, Transition and Turbulent Flow)

Global and/or Subchannel Analyses

- friction factor or pressure drop correlations in bare-rod, grid-spaced, and wire-wrapped assemblies of different geometrical characteristics
- inlet flow distribution between subchannels (flow split) for bare-rod, grid-spaced, wire-wrapped assemblies of different geometrical characteristics
- flow or velocity distribution in channels in the vicinity of spacers.

Local or Distributed-Parameter Analyses

- distribution of time-mean velocity within channel
- distribution of turbulence intensities in the three directions or the distribution of the turbulence kinetic energy
- distribution of turbulent shear stress in the directions parallel and normal to the walls
- distribution of wall shear stress
- measurement of secondary flows.

Thermal-Hydraulics

- "interchannel mixing coefficient" for bare-rod, grid-spaced, and wire-wrapped bundles of different geometrical characteristics from heated pin, salt injection, and hot water injection experiments
- data on intra-assembly flow redistribution due to buoyancy, inter-assembly heat transfer, and interassembly flow redistribution due to buoyancy
- data on flow recirculation within an assembly.

Local Measurements

- local coolant temperature distribution within subchannels
- local cladding axial and circumferential temperature measurements
- temperature distribution under wire wraps and spacer grids
- data for distorted geometries on cladding temperature distribution due to pin-to-pin contact
- heat transfer coefficient estimation by local temperature measurements
- gap conductance between fuel and cladding.

Global Method

- A nodalization study should be performed to ensure that the thermal-hydraulic phenomena are being adequately modeled.
- Validation of the "whole-core" global analysis codes by experimental data in the mixed/natural convection regimes is required. This includes validation of the interassembly heat transfer and fluid flow model, the intra-assembly flow redistribution model, and the flow recirculation model. Tests with prototypic assemblies can be conducted at ORNL and WARD. Code validation in non-prototypic assemblies can be conducted at PNL (2 x 6 assembly). A multiassembly experiment is needed for validation of "whole-core" model for predicting interassembly heat transfer and flow redistribution.

Subchannel Analysis

- Improve the wire-wrap mixing model and develop wire-wrap mixing for alternate wire-wrap configurations.
- Improve the model for predicting flow redistribution behind spacer grids for different geometrical configurations of spacers.
- Improve the model for predicting temperatures in assembly regions with steep temperature gradients, such as duct walls. Conduct experiments specifically to improve temperature predictions in the wall region.
- Streamline the large number of similar subchannel methods that exist and develop one baseline thermal-hydraulic subchannel analysis method for use by LMFBR designers. The verified method should have flexibility and portability for use at different locations within the country.
- Obtain conduction shape factors for mixed/natural convection regimes at low Reynolds numbers.
- Quantitatively define the limits of the usefulness of subchannel analysis in the forced- and mixed-convection regimes.

- Continue conducting experiments to improve understanding of inlet flow split, friction factors, bundle tolerance effects, etc., in laminar and transition regimes, especially for bundles with p/d ratios in the range 1.05 to 1.2.
- Continue obtaining mixing data for development of the wire-wrap model and the eddy diffusivity models, preferably in heated bundles in the geometrical range $1.05 \leq p/d \leq 1.20$ and $8 \leq h/d \leq 52$ for straight start and other wire-wrap configurations.
- Develop experiments to assess the effect of shaved-edge wires and different wire-wrap configurations (including a low lead wire wrap on edge pins) on bundle flow distribution and temperature field.
- Develop experiments for bundles with fuel assembly geometries which contain 91 or 127 heated pins cooled by sodium. Considerable benefits can be accomplished (during hot channel analysis) by reduction in the uncertainty factor due to bundle size effect, flow maldistribution, and flow redistribution. This experiment should be conducted both at full-flow and low-flow conditions.

Distributed-Parameter (DP) Methods

- Existing DP methods for grid-spaced or wire-wrapped LMFBF assemblies need significant improvement, specifically in modeling local turbulence transport properties and how they are affected by the presence of spacers. Reliable verified DP methods are needed to provide input parameters to subchannel analysis and hybrid methods. A detailed review and comparison of existing DP methods that have been used for rod bundle analysis is required. The most promising method should be chosen for further improvements. These improvements include multi-cell calculation capability and inclusion of the effect of a spacer on the flow and thermal field.
- Existing DP methods usually assume fully developed flow. For distorted bundles and spacers, the full momentum equations need to be included.

- Large-scale experiments on simple geometries (for ease of measurements) are recommended for validation and calibration of DP methods. Measurements of local velocity distributions, turbulent intensity, wall shear stress, and turbulent shear stress distribution should be made for developing fully developed flow.
- Local measurements are required for intra-channel coolant temperature distribution, circumferential wall temperature distribution and its axial evolution, and circumferential duct wall temperature distribution.

Hybrid Method

At present the hybrid method needs significant development. The effect of spacers needs to be included prior to determining flow and temperature fields in LMFBF rod bundles.

Experiments of the same type as discussed with DP method needs are also needed for improving the hybrid method.

Two Areas of Special Importance for Research and Development

Analytical and supporting experimental needs have been described for each of the four types of analytical method. However, two important areas related to LMFBF core thermal-hydraulics need to be further emphasized. These are briefly discussed below.

Distorted-Bundles. Irradiation-induced geometrical distortions in a bundle can have a significant impact on the hydrodynamics and thermal-hydraulics. It is expected that the analytical methods, specifically the DP and hybrid methods, will be useful for analysis of distorted bundles. However, the importance of out-of-pile and in-pile thermal-hydraulic data cannot be overemphasized in establishing margins to fuel/cladding lifetime limits. The analysis of distorted bundles is an area of research of development where virtually no work has been done.

Natural Convection. Several technical issues involving in-core natural-convection circulation of coolant require consideration:

1. Dynamics of the transition from forced- to natural-convection flow for all potential initial operating conditions and transient initiators; for example,
 - complete loss of onsite power during normal full-power operation
 - complete loss of onsite power during N-1 or N-2 loop operation
 - complete loss of onsite power during reduced power operation, i.e., during start-up with an overcooled core, during fuel handling, etc.
 - spurious reactor scrams with either a delayed pump trip or subsequent pump seizure
 - loss or degradation of secondary and/or steam heat transport systems
 - temporary loss of heat sink.
2. Thermal and hydraulic interactions between subassemblies; for example,
 - flow redistribution between subassemblies of differing hydraulic, thermal, and neutronic characteristics
 - transverse thermal conduction between adjacent subassemblies
 - gross flow reversal in blanket subassemblies
 - local flow recirculation within subassemblies.
3. Estimation of the conservatism in fuel/cladding lifetime limits
 - dynamics of boiling under natural-circulation conditions
 - subchannel enthalpy peaking factors during transient natural-circulation events
 - contribution of local flow stagnation/recirculation on fuel and blanket elements cumulative damage function.

The technical approaches needed to resolve these issues are:

1. development of a transient fast running, comprehensive whole-core thermal-hydraulic code that can simulate an entire reactor (fuel and radial blanket) and a more rigorous benchmark computer program
2. generation of out-of-pile fuel and blanket simulation subassembly natural-circulation data encompassing the full range of conditions described under issue number 1 above
3. generation of in-reactor natural-circulation data encompassing both issues number 1 and 2 above
4. verification of the code developed in approach number 1 above with data generated from approaches number 2 and 3.

Fuel Rod Modeling Needs

During transients the temperature distribution can be significantly affected by the thermal inertia of the fuel rod. Fuel rod model development must continue in conjunction with gathering of in-pile and out-of-pile data. Data is needed on fuel rod gap conductance, fission gas release, restructuring of rod during irradiation, power-to-melt, etc.

2.1.2 Future Needs

The importance of the analytical and experimental needs identified here is dependent on reactor evolution.

- Development of self-orificing concepts may be needed as well as development of variable orificing for radial blanket assemblies and possibly for control assemblies.
- Thermal-hydraulic performance of ductless and vented assemblies for future LMFBRs should be evaluated and developed, if necessary.
- Methods are needed for predicting flow redistribution behind spacer grids, intra-assembly heat transfer in gridded assemblies; understanding of the grid-spaced assembly performance by out-of-pile and in-reactor tests is extremely important before the best spacer system for future LMFBRs is selected.

- Thermal modeling of fuel rod system for advanced fuels such as carbide and/or thorium fuels is needed with respect to fission gas release rates, power-to-melt, restructuring effects, gap material and conductance, etc.

2.1.3 General Needs in Related Areas

Analytical methods and tests need development for understanding, reducing, and eliminating thermal striping problems in LMFBRs.

- Transient peaking factors should be defined and their use in reactor design should be identified.
- The "gray areas" between thermal-hydraulic design and thermal-hydraulic aspects of reactor safety need detailed definition to prevent them from going unnoticed. These include the effects of gas bubble passage through the core, partial blockages, and nucleate boiling. Understanding these effects will also become more important in future LMFBR designs as thermal-hydraulic operating conditions will be "pushed" for improved performance.
- The needs for in-pile thermal-hydraulic information include power-to-melt, gap conductance, fuel restructuring, programmed startup, and fission gas release.

2.1.4 Project-Related Needs

Project-related needs cover considerations and problems that need to be addressed for a specific project.

- Analyses and tests in water and sodium should be developed to increase understanding of cavitation, the effect on performance of cavitation damage to LMFBR assemblies, limitations on orifice velocities, and the pressure drop characterization of components.
- Wire screens should be developed as an alternative to inlet orifices for LMFBR assemblies. The British have tested and found screens to be reliable and effective in improving cavitation margin.

- Tests are needed to improve the use of piston rings in LMFBR, to reduce leakage and to better understand their lifetime effects.

2.2 CONCLUSIONS AND RECOMMENDATIONS

- The subchannel analysis method in conjunction with existing experimental data is a reliable method for predicting temperature fields in LMFBR assemblies. A few improvements in models are needed, and experimental data is required to improve the models.
- Significant improvements in distributed-parameter (DP) and hybrid analytical methods are needed for LMFBR applications. Local measurements of velocity turbulent intensity, shear stress, and cladding circumferential temperature distributions are needed to improve the DP and hybrid methods.
- Experimental verification of global "whole-core" methods are needed at low-flow, mixed- and natural-convection, steady-state, and transient conditions of reactor operation. Nodalization studies are needed for global methods to determine that thermal hydraulic phenomena are adequately modeled.
- Analytical methods and experimental data are needed for analysis of distorted bundles.
- In-pile data is needed especially in the following areas: fuel models, power-to-melt, gap conductance, fission gas release, restructuring; temperature measurements in bundles at low, medium, and high burnups; forced, mixed, and natural flow regimes.
- A detailed review and comparison of the various local methods (Table 3.1) that have been used for rod bundle thermal-hydraulic analysis should be made, and the most promising method should be chosen for further improvements. These improvements include multi-cell calculation capability, inclusion of the effect of the presence of a spacer, and improvements in modeling of turbulence transport properties.

The author of this report recommends that a detailed work plan be developed for future work with emphasis on the natural-convection cooling of the core and thermal-hydraulic analysis of distorted bundles.

3.0 ANALYTICAL AND EXPERIMENTAL STUDIES IN LMFBR IN-CORE THERMAL-HYDRAULICS

Before specifying how this report is organized, it is necessary to define several terms used to classify the literature on LMFBR in-core thermal-hydraulics.

3.1 DEFINITION OF TERMS

To aid in identification of the generic needs for research, the literature on LMFBR in-core thermal-hydraulics is separated here into several broad, overlapping subject categories and subcategories:

- fuel assemblies with nominal and non-nominal geometries
- nominal and non-nominal reactor operation conditions
- natural, forced, and mixed modes of coolant convection
- local and global hydrodynamics^(a)
- local and global hydrothermodynamics.

Nominal fuel assembly geometry is the as-fabricated geometry that the fuel assembly has at the beginning of life. An assembly is subjected to a wide range of power and power gradients over its life in the stringent reactor environment. This results in non-nominal fuel assembly geometries: pin and duct wall distortions due to temperature gradients across the assembly and to temperature-induced differential swelling and irradiation-induced creep in the pins, cladding, and duct wall.

Nominal operation conditions pertain to steady-state conditions during reactor operation at full or partial load. Since partial load operation can vary over a wide range, it is necessary to consider both laminar and turbulent coolant flow as well as forced and mixed (forced + natural) coolant convection regimes of assembly operation. Nominal conditions refer to steady-state forced convection. Non-nominal conditions refer to transient and/or mixed and natural convection.

(a) "Local" refers to an area which is significantly small compared to the subchannel area. "Global" refers to the entire subchannel area or to a number of subchannels grouped together to form a calculation node.

The main topics of interest in local hydrodynamics are the laminar and turbulent velocity distributions within subchannels (on a local basis); the distribution of turbulent intensities, turbulent shear stress, and wall shear stress; estimation of eddy diffusivity of momentum transfer from the localized prediction or measurement; estimation of secondary flows and anisotropy of eddy diffusivity; axial development of hydrodynamic flow field; and the effect of spacer grids.

The main areas of interest in global hydrodynamics are flow orificing, flow and velocity distributions on a subchannel basis with and without spacers, momentum eddy diffusivity estimations for subchannel codes, and estimation or measurement of inlet flow split in the various channels of the bundle. In addition to the above, estimation of friction factor and pressure drop in bundles of different geometric and operating characteristics in laminar, turbulent, and mixed convection conditions are included in global hydrothermodynamics.

Local hydrothermodynamics concerns the analytical methods and their experimental verification for predicting localized axial, azimuthal, and radial coolant temperature distributions within subchannels and also the local cladding temperature distribution. In addition, localized cladding temperature distribution under wire wrap and spacer grid are included within this category. Specific thermal parameters that need to be determined in order to accomplish the above tasks include the distribution of eddy diffusivities of heat in radial and circumferential directions and the Nusselt number or its equivalent.

Global hydrothermodynamics concerns itself with global analytical and experimental effort for determining subchannel temperature distribution. In addition, fuel-rod-related effort has been included in this category. Entrance region solutions for developing hydrodynamic and thermal fields have been included in this category also. In local and global hydrothermodynamics, laminar, turbulent, and mixed-convection regimes are considered.

The subsections of section 3 divide the state-of-the-art literature according to fuel bundle (assembly) geometry and reactor operation conditions:

1. nominal bundles and nominal conditions including mixed convection (Section 3.2)

2. non-nominal bundles and nominal conditions including mixed convection (Section 3.3)
3. nominal bundles and non-nominal conditions (Section 3.4).

3.2 NOMINAL BUNDLES AND NOMINAL CONDITIONS INCLUDING MIXED CONVECTION

This discussion considers local and global hydrodynamics and then local and global hydrothermodynamics.

3.2.1 Hydrodynamics - Local Studies

Local hydrodynamics concerns the prediction of detailed coolant velocity distributions within rod bundle subchannels. These velocity distributions are required to accurately predict three-dimensional temperature distributions in the coolant and cladding of fuel pins, especially in regions of steep temperature gradients, such as regions near rod surfaces and duct walls. Local methods of predicting flow fields, as well as temperature fields, are also required because they back up the global methods, providing improved descriptions of physical processes from which one can obtain improved coefficients as input parameters to the global (or lumped) methods. Verified local hydrodynamic methods of predicting velocity fields are discussed in detail here, though they are perhaps one of the most important areas for future work. Local hydrodynamics in bare rod bundles is discussed first because an understanding of the thermal-hydraulic phenomena in bare rod bundles is basic to understanding work on bundles with spacers.

Bare Rod Bundle - Analytical Methods

A number of studies have been conducted to predict velocity distributions in rod bundles for various rod spacings and Reynolds numbers. This information is a prerequisite to the solution of thermal energy equations. For triangular rod bundles with no spacers, analytical laminar flow solutions are available for velocity fields (Sparrow 1959, N1) and for the heat transfer for the case of uniform peripheral surface temperature (Sparrow 1961, N3). The analysis presented by Axford (1968, N1) is applied to coolants in either slug or laminar flow. Dwyer (1970, N3) and Hsu (1972, N3) have also obtained laminar flow velocity profiles in rod bundles.

In pioneering work on turbulent flow in rod arrays, Deissler and Taylor (1956, N1) developed an iterative graphical procedure for evaluating velocity and temperature fields. The method essentially consists of dividing the characteristic flow region into a number of radial segments laterally bounded by "velocity gradient" lines, across which no momentum transport occurs. From force balances on each of these segments and assumptions regarding radial velocity distributions (universal velocity profile obtained from circular tube) constant velocity lines may be determined. The calculations are repeated until the requirement that "velocity gradient lines" are perpendicular to constant-velocity lines is satisfied throughout the whole flow region. Dwyer (1966, N1) gives a review of the work prior to 1966. Dwyer's method for determining the turbulent velocity profile is similar to Deissler's but improved on Deissler's thermal solution scheme. Dwyer's and Deissler's methods, like most other methods for predicting local velocity distributions, are for isolated, symmetrical 30° sections (45° sections for square arrays) of channels. Isolated, symmetrical sections are used for analysis rather than a number of adjacent channels because the adjacent channels would require impractically large amounts of computation to predict local flow fields. Based on work extending from 1960-1966 (see Dwyer 1966, N1) one of Dwyer's important findings is that the model for the annular gap can only be used (without major errors) for approximating geometric conditions of the pin bundle if $p/d \geq 1.35$.

With the advent of the present-day computers, a number of numerical solutions to the Navier-Stokes equation for steady, fully developed turbulent flow with incompressible, temperature-independent property fluid have been obtained. The Navier-Stokes equations have used either the vorticity-velocity or the primitive pressure-velocity variables. In all such methods for turbulent flows, one requires a semi-empirical method for evaluating the turbulent diffusion terms in the momentum equations. Table 3.1, which shows the various levels of solution procedures for rod bundle analyses, also shows that the methods designed to provide the Reynolds stresses (as the turbulent diffusion terms) can be generally classified into three categories of decreasing degrees of empiricism, namely:

TABLE 3.1 Solution Procedures for Rod Bundle Analysis

Method	Methodology	Parameters	Type of Flow
1. Lumped-parameter or sub-channel analysis (slug flow in channels)	Finite control volume, mass, momentum and energy balance	f, Nu	laminar or turbulent
2. Integral methods	Integrate N-S and energy equations over finite control volumes using velocity and temperature profiles which satisfy boundary conditions	U, T	laminar or turbulent
3. Differential equations, distributed parameter			
a. Laminar flow	N-S equations and energy equation	μ, k	laminar
b. Eddy diffusivity			
1. Prandtl's mixing length theory or Buleev's model (e.g., law of wall)	Solve N-S and energy equations	ϵ_M, ϵ_H	turbulent
2. Relate turbulent shear stress to axial velocity gradients	Solve N-S and energy equations distributed	ϵ_M, ϵ_H	turbulent
c. Higher-order closure models which use turbulence energy or Reynolds stress transport equation	Solve N-S equations, energy equations, and turbulence transport equations	length scales L, Λ , etc.	turbulent

TABLE 3.1 (Continued)

Method	Methodology	Parameters	Type of Flow
3.c. (continued)			
1. One-equation model	Solve N-S equation, energy equation, and equation for turbulent kinetic energy (K_T)	C_v , where $K_T = C_v k^{1/2} \ell$	turbulent
2. Two-equation model	Solve N-S equation, energy equation, and transport equation for K_T and dissipation	$\ell = \frac{k^{3/2}}{\epsilon}$	turbulent
3. Higher-order models	Solve N-S equation, energy equation, six shear stress equations, and three turbulent heat flux equations	--	turbulent
d. Direct numerical simulation of turbulence	Solve N-S equations, energy equations, and sub-grid scale equations	--	laminar or turbulent

- eddy diffusivity models
- higher-order closure models
- direct numerical simulation of turbulence.

Detailed discussions of these methods follow below.

The eddy diffusivity models can be further reclassified into two categories. In first category are the phenomenological turbulence models, which includes the Prandtl's mixing length and Buleev's model used by Bender and Switick, Chuang, and Ramm and Johannsen (Table 3.2). The usefulness of the

TABLE 3.2. Distributed-Parameter Methods Used for Predicting
Local Turbulent Velocity Fields in Rod Bundles
with No Spacers

Author/Reference	Method Used (see Table 3.1)
Bender and Switick (1969, N1)	3.b.1 (Buleev's mixing length, 1967, N1)
Bender and Magee (1969, N1)	3.b.1 and Ibragimov's method
Carajilescov and Todreas (1976, N1)	3.c.1
Chuang (1977, N3)	3.b.1
Cook (1969, N1)	3.b.1
Deissler and Taylor (1956, N1)	universal law of wall for circular tube used for rod bundles and graphical procedure)
Dwyer (1966, N1)	similar to Deissler and Taylor with improvement on graphical procedure
Ibragimov (1966, N1)(a)	3.b.1 (modified)
Maubach and Rehme (1973, N1)(a)	2
Ramm and Johannsen (1972, N1)(a)	3.b.1 (modified)
Schumann (1973, N1)(a)	3.d (finite difference)
Slagter (1978, N3)(a)	3.b.2 (finite element)
Eifler and Nijssing (1966, N1)(a) (VELASCO) (1972, IAEA)(a)	3.b.2
Meyder (VERA-TERA) (1974, N3)(a)	3.b.2
Grotzbach (1977, N1)(a)	3.d

(a) Work done outside the U.S.

eddy diffusivity concept (Equation 1) is that it plays an important role in the search for universal or invariant quantities which can be measured in simple geometries and transferred to more complex geometries:

$$\text{Definition: } \epsilon_M(\vec{X}) \equiv - \overline{u_j u_i} / (\partial u_j / \partial x_i) \quad (1)$$

In the phenomenological approach, given or assumed eddy diffusivity profile is employed in the following manner. Integration of Equation 1 can yield a velocity distribution if one relates $u_j u_i$ to the shear stress distribution. An example is the Prandtl mixing length theory which, on integration, gives the logarithmic law (Schlichting 1968, Book, Ch. XIX). Another example is Buleev's model, which was used to predict either the eddy diffusivity distribution for the mixing length distribution Bender and Switick (1969, N1) and Ramm and Johannsen (1972, N1) .

The second eddy diffusivity approach is to substitute the eddy diffusivity distribution in the momentum differential equation and to integrate the equations using finite-difference methods. This method does not have to suffer from the limitations of the first methods, namely eddy diffusivities may be forced to zero wherever velocity gradients are zero. In addition, in this method, the boundary conditions for velocity and temperature at the surfaces can be satisfied (or forced to), unlike in the first method, and gradients of the dependent variables can be accurately reproduced. Methods that fall in this category include the work by Eifler and Nijssing on the VELASCO code, by Slagter and by Meyder (Table 3.2). The VELASCO and VERA-TERA (Meyder) codes use the finite-difference techniques for solution of the Navier-Stokes equations whereas the VITESSE code by Slagter uses the finite-element technique.

The methods of Bender and Switick, Bender and Magee, Cook, and Deissler and Taylor (Table 3.2) all neglect secondary flows and hence, predict local wall shear stresses to increase monotonically from the position facing the gap to the position facing the subchannel centerline. The VELASCO code assumes a secondary flow profile. However, the proposed shape of the peripheral velocity distribution is contrary to experimental evidence (Rowe 1973, EN1), which

shows more than one secondary loop can occur in a channel. However, as shown by the experiments of Rehme (see Table 3.3), for low p/d ratios the models for anisotropy ratio ($\epsilon_{M\phi}/\epsilon_{Mr}$) and models for secondary flow need revision. This is because even at low p/d ratios VELASCO uses an anisotropy ratio of 2 to 3, whereas Rehme's experiments show these ratios can be as high as 200 for bundles with low p/d ratios. With increasing p/d, the anisotropy ratio decreases.

TABLE 3.3. Local Detailed Measurements in Bare Rod and Grid-Spaced Rod Bundles (a)

Reference	p/d	L/D _H	Re x 10 ⁻³	Velocity		Turbulent Intensity			Cross Correlations		$\frac{w}{w}$
				Axial	Secondary	$\bar{z}$	$\bar{r}$	$\bar{\theta}$	$\frac{v_r v_z}{\bar{v}_r \bar{v}_z}$	$\frac{v_\theta v_z}{\bar{v}_\theta \bar{v}_z}$	
Eifler-Nijssing (1967, EN1)	1.05	139	15,30,50	X							
	1.10	90	15,30,50	X							
	1.15	65	15,30,50	X							
Subbotin et al. (1971, EN1)	1.05	154	18.1-81	X							X
	1.10	100	18.8-81	X							X
	1.20	72	18.8-81	X							X
Kjellstrom (1971, EN1)	1.217	81	149-373	X	X	X	X	X	X		X
Hall and Svenningsson (1973, EN1)	1.217	81	270		X						X
Trupp and Azad (1975, EN1)	1.2	51	12-84	X		X	X	X	X	X	X
	1.35	30	12-84	X		X	X	X	X	X	X
	1.50	20	12-84	X		X	X	X	X	X	
Rowe (1973, EN1)	1.25	138	50-200	X		X					
	1.25	122	50-200								
Trippe and Weinberg ^(b) (1978, EN1)	1.30	44 107	30-60	X							
Rehme ^(c) (1978, N1)	1.071	133	83-87			X	X	X	X	X	X
	1.148	97	123	X		X	X	X	X	X	X
	1.402	60	194			X	X	X	X	X	X
Carajilescov and Todreas (1976, N1)	1.123	77	27	X		X	X	X	X		

(a) part of this table from Carajilescov and Todreas (1976, N1)

(b) with spacer grid in 19-rod bundle

(c) single rod array enclosed, in duct walls.

The diffusivity approach has limitations. As discussed by Lawn (1970, N1) a well-known phenomenon in turbulent flows is the displacement of the zero-shear surface from the surface of maximum velocity where strongly asymmetric profiles are present. This is apparently caused by the diffusion of turbulence energy across the zero-shear surface. None of the previous work incorporates this mechanism and therefore cannot predict this effect.

The next level of empiricism for modeling the turbulence diffusion or the turbulence shear stress terms are the higher-order closure models. These assume that the fluctuations are random in nature and that a few of the statistical properties of turbulence are supposed to obey laws of generation, dissipation, and transport. The one- and two-equation models relate the Reynolds stresses to axial velocity gradients with the turbulent viscosity or eddy diffusivity as follows. The diffusion term in the Navier-Stokes equation is given by Equation 2.

$$\text{Diffusion term} = \frac{\partial}{\partial x_i} \left\{ \frac{\partial u_j}{\partial x_i} - \overline{u_j u_i} \right\}$$

Using Equation (1),

$$\text{Diffusion Term} = \frac{\partial}{\partial x_i} (\nu + \epsilon_M) \frac{\partial u_j}{\partial x_i} \quad (2)$$

$\epsilon_M = C_\nu K^{1/2} \ell$ where K is the turbulence kinetic energy, ℓ is the mixing length, and C_ν is a constant. The value of K is calculated by a transport equation. The value of ℓ can be given either by an algebraic expression (one-equation model) or a transport equation of a related property (two-equation model). Carajilescov and Todreas (1976, N1) were perhaps the first to apply a one-equation turbulence model to rod bundles for computing the detailed description of axial and secondary velocity fields and the wall shear stress distributions of steady, full developed turbulent flows with incompressible temperature-independent fluid. They also conducted experiments to measure the distributions of axial velocity, turbulence kinetic energy, and Reynolds stresses using a Laser-Doppler velocimeter (Table 3.3). From the

experimental results, Carajilescov and Todreas (1976, N1) proposed a new mixing length. Comparison between experiments and the analytical results was good; however, an experimental search for secondary flows was inconclusive.

Higher-order models (3.C in Table 3.1) have not been used for rod bundle calculations, probably due to computing time limitations. The basic idea is to avoid using the concept of eddy diffusivity to model the turbulence stress tensor; instead, the stresses may themselves be represented by a differential equation obtained as a moment of the momentum equation. This gives the Reynolds stress and turbulence energy transport equations. Higher-order equations of fluctuating quantities may be obtained by taking up further moments (Hinze 1975, Book) of the Navier-Stokes equation.

Work on direct numerical simulation of turbulence has been done by Schumann (1973, N1) and Grotzbach (1977, N1). Their method is based on a difference form of the complete equations of the conservation of mass, momentum and energy. The fluxes of momentum and heat within the grid cells are described by subgrid scale models. At the very low molecular Prandtl number of liquid metals, the model shows that complete resolution of all relevant temperature fluctuations can be achieved on the basis of presently realizable grids. So far the method has been used for simple geometries such as parallel plane channels and concentric annuli. The major advantage of the accurate results is to compute quantities which so far have not been determined experimentally but which are of decisive importance in the development of higher-order statistical turbulence models.

Because of computation time limitations, the higher-order methods have not been used for complex geometries (Table 3.4). However, it is important to note that, as mentioned earlier, the solutions in simple geometries provide valuable guidance on the empirical terms which must be used in lower-order models.

Another powerful tool for analyzing two or three-dimensional flows is the von Karman (Schlichting) integral method. Maubach and Rehme (1973, N1) have used this method for rod bundle calculations. Similar to global (lumped-parameter) methods, differential equations are integrated over control-volumes--but these volumes are of differential length. The integrated

TABLE 3.4. Typical Estimated Computing Times for Rod Bundle Calculations

<u>Analysis</u>	<u>Type of Rod Bundle</u>	<u>Computer</u>	<u>Approximate Time</u>
Slug flow	9 rod bundle (with axial nodes)	IBM 360	30 sec
Integral	19 rod bundle (no axial nodes)	IBM 360/168	0.05 sec
Eddy diffusivity	37 rod bundle	IBM 370/165	2 min
Reynolds stresses	annulus	IBM 370/165	3 hr

equations are not solved directly; instead, suitable empirical profiles for dependent variables u and T (or h) in each control volume cross section are sought. The advantage over subchannel analysis is that the forms of u and T profiles can be chosen to satisfy as many known boundary conditions as possible, particularly the no-slip wall conditions. The velocity and temperature profiles are integrated to determine the integral functions in the integrated differential equations, which are then solved. This method has been widely used in conduction heat transfer (Arpaci, Book, 1966) calculations.

The limitations of the methods of Bender and Switick, Ibragimov, Eifler and Nijssing, Ram and Johannsen have been discussed in great detail by Carajilescov and Todreas (1976, N1). The method of Carajilescov requires considerable computation time and therefore is limited to single-cell, sections of symmetry. Single-cell, segment of symmetry calculations are of interest to understand the local thermal-hydraulic phenomena, but for real bundles where spacer grids or wire wraps are used to support the pins, the assumption of isolated regions of symmetry will not hold good; this is especially true for wire-wrapped bundles where large transverse flows and large axial gradients for transverse flows occur. For this reason, multicell (multi-channel) methods of local velocity predictions are needed for "real"

bundles. The only multi-channel, distributed-parameter method developed so far appears to be that of Slagter (IAEA-13, 1979). However, it is presently limited to assemblies without spacers.

Bare Rod Bundle - Experimental Measurements

The published data for measurements of axial velocity, turbulence kinetic energy, Reynolds stresses, etc., are limited (Table 3.3).

Eifler and Nijssing (1967, EN1) and Subbotin (1971, EN1) both used pitot tubes for velocity measurements. Both investigations show that the proximity of the wall blocking the gap of the subchannel distorted the velocity distribution in a way that should not be expected by symmetry considerations (Figure 3.1). Kjellstrom (1971, EN1) used a larger test section so that measurements were taken far away from the blocking wall as indicated in Figure 3.2. He used Preston tubes to measure wall shear stress in addition to other parameters (Table 3.3). Attempts to measure secondary flows resulted in large scatter as was also found by Hall and Svenningsson (1973, EN1). The most complete experimental work appears to be that of Trupp and Azad (1975, EN1) using a hot-wire anemometer in a test bundle shown in Figure 3.3. Measurements of tangential shear stress ($v'_\phi v'_z$) were very scattered. Table 3.3 shows

FIGURE 3.1. Cross Section of Test Section used by Eifler and Nijssing (1967, EN1) for Velocity Field Measurements

FIGURE 3.2. Cross Section of Test Section Used by Kjellstrom (1971, EN1)

the other experimental parameters measured. Carajilescov and Todreas (1976, N1) used the Laser-Doppler velocimeter to measure the axial velocity, turbulence kinetic energy, and Reynolds stresses. From the experimental data a new mixing length distribution is proposed. Experimentally a search conducted for secondary flows was inconclusive.

More recent experiments (Rehme, 1978, N1) on wall subchannel velocity distribution and shear stress have also failed to find convincing evidence of secondary flows. The existence of secondary flows is unresolved at this time. Trippe and Weinberg (1978, EN1) have measured detailed velocity distributions (130 measurements per subchannel) in a 19-rod assembly with and without spacer grids. The prediction of the VELASCO (Eifler and Nijssing 1966, N1; 1972, IAEA) and VITESSE (Slagter 1979, IAEA) codes matched the data of Trippe and Weinberg well for the case of no spacer grid.

FIGURE 3.3. Cross Section of Test Section Used by Trupp and Azad (1975, EN1)

Grid-Spaced Bundles - Analytical Methods and Experimental Measurements

In a real assembly, the presence of spacers will affect the hydrodynamics and hydrothermodynamics. The two types of spacers generally used are the spacer grids (a number of different designs have been suggested, e.g., spark eroded, honeycomb, line of sight) and wire wraps (a number of wire-wrap configurations have been suggested but the single-wire, straight-start is the most commonly used wire-wrap configuration).

The presence of spacer grids will distort the velocity profiles and perhaps introduce an asymmetry in the velocity distribution as compared to a bundle without spacers (e.g., Nijssing 1967, EN1). This is due to several reasons discussed below. Vortex formation downstream of the grid augments intensity

and scale of turbulence and thus entails an increase of the turbulent diffusivity. The incremental turbulent diffusion increases as the ratio of the free-flow area in the grid to the undisturbed bundle flow area decreases. The relative importance of this effect appears to depend strongly on the shape of the grid device. Due to the decay of turbulence, the diffusivity will decrease with axial distance downstream of a grid toward an asymptotic value. Since grid spacers represent obstructions to the axial coolant flow, they will induce macroscopic interactions between subchannels. This is because the relative flow resistance between the center and the edge or corner channels may change significantly between gridded and ungridded regions, causing a redistribution of flow.

Due to these complexities and due to the difficulties in modeling the turbulence effects, prediction of local velocity distributions in the vicinity of spacer grids is extremely difficult and, although attempts have been made (Nijssing 1974, N4), they have met with little success from a distributed-parameter point of view. It appears that not a single distributed-parameter method exists for predicting local velocity distributions in the presence of spacer grids--either on a single-cell or multicell basis. However, several experiments have been conducted to determine local velocity fields in triangular-rod-array, grid-spaced bundles.

In addition to the experiments of Trippe and Weinberg (Table 3.3), local velocity distributions in 5:1-scale, grid-supported fuel pin arrays have been reported by McAreavey and Betts (1979, IAEA). They have reported work carried out on a 77-pin assembly irradiated in Dounreay Fast Reactor (DFR). There are six grids located at various axial positions. Measurements of detailed flow patterns at several measurement planes were made using a pin-mounted total-head tube capable of covering the measurement patterns of center and edge channels (Figures 3.4 and 3.5). The only U.S. work reported on local velocity measurements in grid-spaced LMFBR geometry bundles is the Westinghouse Research Laboratories (WRL) (Roidt et al. 1974, N2) 11:1-scale test in air.

For the spacer-grid to be given serious consideration as a possible spacer system for LMFBRs significant effort is required for the development of prediction methods and their experimental verification. A more detailed

discussion on aspects of spacer grids as they affect global hydrodynamics and interchannel mixing is postponed until Section 3.2.2.

Wire-Wrapped Bundles - Analytical Methods and Experimental Measurement

Wire-wrap support for assembly pins has been selected by the French for their Phenix and Super-Phenix reactors and by the U.S. for the Fast Flux Test Facility (FFTF) and Clinch River Breeder Reactor (CRBR). It is expected that the first core for the CDS (Conceptual Design Study) will use wire wraps as the spacer. The major advantage of wire wraps is that they can be economically manufactured, as compared to spacer grids, within specified tolerance limits. A number of wire-wrap configurations have been suggested (Kaplan and Punches 1977, E04), the most common being the straight start concept with a single wire around each pin. At high burnups, a possibility of bundle/duct interaction exists if the bundle, by virtue of a higher average temperature, swells more than the surrounding duct. In this situation, the most desirable wire-wrap configuration would be one where the minimum stress was developed due to bundle/duct interaction. The deformation of bundles for the various wire-wrap configurations have been discussed in detail by Kaplan and Punches.

The only published method for predicting fully developed local velocity distributions in a bundle with spacers is that of Chuang et al. (1977, N3). Figure 3.6 shows the unit cell selected for their analysis. The method has several limitations, which are discussed in detail near the end of the section (3.2.3) on local hydrothermodynamics (bundles with spacers). The current understanding of the local thermo-fluid phenomena in wire-wrapped bundles is far from complete and a number of simple, reliable experiments are needed to develop the understanding. Only a few experiments on wire-wrap assembly local velocity measurements have been reported to date as shown in Table 3.5.

Bartholet et al. (1977 EN1) measured the axial velocity contours for interior, wall, and corner subchannels. Figure 3.7 shows a typical axial velocity contour for an interior channel. Two-dimensional COBRA probes were used to measure crossflow in the gaps between adjacent pins. Figure 3.8 shows the average circumferential velocity as a function of the wire-wrap angle. Recent analysis of the data (Basehore and George 1979, N4) shows that there are inconsistencies in the measurement of axial flows. This inconsistency can

TABLE 3.5. Detailed Local Velocity Distribution Measurement in Wire-Wrapped Rod Bundles

Reference	p/d	Geometry		Channel	Scale	Fluid	Reynolds Number	Measured Parameters			
		h/d	N rods					Laminar		Turbulent	
								Axial Velocity	Traverse Velocity	Axial Velocity	Traverse Velocity
Bartholet et al. (1977, EN1)	1.25	11.9	39	interior	10.87:1	Air	turbulent			X	X
	1.25	11.9	39	wall	10.87:1	Air	turbulent			X	X
	1.25	1.9	39	corner	10.87:1	Air	turbulent			X	X
Ohtake et al. (1976, EN1)	1.18	34.8	37	interior	5.5:1	Air	6800			X	X
				wall			11000				
				corner			150000				
Bartholet et al. (1979)	blanket assembly geometry (to be published)			interior wall corner	5:1	Air	---				
Chen et al. (1974, EN1)	1.25	48	61	wall	1:1	Water	Laminar & Turbulent	X		X	
McAreavey and Betts (1979, IAEA)	1.25	14.8	37	interior wall corner	scale not given	Air	Turbulent		X	X (3-D velocity distribution measured by 3-D probe)	

be seen in Figure 3.9, where the axial velocity profile for interior channels B, D, and F in Figure 3.7 are plotted. Channels B, D, and F are located symmetrically with respect to the wire position, and hence their average axial velocity profiles should be similar--they are not. However, the independently measured crossflow data is valuable.

At present, sufficient data have not been obtained in the wall region to make an adequate comparison with the velocity field measured by Chen et al. (1974, EN1) using a laser-Doppler velocimeter. An experimental program has been initiated to obtain additional data at Westinghouse Research Laboratory. Chen measured the development of the axial velocity profile in the wall channels of a 61-pin, wire-wrapped bundle with water as the working fluid. He investigated both the laminar and turbulent flow regimes. Of interest to this discussion is Chen's development of the velocity profile from the entrance region. It seems that approximately two wire-wrap pitch lengths were needed to fully develop the velocity field in turbulent flow. He also observed circular flow and decrease of axial flow velocity in the subchannel containing the spacer wire.

FIGURE 3.4. Center-Channel Scan Details

This result was also observed by Ohtake et al. (1976, EN1), who measured the detailed velocity distributions in interior and wall channels of a 37-pin scaled (x5) bundle (Table 3.5). This result does not agree with the results of Bartholet et al, (1977, EN1) who found that the peak axial velocities occurred with the wire wrap present in the channel, while minimum velocities

FIGURE 3.5. Edge-Channel Scan Details

take place with no wire wrap present in the subchannel. Figures 3.10, 3.11, and 3.12 show the typical detailed transverse and axial velocity profiles obtained by Ohtake et al. They observed circular flow regions (Figure 3.11) in channels containing the wire.

McAreavey and Betts (1979, IAEA) have used a triple-limb yawmeter to determine the magnitude and direction of flow relative to the pin axis. They claim that the flow angle can be determined to an accuracy of about 2%. The velocity field is then used in the energy equation to determine the temperature field in forced convection.

FIGURE 3.6. Triangular Pitch Wire-Wrap Rod Bundle Geometry and FATHOM-360 Unit Cell

The experiments conducted so far for determining detailed velocity fields with wire-wrapped bundles are incomplete, and further experiments are required to gain a better understanding of the thermo-fluid phenomena. Specifically, the transverse interaction between the interior and wall channels needs to be further studied by measurement of flow patterns in the interior channels and the adjacent wall channels. This will provide good information regarding the bundle size effect. More reliable axial velocity measurements are needed. In addition, the measurements of static pressure distributions within the bundle

FIGURE 3.7. Axial Velocity Distribution (ft/sec) 33 Inches From the Downstream End of Pitot Tube Travel

are needed to understand the complex interaction between the pressure-driven diversion crossflow rate and the wire- (obstruction-) driven sweeping crossflow rate.

FIGURE 3.8. Average Circumferential Velocity Versus Wire-Wrap Angle

3.2.2 Hydrodynamics - Global Studies

Global hydrodynamics concerns the subchannel flow distribution and pressure drop in the rod bundle assembly of a reactor. Lumped-parameter exchange coefficients are needed from local hydrodynamic methods for use as input parameters in the global codes.

The problem of subchannel flow distribution in a pin bundle in forced convection resolves itself into a distinction between coolant flow in interior subchannels and coolant flow in edge and corner subchannels. As an approximation, the hydraulic diameter flow split is used to determine the flow split into the three types of channels under the assumption of uniform axial pressure gradient. This approach has two major difficulties. First, for grid-spaced bundles, the result will depend on the way in which subchannels are defined, e.g., whether edge subchannels are rectangular or triangular in shape.

FIGURE 3.9. Axial Profile of Average Axial Velocity for Interior Subchannels B, D, and F

Second, the variation in the relative axial flow resistance between a central and wall channel, such as that caused by a spacer grid, can cause a redistribution of flow, which can significantly influence the transverse interchange of momentum between subchannels, leading to smaller variations in mean velocity. Behind the spacer grid the flow distribution would be over or underpredicted if the hydraulic diameter concept is used and if transverse momentum effects are not adequately accounted for. An attempt to deal with this effect through a momentum mixing coefficient involving some knowledge of turbulent diffusivities has been proposed by Nijssing (1974, N4).

FIGURE 3.10. Transverse Velocity and Flow Direction of Interior Subchannel
(Flow Rate of Subassembly: $0.33 \text{ m}^3/\text{sec}$; Bundle Average
Reynolds Number: 15,100)

FIGURE 3.11. Transverse Velocity and Flow Direction of Interior Subchannel.
 (Flow Rate of Subassembly: $0.33 \text{ m}^3/\text{sec}$; Bundle Average
 Reynolds Number: 15,100)

FIGURE 3.12. Axial Velocity Distribution of Interior Subchannel
(Flow Rate of Subassembly: $0.33 \text{ m}^3/\text{sec}$)

Before discussing the work on grid-spaced and wire-wrapped bundles, it is useful to look at the earlier attempts to obtain a flow distribution and friction factor (or pressure drop) in an ungridded, infinite-rod array bundle. This was done by an extension of the circular tube and the equivalent annulus results using the equivalent-diameter concept. A number of different flow channels can have the same hydraulic diameter as a given tube but obviously different cross-sectional shapes. Momentum and heat transfer can only be predicted using the hydraulic diameter concept if the momentum and heat transfer resistances are not dependent upon the actual cross-sectional shape. If the resistances are primarily in the wall region, then the hydraulic diameter concept would work well since the Reynolds number similarity [i.e. $Re = (4/\mu \cdot m/p_H)$] or the perimeter per unit mass flow is maintained as a consequence of the definition of hydraulic diameter. This would be true for turbulent flows, but not for laminar flows for which the momentum resistance is uniform in the cross section. For turbulent flows the use of the equivalent diameter fails when p/d decreases to the extent that the wall sublayer (where the momentum resistance exists) thickness becomes large relative to the distance between the tubes. Similar arguments hold good for the Nusselt number. For liquids with Prandtl number near 1, the Nusselt number is insensitive to the boundary conditions since the resistance is concentrated near the wall. This is not true for liquid metals.

For rod arrays, in general, it is necessary to solve the basic governing equations (in detail within a subchannel) to determine friction factor and Nusselt number (or by detailed measurement) unless the flow is turbulent ($p/d \geq 11$ and $Pr \approx 1$), in which case circular tube results are good to within $\pm 10\%$. This is the reason why most of the global hydrodynamic work in rod arrays has been experimental, especially for liquid metals.

Figure 3.13 shows a comparison of the friction coefficients for fully developed laminar flow parallel to an array of un baffled circular tubes (Sparrow and Loeffler 1959, N1). Other predictions (Tang et al., 1959, Book, p. 133) for unwrapped rod bundles show a strong dependence of friction factor on the p/d ratio.

FIGURE 3.13. Friction Coefficients for Fully Developed Laminar Flow Parallel to an Array of Circular Tubes (Sparrow and Loeffler 1959, N1; redrawn by N. E. Todreas, MIT)

The British (McAreavy and Betts 1979, IAEA) and the Germans have performed detailed experiments to study the flow distribution and pressure drop characteristics of grid-spaced assemblies and assemblies without grid spacers.

Experiments for determining flow distribution in grid-supported fuel pin arrays have been performed on large-scale (X5) models of representative portions of assemblies using air as a working fluid and Reynolds number similarity. In addition, detailed flow patterns have been obtained in a 77-pin DFR-irradiated assemblies which contained six spacer grids. The results of these experiments have been described by McAreavy and Betts (1979, IAEA) as follows. These tend to support the results of other investigators.

- Flows in subchannels appeared to behave very much as independent flows in terms of flow distribution. Thus the mass flow in an edge channel tended to be constant from a position just downstream of a grid to just upstream of the next grid. The ratio of flows in center and edge channels was sensitive to grid geometry and channel geometry. Thus overcooling in edge channels could be corrected by use of deliberate blockage features attached to grid edges.
- The flow split between subchannels appeared to be amenable to simple flow split calculations based on entry, exit, and friction losses, and hydraulic diameter concepts with accuracies no better than about 10%. The effect of pin support dimples in grids was found to be very marked, both in terms of flow distribution and coolant mixing.
- Pressure drop was calculable by simple means, but empirical factors for friction loss and for support dimple loss were necessary. The friction loss due to pins in particular appeared to be affected considerably by the presence of the grids, and values 30% higher than standard smooth pipe friction loss correlations were frequently observed, with no sign of decay within the grid separation distance (typically about 20 diameters). This exceeded the excess friction loss as a function of pitch/diameter ratio predicted by Ibragimov et al. (See McAreavey and Betts 1979, IAEA) and Rehme, (1972, N2), for example.

The correction of overcooling of the edge channels, discussed in the first bulleted paragraph above, by use of deliberate blockage features (skirts) attached to grid edges, has also been described by Weber and Cornet (1979, IAEA). The Germans (Hoffmann, Hofmann, and Rehme 1979, IAEA) have also

reported a considerable amount of experimental work for determining flow distribution and pressure losses in rod bundles with and without spacer grids. Considerable attention was given to spacer-grid design and its effect on flow distribution, pressure losses, and flow mixing. Figure 3.14 shows the spacer drag coefficients (K-factors) versus Reynolds number in a bundle consisting of 12 rods. In order to study the effect of the leading edge on the drag coefficient, the leading edge was chamfered (SII). Because of a burr caused by the working process, this resulted in an increase in the drag coefficient. As a consequence of the increased drag coefficient, the blockage of the flow cross section by the grid was reduced from 35% (blockage) to 28%, despite a sharp leading edge. Rounding of the leading edge (SIIIR) greatly reduced the drag coefficient even further.

FIGURE 3.14. Drag Coefficients of Various Grid-Type Spacers

Hydraulic experiments by Hoffman, Hofmann, and Rehme (1979, IAEA) were performed on a 19-rod assembly enclosed in a hexagonal duct with and without spacers. Local velocity fields (within subchannels) were determined using Pitot tubes - up to 130 measuring points per subchannel. The spacer height and the blockage ratio were varied to see their effect on channel flow redistribution. The undisturbed (no spacers), fully-developed flow distribution was determined as the reference condition. The prediction of the VELASCO (Eifler and Nijssing 1973, 01) and VITESSE codes (Slagter 1979, IAEA) showed very satisfactory agreement with data for local velocity fields. The mean subchannel mass flow computed from the local velocity measurements agreed well with data. However the Germans' own global code ARTIS was unable to predict the mean subchannel flows until effort was made to match the data.

Velocity fields measured at various distances (L) downstream of grid-type spacers showed (Figure 3.15) no undisturbed flow distribution in this case. The isotach field at 3.6 hydraulic diameter, D_H , downstream of the spacer shows that the effect of the spacer contours are still present. At $18 D_H$ the local disturbances within subchannels have balanced out but undisturbed subassembly flow distribution has not been achieved. Figure 3.16 also shows the channel mass flow rates obtained by integration of the measured isotach fields. Comparison of the flow distribution behind the spacer with the modified code ARTIS (see previous paragraph) shows that the axial mass flow distributions in the various channels behind a spacer are not predicted well. Figure 3.17 shows the results from a similar set of experiments but with different spacer designs. It can be seen that the height of the spacer has little influence on the flow redistribution behind a spacer but that the spacer blockage has a profound influence. These results are important in deciding the most suitable spacer designs. Other important grid parameters which affect flow distribution are the detailed design of the grid pips. Grid losses depend strongly on the particular design style and do not lend themselves to general correlations. Therefore experiments are required to determine pressure losses across grids.

Fully Developed Velocity Distribution

Axially Developing Velocity Profiles

FIGURE 3.15. Velocity Distribution and Profiles in a 19-Rod Bundle ($Re = 60,000$; $p/d = 1.30$; $w/d = 1.17$)

The U.S. effort has mainly centered on wire-wrapped assemblies. The single-wire-per-rod, straight-start configuration is the current reference design, although other configurations have been investigated (Wei 1977, N4). The presence of wire wrap considerably changes the flow field within an assembly as compared to a spacer-gridded bundle. As the fluid tends to flow vertically upwards, the wire wrap imparts a relatively large transverse momentum to the fluid, which results in a continuous flow redistribution within the bundle. At the duct wall there is net swirl flow parallel to the duct wall.

FIGURE 3.16. Mass Flow Distribution in a 19-Rod Bundle With Grid Spacers ($Re = 60,000$; $p/d = 1.30$; $w/d = 1.17$)

One of the concerns in wire-wrapped bundles is the method of computing the division of flow between the central, wall, and corner channels in the presence of wire wraps. Novendstern (1972, N2) developed a method for predicting the flow split between channels using the hydraulic diameter concept. However a number of tests conducted on assemblies of different sizes (Chen 1974, EN1; Pederson 1974, EN2; Lorenz and Ginsberg 1977, EN4) show that the hydraulic diameter concept does not hold good for predicting flow split in assemblies that have a geometrical characteristic similar to the reference fuel assembly. The experiments show a relatively uniform average axial velocity in the three different types of channels in the bundle.

FIGURE 3.17. Mass Flow Distribution in a 19-Rod Bundle With Varying Grid Spacer Blockages and Grid Heights ($Re = 60,000$; $p/d = 1.3$; $w/d = 1.17$)

McAreavy and Betts (1979, IAEA) report similar findings in Britain for fuel assemblies with a p/d close to 1.26 and have attributed the uniform axial velocity in the bundle to a "relatively greater transverse momentum exchange". For the typical blanket assemblies with a p/d of ~ 1.04 , the British found the hydraulic diameter concept predicts the flow split well. However, U.S. experiments (Chiu 1977, EN2) have shown that the hydraulic diameter method does not satisfactorily predict the data for a 61-pin blanket assembly (Figures 3.18, 3.19, 3.20, and 3.21). The data clearly show that the flow split parameters X_1 and X_2 (X_1 and X_2 are defined as the ratios of the average axial interior and wall subchannels to the bundle average axial velocity, respectively) saturate to a constant value when the bundle Reynolds number (Re) is greater than 10,000. There is a transition flow regime in which flow split parameters change as Re changes. As Re increases from laminar to turbulent flows, flow split parameters change steadily from their laminar values to their saturated

FIGURE 3.18. Interior Subchannel Flow Split Parameter X_1 Versus Re for the Section of $h/d = 8.0$ and $p/d = 1.063$

FIGURE 3.19. Edge Subchannel Flow Split Parameter X_2 Versus Re for the Test Section of $h/d = 8.0$ and $p/d = 1.063$

FIGURE 3.20. Interior Subchannel Flow Split Parameter X_1 Versus Re for the Test Section of $h/d = 4$ and $p/d = 1.063$

FIGURE 3.21. Edge Subchannel Flow Split Parameter X_2 Versus Re for the Test Section of $h/d = 4$ and $p/d = 1.063$

values. An interesting result found by Chiu is the effect of wire lead length on flow split parameters. In addition, the bundle tolerance factor affects flow split. The parameter F , called the tolerance factor, is defined such that when $F=1$, all the tolerance in the bundle is uniformly scattered among the gaps between interior subchannels, and when $F=0$, all the tolerance is uniformly scattered among the gaps between edge rods and assembly duct. When $F=0.78$, the tolerances existing in internal gaps and edge gaps are equal. The tolerance factor has a significant effect on the flow split. Based on these observations Chiu developed a model for estimating turbulent flow split between channels (Chiu et al. 1978, N2). Chiu assumes that the flow split for the corner and wall channels are equal because the large swirl flow will equalize the flows in the edge and corner channels. Chiu's predictions matched the data within $\pm 5\%$ for bundles with geometrical parameters in the range $1.063 \leq p/d \leq 1.28$ and $4 \leq h/d \leq 52$ and for turbulent flow. Flow split models in the laminar flow range are discussed on page 41.

For calculating pressure drop in wire-wrapped fuel assemblies, the most commonly used method is that of Novendstern (1972, N2). The semi-empirical method predicts a number of sets of data (Rehme 1967, EN2; Reihman 1969, EN2; Rehme 1965, EN2) to $\pm 14\%$ at a 95% confidence level. The data were taken in the turbulent flow regime over a wide range of geometries. The Novendstern correlation, recommended for high Reynolds number (≥ 5000), is

$$\Delta P = M f_2(L/De) \left(v^2/2g_c \right) \quad (3)$$

where M is a multiplication factor that increases the pressure drop above that calculated without the spacer, and f is the smooth-tube friction factor. The factor M , which accounts for the wire lead and the fuel element p/d ratio is

$$M = \frac{1.034}{(p/d)^{.124}} + \left(\frac{29.7(p/d)^{6.94} Re^{0.086}}{(h/d)^{2.239}} \right)^{0.855} \quad (4)$$

An interesting result for friction factor in fuel assemblies at low Reynolds number is the smooth transition from turbulent to laminar flow. For a typical fuel assembly configuration, Additon and Parziale (1977, 04) show a plot of f versus Re which has the form (also see Tang et al. 1978, Book)

$$f = 84/Re \quad (5)$$

where $Re \leq 1000$.

The smooth transition from laminar to turbulent flow in the rod bundle has been explained by Ramm et al. (1974, N3) as due to a progressive movement of the minimum rod gap laminarization in the radial and circumferential directions of a subchannel.

One of the latest correlations for predicting the bundle average friction factor as well as the friction factors for individual channels in forced convection is due to Chiu, et al. (see Appendix B). Chiu et al. emphasize obtaining correlations for individual channel friction factors from which bundle average friction factors were derived. Rehme, Novendstern and Chiu predict bundle average friction factors (turbulent flows) to $\pm 15\%$ of data. The applicable range for the correlation developed is:

$$\begin{aligned} 1.06 &\leq p/d \leq 1.317 \\ 4 &\leq h/d \leq 96 \\ 400 &\leq Re \leq 15,000 \\ 19 &\leq N_R \leq 217 \end{aligned} \quad (6)$$

where N_R is the number of rods within the assembly.

For wire-wrapped blanket assemblies, Engel et al. (1979, EN2) have developed correlations for friction factor in the laminar, transition, and turbulent flow regimes for low- p/d -ratio assemblies typical of blanket assemblies by analysis of a number of European and U.S. data. A smooth transition region from turbulent to laminar flow that occurred over the Reynolds number range from 5000 to 400 characterized the resulting friction factor behavior. The continuous transition region was explained in terms of the fraction of the flow area in turbulent flow. The authors found that the friction factors calculated from individual subchannel measurements could be correlated by the same expression found for rod-bundle-averaged conditions.

In calculating flow areas and equivalent diameters, it is assumed that the wire wrap is distributed uniformly in all subchannels.

The British (McAreavy and Betts 1979, IAEA) and the Germans (Hoffmann, Hofmann, and Rehme 1979, IAEA) have reported that tests over a wide range of bundle geometries and bundle sizes have shown that the bundle average pressure drop can be predicted to $\pm 5\%$ by Rehme's correlation (1967, EN2, 1979, IAEA-5). The range of geometrical parameters tested by the British are from $p/d = 1.03$ to 1.26, h/d from 9 to 23, and bundle sizes from 37 to 325 pins. Figure 3.22 shows a plot of the ratio of the effective friction factor to smooth-tube friction factor versus p/d ratio at a Reynolds number of 10^5 . Rehme's correlation has been verified over a wide range of Reynolds numbers in the turbulent flow regime. Some U.S. findings show that Rehme's correlation does not predict well in the laminar and transition regimes.^(a)

FIGURE 3.22. Friction Factors as Function of p/d and h/d Ratio for 61-Rod Bundles; Helical Wire Spacer Types (37/61-Rod Bundles)

(a) The author obtained this information in personal communications with R. Markley of Westinghouse, Advanced Reactor Division.

In the laminar range, the friction factor was correlated by the expression $f = 110/Re$, in the turbulent range by $f = 0.55/Re^{0.25}$, and in the transition range by

$$f = \frac{110}{Re} (1-\psi)^{1/2} + \frac{0.55}{Re^{0.25}} \sqrt{\psi} \quad (7)$$

where $\psi = \frac{Re-400}{4600}$ is the intermittency factor.

3.2.3 Hydrothermodynamics - Local Studies

Global methods of predicting temperature distributions (Section 3.2.4) have shown promise for design calculations but have required extensive experimental verification, usually under prototypic conditions, because of the inherent assumptions and simplifications in these methods (discussed in Section 2.0). Because these simplifications preclude accurate predictions, local distributed-parameter or sub-subchannel analysis methods are needed:

- Calculation (France and Ginsberg 1972, N3) have shown that large uncertainties exist with respect to the transport properties for momentum and energy transfer. These transport properties are used for the calculation of the respective quantity between subchannels. These properties cannot be described by simple correlations but are dependent upon various boundary conditions (Ramm and Johannsen 1975a, N3)
- Prediction of cladding temperatures are not accurate enough for subsequent structural analysis because the resolution in the circumferential direction is not high enough
- The use of the concept of a heat transfer coefficient in strongly varying geometries is questionable as indicated by Hsu (1972, 03) as it can generate locally infinite values for h . In rod-bundle global analysis methods the correlations for " h " may not reflect the true geometry, and under conditions of steep power gradients the mixed-mean sub-channel temperature may be higher than adjacent cladding temperature; i.e., the heat transfer coefficient may be negative.

For these reasons, distribution of local velocity and temperature and heat flux are needed within subchannels.

The local hydrothermodynamic studies in rod bundle geometry have received considerable attention from as early as the late 1950s. Nearly all studies with a very few exceptions are for bare rod bundles. These studies can be separated into two categories: single-cell or multicell studies. Single-cell studies usually confine attention to a "unit cell" within the bundle which is isolated from other cells; i.e., there is no mass, momentum, or energy transport across its boundaries. The single-cell analysis is only correct in the case of an infinite rod array with uniform power generation. If the bundle is enclosed by a wall, distortion of geometry occurs, or a power gradient exists across the bundle (conditions usually prevalent for LMFBR assemblies) then the single-cell analysis can no longer give satisfactory results. Consequently, a multicell analysis is needed for describing the local flow and temperature fields in LMFBR assemblies.

The single-cell and multicell studies can further be classified as single-region or multiregion studies. Single-region studies describe the temperature field in one of the three regions--coolant, cladding, or fuel. Multiregion studies require the simultaneous solution of the velocity distribution in the coolant and the simultaneous solution of the temperature field in the coolant, cladding, and fuel. Table 3.6 shows the various local hydrothermodynamic studies in rod bundles with no spacer grids. Unless specified, the studies are for the fully developed hydrodynamic and thermal fields.

Laminar Forced-Convection Studies - Bare Rod Bundles

For fully developed laminar flow in circular tubes, the Nusselt number (Nu) is only a function of heating boundary condition. For uniform heat flux and uniform temperature boundary condition, the Nusselt numbers, Nu_Q and Nu_T , are 3.656 and 4.364, respectively (Rohsenow and Choi 1961, Book). Circular tube or equivalent annulus results for f and Nu cannot be used for rod arrays using the hydraulic diameter concept unless the flow is turbulent, $p/d > 1.15$, and $Pr \sim 1$. The reasons for this are similar to those discussed in Section 3.2.1 for local hydrodynamics. It is therefore necessary to solve

TABLE 3.6. Single-Region and Multiregion Analytical Methods of Local Distributed-Parameter Analysis^(a) (Symmetrically Located Rods)

Method Single-Region Method (boundary conditions)	Laminar		Laminar-Mixed Convection		Slug		Turbulent	
	interior	edge/ corner	interior	edge/ corner	interior	edge/ corner	interior	edge/ corner
$U_w = 0$	1	--	--	--	--	--	--	--
$T_w = \text{const.}$	2	--	--	--	6	--	12,13	--
$q_w = \text{const.}$	3	--	--	--	6	--	12,13,14	--
unspecified since $Pr = 0.7$	--	--	--	--	--	--	15,16	--
Multiregion Method	4	--	5	5	7,8,9	10,11	17,18,19	17

1. Sparrow and Loeffler (1959, EN1)	11. Wolf and Johannsen (1971, N3)
2. Sparrow et al. (1961, N3)	12. Dwyer and Berry (1972, N3)
3. Dwyer and Berry (1970c, N3)	13. Nijsing and Eifler (1973, N3)
4. Dwyer and Berry (1970b, N3)	14. Dwyer (1966, N3)
5. Kim and Wolf (1977, 03)	15. Nijsing et al. (1965, N3)
6. Dwyer and Berry (1970d, N3)	16. Nijsing et al. (1966, N3)
7. Axford (1967, N3)	17. Burkov (1972, N3)
8. Dwyer and Berry (1970a, N3)	18. Dwyer et al. (1972, N3)
9. Magee and Tromel (1966, N3)	19. Dwyer et al. (1972, N3)
10. Javeri (1976, N3)	

(a) Due to N. E. Todreas (M.I.T.)

the governing equations in detail within a subchannel to determine Nu for low-Prandtl-number fluids, for arrays with low p/d, and for nonhomogeneous boundary conditions. Alternatively experiments are required with detailed temperature measurements within subchannels.

For laminar flow there are a number of analytical solutions for predicting coolant and cladding temperature distributions in infinite rod arrays. These include the methods of Mottaghian and Wolf (1974, N3), Sparrow (1959, N3), Dwyer and Berry (1970, N3) and Axford (1965, N3).

Mottaghian and Wolf obtained solutions for round rod bundles using functions with special periodicity properties which lead to a totally analytical solution for pin bundles where the rods are arranged on circles. Sparrow et al. assumed uniform cladding (outside) temperature and obtained temperature fields for the coolant Figure 3.23. It is obvious that at low p/d ratios the

FIGURE 3.23. Nusselt Numbers for Fully Developed Laminar Flow Parallel to an Array of Circular Tubes (Redrawn by N. E. Todreas, MIT)

equivalent annulus concept does not hold good. As indicated by Nijssing (1969, N3) this kind of single-region analysis becomes questionable if the irregularity of the flow area increases, thus the need for multiregion analyses.

A multiregion analysis was performed by Axford (1965, N3) for the internal cell with coolant both in slug and laminar flow, utilizing the method of finite Fourier cosine transforms. Hsu (1972, N3) performed the multiregion

analysis in the same geometries for both laminar and slug flow. In all the previous references, the power generation rate was assumed uniform within the fuel region. This was extended by Wolf (1973, N3) to account for a tilted flux across the fuel, but the analysis is limited to slug flow. Dwyer and Berry (Oct. 1970, N3) used a numerical procedure to obtain a single-region laminar flow heat transfer solution for the boundary condition of circumferentially uniform heat flux on the rod surface (Figure 3.24). Values of rod-average Nusselt numbers are presented along with local wall temperature variation. This was extended by the same authors (#2, Oct. 1970, N3) to a two-region problem.

FIGURE 3.24. Wall Temperature Variation for Fully Developed Laminar Flow Parallel to an Array of Circular Tubes (Redrawn by N. E. Todreas, MIT)

The two-region problem was solved by an iterative, numerical solution. Figures 3.25a and 3.25b show the laminar flow region variation of Nusselt number with p/d for different cladding thickness and relative cladding thermal conductivity. The Nusselt number has been normalized by dividing by the average Nusselt number for uniform wall heat flux, $(\overline{Nu}_L)_{q''}$. Table 3.7 gives the values for the Nusselt number, $(Nu_L)_{q''}$. A number of slug-flow (uniform

FIGURE 3.25a. Typical Curves Showing the Effects of Variations in Relative Cladding Thickness and Relative Cladding Thermal Conductivity on the Average, Normalized, Laminar-Flow Nusselt Number

velocity with free slip at the wall) solutions have also been shown in Table 3.6 but will not be discussed here for the following reason. As indicated by Nijssing (1973, N3), the slug flow analysis is based on two simplifying assumptions: "(a) uniform velocity distribution (optimistic), (b) neglect of turbulence and secondary flow contribution to turbulent heat transport (pessimistic).

FIGURE 3.25b. Typical Curves Showing the Effects of Variations in Rod Spacing and Relative Cladding Thickness on the Average, Normalized, Laminar-Flow Nusselt Number

As a consequence, slug-flow calculations will yield heat-transfer predictions that are too high at low Peclet numbers (Nijssing 1969, N3) and too low at high Peclet numbers. Agreement of slug-flow predictions with experimental data or with results of more realistic analyses therefore will occur only under special conditions that cannot be a priori specified. The chief value of slug-flow results is that new calculation methods can be tested against them, as a limiting condition."

TABLE 3.7. Rod-Average Laminar-Flow Nusselt Numbers Illustrating the Effects of Variations in the Independent Parameters p/d , $(r_2 - r_1)/r_2$ and k_w/k_f

p/d	$\overline{Nu}_L$ q	Nu_L r	$\frac{r_2 - r_1}{r_2}$	Nu_L						
				$k_w/k_f = 0.1$	$k_w/k_f = 0.2$	$k_w/k_f = 0.4$	$k_w/k_f = 1.0$	$k_w/k_f = 2.0$	$k_w/k_f = 3.0$	$k_w/k_f = 4.0$
1.05	1.055	2.825	0.025	1.078	1.098	1.139	1.249	1.402	--	--
			0.050	1.097	1.138	1.212	1.399	1.626	--	--
			0.075	1.116	1.173	1.275	1.514	1.778	--	--
			0.100	1.131	1.204	1.327	1.602	1.885	--	--
			0.150	1.158	1.248	1.401	1.716	--	--	--
			0.200	1.173	1.275	1.443	1.779	--	--	--
			0.300	1.186	1.297	1.477	1.822	--	--	--
1.06	1.359	3.183	0.025	1.383	1.404	1.446	1.566	1.728	1.859	1.968
			0.050	1.404	1.446	1.527	1.725	1.964	2.137	2.262
			0.075	1.424	1.485	1.594	1.846	2.125	2.303	--
			0.100	1.441	1.517	1.649	1.938	2.230	--	--
			0.150	1.468	1.565	1.727	2.058	2.362	--	--
			0.200	1.485	1.593	1.772	2.121	2.424	--	--
			0.300	1.498	1.617	1.807	2.168	2.469	--	--
1.07	1.705	3.545	0.025	1.730	1.752	1.797	1.920	2.089	2.224	2.335
			0.050	1.755	1.796	1.880	2.085	2.330	2.503	2.639
			0.075	1.773	1.836	1.950	2.211	2.492	2.673	2.798
			0.100	1.790	1.870	2.007	2.305	2.607	2.781	--
			0.150	1.817	1.920	2.087	2.426	2.731	--	--
			0.200	1.836	1.949	2.134	2.491	2.796	--	--
			0.300	1.850	1.973	2.170	2.540	--	--	--
1.10	2.936	4.620	0.025	2.961	2.986	3.028	3.150	3.313	3.443	3.548
			0.050	2.986	3.027	3.110	3.310	3.543	3.703	3.820
			0.075	3.009	3.064	3.178	3.430	3.693	3.857	3.977
			0.100	3.022	3.100	3.234	3.519	3.793	3.960	4.058
			0.150	3.048	3.149	3.312	3.633	3.911	4.063	4.156
			0.200	3.064	3.178	3.356	3.692	3.968	4.109	4.200
			0.300	3.077	3.201	3.392	3.737	4.014	4.151	--
1.15	5.135	6.222	0.025	5.152	5.169	5.203	5.274	5.388	5.473	5.542
			0.050	5.169	5.202	5.250	5.386	5.539	5.641	5.715
			0.075	5.182	5.223	5.298	5.465	5.635	5.638	5.808
			0.150	5.212	5.273	5.387	5.596	5.772	5.865	5.923
			0.300	5.232	5.314	5.440	5.663	5.833	5.924	5.974
1.20	6.902	7.484	0.025	6.911	6.920	6.937	6.974	7.033	7.077	7.112
			0.050	6.919	6.937	6.962	7.032	7.111	7.164	7.202
			0.075	6.927	6.953	6.983	7.073	7.161	7.214	7.250
			0.150	6.946	6.974	7.033	7.141	7.232	7.280	7.309
			0.300	6.959	6.990	7.060	7.175	7.263	7.306	7.332
1.30	9.033	9.194	0.025	9.033	9.036	9.040	9.052	9.059	9.072	9.080
			0.050	9.036	9.040	9.048	9.059	9.080	9.092	9.101
			0.075	9.038	9.044	9.051	9.071	9.092	9.104	9.112
			0.150	9.042	9.052	9.059	9.087	9.108	9.119	9.126
			0.300	9.045	9.053	9.068	9.095	9.116	9.126	9.132

Turbulent Forced-Convection Studies - Bare Rod Bundles

Dwyer (1969, Survey) made a fairly extensive review of the work on heat transfer to liquid metals flowing in-line through unbaffled rod bundles. The review considers a triangular rod array with a uniform wall heat flux boundary condition for fully developed turbulent velocity and temperature fields. Theoretical predictions and experimental data for Nusselt number are compared for both close and wide spacings. An expression for the turbulent diffusivity ratio ($\bar{\psi} \equiv \epsilon_h/\epsilon_m$; this is discussed later in more detail) was determined empirically from data.

$$\bar{\psi} = 1 - \frac{(0.2/Pr) - 2}{\left(\frac{\epsilon_m}{\nu}\right)^{0.9}} \quad (8)$$

The results for Nusselt number, Nu, are correlated by

$$Nu = f(p/d, \bar{\psi} Pe)$$

Caution should be exercised in using Equation 8: it can give negative values for ψ in some instances. Figure 3.26 a and b shows a comparison of experimental data with analytical predictions of Dwyer for bundles with two different p/d ratios. The references to the data are given in section EN3 of the appendix. The Russian data appeared to fall below predictions which led Dwyer to believe that the data suffered from the presence of added resistance to heat transfer. Figure 3.27 shows a comparison of the previous Dwyer predictions with data taken at Brookhaven in a rod array with p/d ~ 1.75 . Dwyer also discusses circumferential variation of rod temperature, effects of rod displacement on (reduction in) Nusselt number and the problems of wetting, oxides and inert gasses which affect experimental data. The following section will discuss the state of the art in these areas. First a brief description is given of the eddy diffusivity models which are currently being used for predicting $\psi(r)$ and $\psi(\phi)$ in rod bundles.

FIGURE 3.26a. Comparison of Analytical Predictions with Experimental Results for In-Line Flow of Liquid Metals Through Unbaffled Rod Bundles Under Conditions of Uniform Heat Flux and Fully Developed Velocity and Temperature Profiles. (Dwyer 1969, survey)

FIGURE 3.26b. Comparison of Analytical Predictions with Experimental Results for In-Line Flow of Liquid Metals Through Unbaffled Rod Bundles Under Conditions of Nominally Uniform Wall Heat Flux and Fully Developed Velocity and Temperature Profiles. (Dwyer 1969, survey)

FIGURE 3.27. Experimental Heat Transfer Results for Liquid Metals Flowing In-Line Through Unbaffled Rod Bundles Under Conditions of (a) Uniform Wall Heat Flux, (b) Fully Developed Velocity and Temperature Profiles, and (c) Absence of Side-Wall Effects. Comparison of Results with Theoretical Predictions (Dwyer 1969, survey)

In turbulent flows the momentum and heat fluxes incorporate transport contributions not only due to viscous effects but also due to turbulent velocity fluctuations (and perhaps secondary flow, which is not considered here). This is expressed by the equations (following Nijssing 1973, N3)

$$\tau_r = -\rho \nu \frac{\partial u}{\partial r} + \overline{\rho u'v_r'} \quad (9)$$

$$\tau_\phi = -\rho \nu \frac{\partial u}{r \partial \phi} + \overline{\rho u'v_\phi'} \quad (10)$$

where ρ and ν are the fluid density and kinematic viscosity, u is the time-average axial velocity, and u' , v_r' , and v_ϕ' are fluctuating components of u , v_r and v_ϕ . The turbulent stress terms $u'v_r'$ and $u'v_\phi'$, representing one-point correlations of mutually perpendicular velocity fluctuations, have a value

different from zero since the turbulent field is non-homogeneous. In a similar manner, the heat fluxes q_r'' and q_ϕ'' can be expressed as

$$q_r'' = -c_p \rho \alpha \frac{\partial T}{\partial r} + c_p \rho \overline{T'v_r'} \quad (11)$$

$$q_\phi'' = -c_p \rho \alpha \frac{\partial T}{r \partial \phi} + c_p \rho \overline{T'v_\phi'} \quad (12)$$

where T is the time-average temperature and T' is the fluctuating component. One of the important areas of work in turbulent heat transfer is the development of the relationships between the turbulent fluxes and the mean flow properties. The complexity of the flow situation in rod-bundle geometries leads to more stringent demands on the knowledge of turbulent transport characteristics than required for the description of turbulent flow in simple geometries, such as circular duct. The various models and methods used for determining the turbulent transport properties have been described in section 3.2.1 on local hydrodynamics. Here the eddy diffusivity method is further discussed since it is most commonly used. The radial eddy diffusivity of momentum and heat may be expressed as:

$$\overline{u'v_r'} = \epsilon_{m,r} \frac{du}{dr} \quad (13)$$

$$-\overline{T'v_r'} = \epsilon_{h,r} \frac{dT}{dr} \quad (14)$$

One of the areas of significant research has been to determine expressions for variation of eddy diffusivities $\epsilon_{m,r}$, $\epsilon_{m,\phi}$ and $\epsilon_{h,r}$, $\epsilon_{h,\phi}$ within rod bundle channels. Nijssing (1972, IAEA; 1973, N3) gives a detailed survey of the turbulent transport phenomena. Here Nijssing's results are presented without discussions.

Nijssing adopted the following expression for the turbulent diffusivity for momentum transfer in the radial direction:

$$\epsilon_{m \cdot r} = y_0 \frac{\kappa}{2(1+b)} [b + x^2] [1 - x^2] \cdot \frac{2K - X(K-1)}{(K+1)K - X(K-1)} \left(\frac{\tau_w}{\rho} \right)^{1/2} \quad (15)$$

where y_0 = distance between wall and zone boundary

R = rod radius

d_e = equivalent hydraulic diameter

$X = 1 - y/y_0$

(and $\kappa = 1$ (for channels without rods, whereas for flow zones pertaining to a rod, $\kappa = 1 + y_0/R$) (In a similar manner $\kappa = 0.407$ for wall flow zones and otherwise

$$\kappa = 0.407 + 0.02y_0/R \quad (16)$$

whereas $b = 0.55$ (wall flow zone) or otherwise

$$b = [1.82 - 1.3 (y_0/R)] \left\{ 1 + 0.14 (\log Re_\ell - 5) \right\}^{-1} \quad (17)$$

where Re_ℓ is a local Reynolds number at the peripheral position considered. Values of $\epsilon_{m \cdot r}$ predicted by Equation 15 are 10 to 15% higher than given by Dwyer's equation (Equation 8). The expression adopted for the turbulent momentum diffusivity in the circumferential direction is:

$$\epsilon_{m \cdot \phi} = 0.154y_0 \left(\frac{\tau_w}{\rho} \right)^{1/2} \quad (18)$$

For the diffusivity ratios ψ_r and ψ_ϕ relating the turbulent heat diffusivities $\epsilon_{h \cdot r}$ and $\epsilon_{h \cdot \phi}$ to the corresponding momentum diffusivities, the following expressions have been adopted:

$$\frac{\epsilon_{h,r}}{\epsilon_{m,r}} \equiv \psi_r = 1.45 (u/u_{\max})^{1/2} \left[1 - \exp(-1.24 \times 10^{-4} \frac{y_0}{d_e} \text{Re} \sqrt[3]{\text{Pr}}) \right] \quad (19)$$

$$\psi_\phi = 1.45 (u/u_{\max})^{1/2} \left[1 - \exp(-2.48 \times 10^{-4} \frac{y_0}{d_e} \text{Re} \sqrt[3]{\text{Pr}}) \right] \quad (20)$$

Here $u_{\max}$ is the value of u at the zone boundary. Worth noting is that in the viscous region ($y^+ < 45$), $\epsilon_{h,r}$ is set equal to zero for coolant with $\text{Pr} \ll 1$.

The circumferential eddy diffusivity ($\epsilon_{m,\phi}$), is not equal to the radial eddy diffusivity, $\epsilon_{m,r}$ (anisotropy of turbulence), especially for wall channels and for bundles with low p/d ratios. Nijssing generally underestimates the anisotropy factor $\frac{\epsilon_{m,\phi}}{\epsilon_{m,r}}$ for wall channels and bundles with low p/d ratios. Recent data by Rehme (1978, N1) show that the momentum anisotropy factor for wall channels can be as large as 200 for bundles with low p/d and decreases with increasing p/d . No evidence of secondary flow was found by Rehme in the wall channels and some evidence in corner channels. Since Nijssing's (1973, N3) equation underestimates the anisotropy factor for low p/d ratios, this may have necessitated the introduction of a secondary flow (Figure 3.28) to match the data of Eifler and Nijssing (1967, N3). In Figure 3.29, is the ratio of the local mean coolant velocity at a position to the bulk coolant velocity u_b for a triangular rod array of $p/d = 1.05$.

Other models for predicting turbulent transport properties have been proposed by Ramm et al. (1975, N3) and Sha and Launder (1979, N3) for local transport properties. Sha and Carelli (1975, N4) have determined channel averaged values for $\epsilon_{m,r}$ and $\epsilon_{m,\phi}$ for use in a global (lumped or subchannel) approach based on Equations 15 to 20 originally developed by Nijssing (1973, N3).

FIGURE 3.28. Characteristic Domain of Infinite Rod Array

FIGURE 3.29. Circumferential Distribution of Dimensionless Velocity

The average turbulent eddy diffusivities over a subchannel for momentum transfer in the radial ($\epsilon_{M,r}$) and circumferential ($\epsilon_{M,\phi}$) directions are expressed as follows:

$$\overline{\epsilon_{M,r}} = Y_o \frac{\kappa b}{(1+b)} \left(\frac{1}{C+1} \right) \left(\frac{\tau_w}{\rho} \right)^{1/2} \quad (21)$$

$$\overline{\epsilon_{M,\phi}} = 0.154 Y_o \left(\frac{\tau_w}{\rho} \right)^{1/2} \quad (22)$$

where

Y_o = distance from wall to appropriate centroid of the channel under consideration

$\kappa = 0.407 + 0.02 Y_o/R$

R = rod radius

$C = 1$ for channels without a fuel rod

$= 1 + Y_o/R$ for channels with fuel rod

$b = \left[1.82 - 1.3 (Y_o/R) \right] \left[1 + 0.14 (\log Re - 5) \right]^{-1}$

ρ = coolant density

τ_w = wall shear stress: $0.03955 \rho w^{7/4} \nu^{1/4} De^{-1/4}$

w = axial coolant velocity

ν = viscosity

De = equivalent hydraulic diameter.

The ratios $\overline{\psi_r} \equiv \overline{\epsilon_{h,r}}/\overline{\epsilon_{m,r}}$ and $\overline{\psi_\phi} = \overline{\psi_h} / \overline{\epsilon_{m,\phi}}$ given by Equations 19 and 20 were modified to yield better agreement with experimental data. The following correlations were proposed by Sha and Carelli (1975, N3).

$$\overline{\psi_r} = 2.2 \left[1 - \exp \left(-1.24 \times 10^{-2} \frac{Y_o}{D_e} Re^{1/2} Pr^{1/3} \right) \right] \quad (23)$$

$$\overline{\psi_\phi} = 2.2 \left[1 - \exp \left(-2.48 \times 10^{-2} \frac{Y_o}{D_e} Re^{1/2} Pr^{1/3} \right) \right] \quad (24)$$

A number of computational methods have been developed for predicting turbulent velocity and temperature fields in triangular array of rods. These include the graphical methods of Dwyer (1966, N3), the approximate analytical method of Nijssing et al. (1969 and 1973, N3), Johannsen^(a) et al. (1971, 03), Axford (1967, N3), and Magee and Tromel (1966, N3), the numerical methods of Dwyer and Berry (1972, N3), Dwyer, Berry, and Hlavac (1972, N3), Bender and Magee (1969, N3), and Meyder (1974, N3), and the finite element methods of Slagter (1975, 1978, N3).

Among the numerical methods, Slagter's is the only method for multicell (or multirod) and analysis, but like all other methods, it does not account for the presence of spacers.

Using Equations 13 and 14, one can write the steady-state momentum and energy balances for a cylindrical shell of fluid (in the (r, ϕ) coordinate system) as follows:

$$\frac{1}{\alpha} \frac{\partial}{\partial r} \left[\rho r (\nu + \epsilon_{m,r}) \frac{\partial u}{\partial r} \right] + \frac{1}{r^2} \frac{\partial}{\partial \phi} \left[\rho (v + \epsilon_{m,\phi}) \frac{\partial u}{\partial \phi} \right] = -\frac{4}{de} \tau_{w,av.} = \frac{\partial P}{\partial z} \quad (25)$$

$$\text{and} \quad \frac{1}{r} \frac{\partial}{\partial r} \left[\rho r (\alpha + \epsilon_{h,r}) \frac{\partial T}{\partial r} \right] + \frac{1}{r^2} \frac{\partial}{\partial \phi} \left[\rho (\alpha + \epsilon_{h,\phi}) \frac{\partial T}{\partial \phi} \right] = \frac{4}{de} q''_{w,av.} \frac{u}{u_b} = \rho u \frac{\partial T}{\partial z} \quad (26)$$

Buleev et al. (1964, N3) made the assumption of directionally independent turbulent-diffusivity and adopted identical expressions for ϵ_h and ϵ_m derived theoretically for a circular tube on the basis of mixing length arguments and solved the above equations in a finite-difference form. Oberjohn and Bender and Magee treat the problem in finite difference without using the above equations as the starting-point. Their method is based on momentum and heat balances set up for macroscopic fluid elements $r\Delta r \Delta \phi$ (Figure 3.30).

(a) This analysis treats not only central regions but wall regions also.

FIGURE 3.30. Discretization of Coolant Region in Finite Difference Approach

This, together with assumptions regarding the type of velocity and temperature variation between fluid elements, leads to a finite-difference description in terms of velocities (temperatures) which represent the integrated-average values over a fluid element. As Nijssing (1973, N3) indicates, this method has the following advantages: "(a) it ensures that the conservation equations are satisfied over any part of the coolant region and (b) it provides the possibility of using different types and sizes of elements. The finite-difference procedure is somewhat problematic in the wall region, characterized by sharp radial velocity and temperature variation, in view of the circumferential variation of the viscous layer thickness."

Nijssing (1973, N3) extends a previous method, Nijssing (1967, N3) to turbulent flows where, in addition to conduction, account is taken of turbulent diffusion and secondary flow in the coolant. Results from Nijssing's (1973, N3) analyses are presented below as described by Nijssing.

"Figure 3.31 shows the variation of Nusselt number with rod spacing (p/d), for a triangular array and given Reynolds and Prandtl number. Two limiting thermal boundary conditions are considered, corresponding to the case of a uniform heat flux distribution and that of a uniform temperature distribution at the outer cladding surface. For comparison, theoretical results for the "equivalent annulus" and for the case of slug flow, with uniform heat flux at the outer cladding surface are also shown."

As also discussed by Dwyer (1966, N3), the rod assembly heat-transfer predictions always are below the "equivalent-annulus" predictions until p/d equals 1.15 when the difference is less than 10%. As indicated earlier at high Peclet numbers, the slug-flow analysis underpredicts the Nusselt number whereas at low Peclet numbers it will overpredict it. Figure 3.32 shows the effect of turbulent transport properties on circumferential outer cladding temperature expressed in terms of the dimensionless variable $k_f \Delta T_{o,c} / q'$, where k_f is the coolant thermal conductivity, $\Delta T_{o,c}$ is the difference

FIGURE 3.31. Variation of Nusselt Number with Rod Spacing

FIGURE 3.32. The Effect of Turbulent Heat Transport on Circumferential Cladding Temperature Variation

between the local and average outer cladding temperature and q' the heat generation rate per unit rod length. This figure from Nijssing (1973, N3) needs a detailed comment. Nijssing attributes the small effect of secondary flow on circumferential heat transport in the coolant to the small magnitude of the radial temperature variation in the coolant; i.e., in the limit if there was zero radial coolant temperature variation, the contribution of secondary flow to circumferential heat transport must be obviously zero. However, recent data by Rehme (see Hoffmann et al. 1979, IAEA) shows that for low p/d ratio bundles, the turbulence anisotropy factor ϵ_ϕ/ϵ_r can be as high as 200. In fact Nijssing had introduced his "single-loop" secondary flow model to explain (Figure 3.29) the large circumferential transport of momentum. With Rehme's data it now appears that the circumferential transport of momentum (and possibly of heat) is predominantly due to the large anisotropy factor. However, this issue is still undecided at this time.

The effect of thermal boundary conditions on circumferential cladding temperature variations is shown in Figures 3.33. Figure 3.33a shows the amplitude of the circumferential outer cladding temperature variation in terms of the dimensionless variable $k_f \Delta T_{o,cl,m} / q'$ where $\Delta T_{o,cl,m}$ is the difference between outer cladding maximum and average temperature. Two boundary conditions are considered, corresponding to uniform heat fluxes at the outer cladding surface and at the inner cladding surface, respectively. The figure shows that circumferential conduction in the cladding has a significant favorable effect on cladding temperature variations for close rod spacings. At rod spacings beyond 1.15 this effect is negligible. Figure 3.33b also shows the attenuation of the cladding temperature variation at low p/d ratios due to circumferential conduction in the various rod materials.

The aforementioned distributed-parameter analysis methods require excessive computational time and storage; as a result, efforts have been made to reduce these. Meyder (1974, N3) introduced a curvilinear orthogonal coordinate approach which reduces the necessary nodes for internal cells by a factor

FIGURE 3.33a. Amplitude of Circumferential Cladding Temperature Variation as a Function of Rod Spacing

of 10. However, the generation of the orthogonal net may create some difficulties in the vicinity of the bundle walls and at cell interfaces. In order to bridge the gap between the large number of cells required per subchannel in the distributed-parameter methods and the single-node sub-channel analysis method, Eifler and Nijssing (1978, 03) have introduced a sub-subchannel analysis method which would reduce the number of cells considerably and can be useful for interpreting local measurements in experiments. The number of cells used in the analysis can be varied easily to test the influence of discretization on the results. The major disadvantage of the method is the difficulty in describing the turbulent momentum transport parameter. However, the method appears to be promising and ought to be pursued further.

FIGURE 3.33b. Effect of Boundary Conditions on Circumferential Variation of Outer Cladding Temperature

Mixed-Convection Studies - Bare Rod Bundles

There are relatively very few methods that have been developed for predicting detailed velocity and temperature distributions within subchannels in either triangular or square rod arrays in mixed (forced + natural) convection. Tsai et al. (1973, N3) developed a rectangular strip model to simulate the buoyancy effects on the velocity and temperature distributions in the radial blanket assembly. The frictional drag due to the presence of the rods was neglected, as a result of which, much higher buoyance-induced flow redistribution was predicted. Igbal (1970, N3) considered the internal cell with boundary conditions of the first and second kind around the circumference of the outside cladding surface for a large range of Rayleigh number. In Davis and Perona's work (1973, E03) the hexagonal geometry was approximated by an equivalent annulus. This approximation is reasonable for large p/d values, which were studied analytically and experimentally by the authors. Similar analyses can also be found in Axford's work (1965, 03). Kim and Wolf (1977, 03) extended the analysis to include the clad region and simulated the power tilting effect by varying the heat flux at the cladding inside surface. They obtained the coolant shear stress distribution in the mixed convection regime, and they are attempting to obtain mixed convection friction factors for lumped-parameter (global) analysis. Figure 3.34 shows the dimensionless 2-D velocity and temperature distributions in a corner cell. The upper part depicts that regimes of near-stagnant and even reversed coolant flow exist. The associated coolant and cladding temperature fields are shown in the lower part of Figure 3.34. Figure 3.34 also shows that, although temperature gradients are more equalized because of buoyancy, there exists a critical Rayleigh number beyond which a sharp decrease in $\bar{Nu}$ occurs - even below its laminar value. The designer should avoid this regime by carefully controlling the power to flow ratio.

Experimental Studies - Bare Rod Bundles (Forced- and Mixed-Convection Regimes)

FIGURE 3.34. Isovels and Isotherms in a Corner Cell (left) and Average Nusselt Number Versus Rayleigh Number in Edge Cells (right)

There are very few experimental studies on local cladding/coolant temperature measurements in rod bundles using liquid metal coolants. These include the experimental data shown in Figures 3.26 and 3.27 - Borishansky and Firsova (1963 EN3), Subbotin (1964 EN3), Hlavac et al. (1968 E03), Kalish and Dwyer (1967 EN3), and Maresca and Dwyer (1964 EN3). In addition, cladding temperature measurements were made by Subbotin et al. (1971, EN3) for all types of cells in hexagonal bundle for Peclet numbers as low as 20. Of great interest is the recent three-dimensional cladding temperature measurements in edge and corner fuel rods of a sodium-cooled, bare 19-rod bundle by Moller and Tschoke (1978, EN3) which shows that temperature variations are greater than those even measured by Subbotin (1971 EN3). Figure 3.35 shows the temperature distribution at the circumference of the side and corner rod. Comparison with

FIGURE 3.35a. Temperature Distribution at the Circumference of the Side Rod

global calculations are shown in Figure 3.36, which clearly demonstrates the shortcomings of the global method in attempting to predict local temperature distributions, especially in the vicinity of the wall. Möller et al. (1978, EN3) have compared the temperature measurements to existing methods for local temperature calculations. As described before, most methods, except Slagter's, have been developed for single cells with simplifications of models and boundary conditions. Slagter's method for turbulent flow and Wong and Wolf's

FIGURE 3.35b. Temperature Distribution at the Circumference of the Corner Rod

(1978, N3) method for slug flow do take into account the coupling of cells. Figure 3.37 shows a comparison of Wong's method with the data of Möller. These are three-dimensional, two-cell slug-flow computations. The circumferential maximum temperature difference at the circumference of the side rod has been plotted as the heated length for four Peclet numbers. The comparison shows that circumferential temperature differences are calculated too low at the bundle inlet zone and differences are calculated too high for the major

- EXPERIMENT $TE A[]$, CORRECTED TO CLADDING SURFACE
- MEAN SUBCHANNEL TEMPERATURE (ARTIS-CODE)
- MEAN ROD SURFACE TEMPERATURE $Nu = 12$ ① ② ③
- MEAN ROD SURFACE TEMPERATURE $\begin{cases} Nu = 7 & \text{SIDE CHANNEL ①} \\ Nu = 14 & \text{CENTRAL CHANNEL ① ② ③} \end{cases}$

FIGURE 3.36. Experiment / SUBCHANNEL Calculations

FIGURE 3.37. Experiment/Two-Cell Calculation

part of the bundle length, especially at the end of the core. This is not unexpected, because the slug flow neglects turbulent momentum and energy transport effects between channels.

In the mixed-convection regime the experimental study of Dutton and Welty (1975 E04) and of Quigley et al. (1977 E04) for square rod arrays are of interest. The results of Quigley et al. show that significant flow redistribution occurs when a power skew across the bundle is superimposed on a relatively low-flow regime ($Re \sim 900-1200$). Local velocity distributions clearly show a flattening of the rod array velocity profiles due to the buoyant effects. An interesting phenomena observed in the experiments was the presence of thermal plumes in a predominantly laminar flow field, which grew in the lateral direction and then interacted with each other. The presence of turbulent thermal plumes complicates the flow and temperature field considerably.

FIGURE 3.38. Effect of Wire Wrap Spacer on Cladding Mid-wall Temperature Variations at Different Angular Positions in a Fuel Element

Bundles with Spacers

As mentioned in the section on local hydrodynamics, the only published method for predicting fully developed local velocity distributions in wire-wrapped bundles is due to Chuang et al. (1977, N3), which is described below.

Figure 3.6 shows the isolated unit cell selected by Chuang for determining detailed velocity and temperature fields. Chuang closely follows the method of Buleev (1964, N3) described earlier in this section solving the detailed two-dimensional coolant momentum and energy equations. These equations are identical to those used by Buleev, Equations 25 and 26. Buleev's eddy diffusivity of momentum model is also used by Chuang et al. First the bare-rod velocity profile is determined. Then the results are used in the coolant energy equation (which is coupled to the energy equation for the cladding, wire wrap, and fuel) to solve for the temperature fields. The effect of the wire wrap on turbulent transport of momentum and heat has not been included in the analysis. This may lend to conservative predictions of circumferential

cladding temperature gradients. The axial pressure gradient is determined by using Novendstern's correlation (1972, EN2) which includes presence of wire wrap. The method is capable of treating power skew across the rod pellet and capable of calculating the rod temperature field under eccentric pellet conditions. A quasi-three-dimensional temperature and velocity field can be obtained when Chuang's method is used in conjunction with a subchannel analysis code. Further development of this work for extension to three dimensions by L. Wolf has been initiated.^(a) Figure 3.38 (from Chuang et al.) shows the increase in the variation of the circumferential cladding temperature due to the presence of spacer wire as compared to the bare-rod bundle.

The presence of wire wrap makes a subchannel analysis code and a code for local thermal-hydraulic predictions very difficult to couple: it is very difficult to define an isolated unit cell in the same sense as was done for bare rod bundles because of the significant cross flow caused by the presence of wire wrap. In fact, the boundary condition of an adiabatic, no-shear surface is far from real because the transverse flow components and their axial derivatives are relatively large.

An iterative method of solution, where the boundary conditions are assumed at the start of the solution, will not work as it will give multiple solutions unless the presence of the neighboring channels is taken into account. It seems that for wire-wrapped assemblies one may have to resort to a multicell solution in order to describe the flow field. Another drawback of the method is that Buleev's model for determining radial and circumferential eddy diffusivities may not be strictly applicable to wire-wrapped assemblies. However, the presence of wire wrap may increase the magnitude of radial and circumferential diffusion of heat, so the method is expected to yield conservative results for circumferential cladding temperature gradients. Work in this area is needed, but the difficulties of obtaining reasonable, realistic solutions are enormous - in fact, there is not a single distributed-parameter method yet developed which can include the effect of either grid spacers or wire wrap adequately.

(a) The author received this information in personal communications from L. Wolf.

Experimental data for local cladding or coolant temperature distribution in assemblies with wire wrap spacers is not available and is needed.

Recommended Design Correlation for Nusselt Number for Wrapped Rod Bundle

Kazim and Carelli (Tang et al. 1978, Book) recommend the following empirical correlations for Nusselt number for design purposes which have been compiled on a large number of experimental data. Tang mentions that experimental evidence indicates that the presence of gas and less-than-perfect wetting in the lower Peclet number range can result in a low heat transfer coefficient being measured, even under supposedly well-controlled experimental conditions.

The recommended correlations for rod bundle heat transfer are for $1.15 \leq p/d \leq 1.3$; $10 \leq Pe \leq 5000$

$$Nu = 4.0 + 0.33 (p/d)^{3.8} (Pe/100)^{0.86} + 0.16 (p/d)^{5.0} \quad (27)$$

for $1.0 \leq p/d \leq 1.15$; $150 \leq Pe \leq 1000$

$$Nu = [-16.15 + 24.96 (p/d) - 8.55 (p/d)^2] (Pe)^{0.3} \quad (28)$$

for $Pe \geq 150$

$$Nu = 4.50 [-16.15 + 24.96 (p/d) - 8.55 (p/d)^2] \quad (29)$$

These correlations are expected to be reasonably conservative in the design range.

Three-Dimensional Solutions

Extremely few three-dimensional solutions are available to describe the entrance region developing hydrodynamic and temperature fields in rod bundles. Meier (1974, N3) obtained the solution of the heat transfer problem in the various unit cells of a hexagonal bundle for slug flow. The method is a single-region analysis with a constant heat flux imposed around the cladding outside surface. The analysis accounts for an arbitrarily and axially varying

heat flux distribution by using Duhamel's theorem. (Also see Minkowycz and Stein 1973, N4). The solution technique relied on an improved point-matching procedure, using an orthonormalization method for solving the overdetermined set of equations. Nijssing (1973, N3) also treats the development of the three-dimensional thermal field in a bundle. (Also see Hofmann 1973, 03 in the next section). Figure 3.39 shows that the circumferential cladding temperature variation continuously develops axially from inlet. Recently the three-dimensional problem was further extended by Wong (1978, N3) from a single-cell to a multicell slug-flow analysis. Entrance region local solutions for turbulent flow and temperature fields in rod bundles are needed, especially in the presence of spacers.

Effect of Rod Displacement

A number of methods have been developed for predicting local coolant and cladding temperature distributions and Nusselt numbers for the situation of a rod asymmetrically located in an otherwise uniform triangular array in the absence of spacers. These include the work of Dwyer et al. (1970, 03),

FIGURE 3.39. Axial Variation of Cladding Temperature Difference of Wall Adjacent Fuel Rod

Hishida (1974, 03), Hofmann (1973, 03), Hsu (1972, 03), Johannsen and Wolf (1971, 03), and Chen (1973, 03). The methods developed are of interest as a limiting case for analysis of geometrically distorted bundles.

Figure 3.40 shows a general multiregion model for displaced central, corner, and side fuel elements proposed by Johannsen and Wolf who performed a fully developed slug-flow analysis for the isolated channels. The multiregion analysis required the simultaneous solution of the energy equations for fuel, cladding, and coolant, which are coupled by the local temperature and heat flux conditions existing at the fuel-cladding and cladding-coolant interface, respectively. The adiabatic boundary condition applied at the outer contour of the general coolant region (Figure 3.40) was satisfied in the r - ϕ coordinates by point-matching technique. The analytic solution is obtained in terms of Fourier series. Numerical calculations performed for a number of parameters showed that the amplitude of circumferential temperature variations is most sensitive to effects resulting from changes in geometry. This is especially true for the asymmetric central element where even small displacements from the symmetric position result in large cladding temperature variations as shown in Figure 3.41. The multiregion analysis, if restricted to the coolant region (which assumes constant heat flux at the rod surface) would be quite erroneous. Johannsen and Wolf do not take into account the temperature dependence for the physical properties of fuel, cladding, and coolant. Because of these restrictions, the range of application of Johannsen's analysis is limited to the Peclet number range $500 \leq Pe \leq 1500$, and to bundle geometries with $p/d \leq 1.25$.

Hishida's work (1974, 03) is also for fully developed velocity and temperature fields. However, Hofmann (1973, 03) indicates that the assumptions of the fully developed thermal field do not hold good for asymmetrical bundle geometries and for wall and corner positions for normal designs. The hydraulic field is probably fully developed due to the presence of a lower axial blanket prior to the active fuel. Hofmann (1973, 03) does not make a slug-flow assumption, instead the turbulent flow field is determined in the (X, ϕ) geometry, i.e. radially lumped geometry, using Karman's velocity distribution.

FIGURE 3.40. General Multiregion Model for Displaced Central, Corner and Side Fuel Elements

The turbulent momentum and energy exchange in the azimuthal direction is taken into account by an eddy diffusivity model due to Nijssing et al. (1967, N3). Dwyer's equation (Equation 8) is used to determine the ratio $\epsilon_{H\phi}/\epsilon_{M\phi} (\equiv \psi)$.

The results show that with sodium coolant even minor rod displacements can produce major velocity and temperature variations at the rod circumference which rise severely with eccentricity. At the same time, the absolute maximum values of the circumferential temperature differences are displaced axially towards the end of the fuel element; i.e., to zones where the nominal design temperatures of the coolant and in particular the cladding are already high. The circumferential temperature variation is damped in the fuel much more than in the cladding.

Hofmann's method has a number of limitations although the results are of qualitative significance. Hofmann cannot treat the presence of secondary flows, and the mathematical simplifications made in the analysis conflict with boundary conditions at the symmetry boundaries as discussed by Nijssing (1973, N3). Another limitation is that the assumption of a fully developed flow field is an approximation for bundles with significant displacements or asymmetries.

FIGURE 3.41. Circumferential Variation of Temperature at Outer Cladding Surface and at Outer Contour of Coolant Region for a Symmetric Central and for Asymmetric Elements of Different Displacement

More work is required for developing flow and thermal fields with rod displacement, especially because the analysis of geometrically distorted bundles is of great significance.

Application of Local Hydrothermal Studies for Evaluating Input Coefficients for Global Analysis

A number of two-dimensional (fully developed) methods for predicting local flow and temperature fields have been developed, as discussed in previous sections. The computation of detailed three-dimensional coolant temperature

fields for realistic reactor rod bundles on a single-cell basis and especially on a multicell basis is not feasible with present-day computers because of the enormous computation and storage requirements. Moreover, the effect of the presence of spacers on local transport characteristics in the coolant is not known with sufficient accuracy. These difficulties have resulted in the development of simplified global methods for heat transfer predictions of large rod bundles. The most commonly used global method is the method of subchannel analysis, where a subchannel is the area enclosed by three adjacent rods or four adjacent rods, depending on whether it is a triangular or square rod array. Within a subchannel the hydrodynamic and thermal coolant conditions are characterized by bulk average values. This not only simplifies the mathematical analysis but allows the calculations to be performed with a knowledge of the global coefficients for transport mechanisms. For bare rod bundles, cross-flow resistance and the mixing coefficient are the two global parameters of interest required to describe momentum and energy transport between sub-channels. Bundles with spacers require additional empirical constants as input, which is described later in the next section.

In subchannel analysis the use of a single average coolant velocity and temperature in a channel amounts to integration of the governing equations over the channel. This reduces the order of the differential equations. If the higher-order derivatives are suppressed, the no-slip and heat flux boundary conditions at the surfaces cannot be satisfied. Thus, in regions where coolant temperatures vary significantly, subchannel analysis cannot perform satisfactorily. Because of the simplifications in subchannel analysis, a number of further assumptions have to be made in describing thermal interactions between adjacent coolant elements. These include the following:

- The local coolant temperature (enthalpy) at the center of the channel equals the average bulk coolant temperature (enthalpy). The temperature (enthalpy) gradient at the interface of two channels (which is proportional to the heat transferred between the channels) is equal to the ratio of the temperature difference between the centroids of the adjacent channels divided by the centroid-to-centroid distance.

- Along the boundaries between adjacent channels, the distribution of turbulent property values of temperatures (enthalpy) and also temperature (enthalpy) gradients normal to the boundaries is uniform.

For bare rod bundles, it is assumed that molecular diffusion and turbulent diffusion represent the sole mechanisms of transport. For the case of heat transport, the approach generally followed is to use the following equations for interchannel heat flux:

$$q''_{ij} = \rho c_p (\epsilon_H + \alpha) \left(\frac{\partial T}{\partial Y} \right)_{ij} \quad (30)$$

where $\left(\frac{\partial T}{\partial Y} \right)_{ij}$ is the temperature gradient at the interface, ϵ_H is the global turbulent heat diffusivity, α is the thermal diffusivity, ρ is the density, and c_p is the specific heat. As described earlier, the average temperature gradient between channels is given by:

$$\left(\frac{dT}{dy} \right)_{ij} = \frac{T(i) - T(j)}{\eta_{ij}} \quad (31)$$

where $T(i)$ and $T(j)$ are the bulk coolant temperatures in subchannels i and j , respectively (as indicated in Figure 3.42), and η_{ij} is the centroid-to-centroid distance. In terms of enthalpy, if c_p is nearly uniform

$$dh = c_p dT \quad (32)$$

$$q''_{ij} = \rho (\epsilon_H + \alpha) \frac{h(i) - h(j)}{\eta_{ij}} \quad (33)$$

FIGURE 3.42. Cross Section of Hexagonal Fuel Rod Assembly

If a mixing coefficient $\beta(i,j)$ is defined as follows [$\beta(i,j)$ is replaced by β]

$$\beta = \frac{q''_{ij} \cdot S_{ij}}{[h(i) - h(j)]} \quad (34)$$

$$\beta = \frac{\rho(\epsilon_H + \alpha) S_{ij}}{\eta_{ij}} \quad (35)$$

and

$$q' = \beta [h(i) - h(j)] \quad (36)$$

where q' is the heat flux per unit axial length and S_{ij} is the length of the common boundary between channels.

The mixing coefficient, β , can be determined by rod bundle flow and heat transfer experiments where the predicted and experimentally measured temperature distributions are matched by adjusting β . Basic objections to the use of Equations 30 and 35 for describing transverse energy transport are that: Equation 30 assumes that a two-dimensional heat transfer between adjacent channels can be modeled by a one-dimensional form of the heat transport equation; Equation 35 assumes the mixing coefficient to be independent of thermal boundary conditions (although β is governed by geometrical configuration and spatial heat generation distribution). France and Ginsberg (1972, N3) and Yeung and Wolf (1977, N4) have sought to find a correction factor in the form of an effective mixing length for conduction heat transfer between two adjacent channels as follows:

Equation 33 can be rewritten as

$$q' = k \frac{[T(i) - T(j)]}{\eta_{ij}^*} \cdot S_{ij} \quad (37)$$

$$q' = k \frac{[T(i) - T(j)]}{\eta_{ij}} \frac{S_{ij}}{N_{ij}} \quad (38)$$

where η_{ij}^* is the effective conduction mixing length, and N_{ij} is dimensionless correction factor, which is unity if $\eta_{ij}^* = \eta_{ij}$. Equation 38 can be rewritten as

$$N_{ij} = \frac{k S_{ij}}{i_j} \frac{[T(i) - T(j)]}{\int_{S_{ij}} q_{ij}''(s) ds} \quad (39)$$

where $q_{ij}''(s)$ is the heat flux distribution along the common boundary in the coolant between subchannels i and j . If the local coolant temperature field is known from local hydrothermal methods, Equation 39 will yield an accurate value of N_{ij} since all quantities on the right hand side of Equation 39 can

FIGURE 3.43. Dimensionless Effective Conduction Mixing Lengths Versus Wall Distance in the Range of Interest

be explicitly evaluated. Figure 3.43 shows N_{ij} as a function of the dimensionless wall distance for the corner region of a hexagonal bundle with $p/d = 1.20$. The results reveal that N_{ij} deviates quite substantially from unity. This has also been found by Ginsberg and France (1973, N3). Clearly, the centroid-to-centroid approximation to the effective conduction mixing length is highly questionable. Accounting for the analytically determined correction factor would reduce interchannel coolant temperature differences for typical LMFBR operating conditions. This illustrates another advantage of developing local hydrothermal analysis methods.

3.2.4 Hydrothermodynamics - Global Studies

In the previous section, which discusses the application of the local hydrothermal methods for evaluating input coefficients for global subchannel analysis, a few basic assumptions and approximations in the subchannel approach

were described. Momentum and energy exchange between channels can occur by a number of mixing mechanisms, discussed by Rogers and Todreas (1968, Survey). These include turbulent exchange (including flow scattering), diversion cross flow, and flow sweeping. In addition, energy exchange can occur by thermal conduction. In bare rod bundles, thermal conduction and turbulent exchange are the predominant modes of energy exchange. Diversion cross flow due to a pressure differential may occur because of dissimilarity in the hydrodynamic characteristics of two adjacent subchannels. In assemblies with spacer grids, the relative flow resistance between the center and edge or corner channels may change significantly between gridded and ungridded regions. This will induce diversion cross flow, which can have a profound influence on the transverse interchange of momentum between channels. In wire-wrapped bundles, the wire may impede the flow causing the flow to "sweep" along the local direction of the wire. Near the wall region the wire induces a flow parallel to the duct wall. This is known as swirl flow. Flow scattering can be considered an enhanced turbulent interchange caused by devices such as grid spacers, local projections, and bundle skirts. The various modes of mixing within a bundle interact with each other and the assumption of these effects being separable and superposable may introduce error (Khan et al. 1974, N4).

Bare Rod Bundles

For bare rod bundles, the predominant modes of energy exchange in the absence of dissimilar channels are by thermal conduction (for low-Pr fluids) and momentum and energy exchange by turbulent mixing. The turbulent mixing coefficient, β , was defined in Equation 34. Determining the coefficient β has been the subject of a number of analytical and experimental effort (Rogers and Todreas 1968; Kjellstrom, 1974; Nijssing 1972; all from Survey section). Correlations for β are discussed later in this section. Nijssing and Eifler (1974, N4) have analytically determined β (and a corresponding coefficient for momentum transfer) by taking into account the shape of the interacting subchannels and of the locally varying turbulent transport characteristic inside a given subchannel. The critical assumptions underlying this procedure are the position of bulk coolant temperature (enthalpy), which is taken at the subchannel centroid and the circumferential variation of heat flow (assumed to

be constant between the subchannel centroids). As discussed in the previous section, the mixing coefficients depend upon the thermal boundary conditions (power skew and geometry) and are at the origin of the limited physical validity of the mixing coefficient concept. If q'_{mom} and q'_{eny} are the inter-channel flow of momentum and energy per unit axial distance and if $U(i)$ and $H(i)$ denote the average coolant velocity and enthalpy in a channel, the momentum and heat transfer mixing coefficients can be defined as

$$\alpha(i,j) = \frac{q'_{\text{mom}}}{\rho[u(i) - u(j)]} \quad (40)$$

$$\beta(i,j) = \frac{q'_{\text{eny}}}{[H(i) - H(j)]} \quad (41)$$

Figure 3.44 shows the model for diffusive interaction used by Nijssing (1974, N4). Integration of differential expressions developed for q'_{mom} and q'_{eny} with respect to ϕ yields the following expression for $\alpha(i,j)$ and $\beta(i,j)$:

$$\{\alpha(i,j)\}^{-1} = \int_0^{\phi_e(i)} \frac{P_R(i,j)R}{2Y_\phi \epsilon_\phi} d\phi + \int_0^{\phi_e(j)} \frac{P_R(i,j)R}{2Y_\phi \epsilon_\phi} d\phi \quad (42)$$

$$\{\beta(i,j)\}^{-1} = \int_0^{\phi_e(i)} \frac{P_R(i,j)R}{2Y_\phi (\psi \epsilon_\phi + \alpha)} d\phi + \int_0^{\phi_e(j)} \frac{P_R(i,j)R}{2Y_\phi (\psi \epsilon_\phi + \alpha)} d\phi \quad (43)$$

where ϵ_ϕ is the angular-position-dependent radially averaged turbulent diffusivity for circumferential momentum transfer, ψ denotes the ratio of $\frac{\epsilon_{H\phi}}{\epsilon_{m\phi}}$ and Y_ϕ is a radial distance given for the triangular^(a) subchannels of Figure 3.44 by

(a) For wall channels $Y_\phi = \frac{(B + R \sin \phi) \tan \phi/2}{1 - \sin \phi \tan \phi/2}$; $B = (P_w R - \frac{R}{\cos \phi}) / \tan \phi$; and P_w is the distance from the wall to the center of the rod.

FIGURE 3.44. Discretization of the Coolant Regions Considered

$$Y_{\phi} = \frac{P_R(i,j)}{\cos \phi} - R \quad (44)$$

where $P_R(i,j)$ is the pitch-to-diameter ratio of the rods separating channels i and j , and R is the rod radius. The integration of Equations 42 and 43 are carried from centroid to centroid (Nijssing 1974, N4); i.e., $\phi_e(i) = \phi_c(i)$ and $\phi_e(j) = \phi_c(j)$. The computation of (i,j) and (i,j) require expressions for ϵ_{ϕ} and ψ . Nijssing recommends the following expression for ϵ_{ϕ} :

$$\epsilon_{\phi} = 0.154 Y_{\phi} (\tau_w / \rho)^{1/2} \quad (45)$$

where τ_w , the wall shear stress is assumed uniform inside a given channel. For $p/d \geq 1.15$ this assumption is justified (Nijssing 1973, N3). For liquid metal coolants ψ is significantly influenced by conduction effects. The current knowledge regarding to the effect of conduction on ψ has been summarized by Nijssing (1972, R2). The expression used for ψ is an expression modified by Bobkov (1972, N3):

$$\psi = 1.38 \left[1 - \exp \left\{ -12.4 \times 10^5 F_\alpha (Y_{av}/d_e) \text{RePr}^{1/3} \right\} \right] \quad (46)$$

where F_α is an anisotropic factor denoting the ratio of turbulent diffusivities for transport in the circumferential and radial directions, d_e is the subchannel hydraulic diameter, and Y_{av} is the circumferentially averaged Y_ϕ within a subchannel. For an infinite rod array

$$Y_{av} = \frac{d_e}{2 \left\{ 1 + (p/d) (0.6/\pi \tan \frac{\pi}{6})^{1/2} \right\}} \quad (47)$$

Table 3.8 (from Nijssing, 1974, N4) gives the detailed experimental conditions for which the expression in Table 3.9 were developed. Figure 3.45 shows a comparison of $\frac{\beta}{\mu}(I,J)$ obtained from the integration of Equation 43 against the experimental data. The spread in the data is not surprising as the mixing coefficient value is very sensitive to experimental errors. The most important factors affecting the accuracy of experimentally determined mixing coefficients are as discussed by Nijssing (1974, N4):

1. insufficient axial mixing distance for establishing "pseudo" equilibrium temperature (concentration) profiles
2. uncertainties in the subchannel mass flow rates employed for evaluating mixing coefficients
3. intersubchannel cross-flow phenomena caused by anomalous entrance conditions or by rod misalignment
4. enhanced intersubchannel transport at the start of the mixing section, due to extra turbulence introduced by tracer fluid injection or by the trailing edge of subchannel separators
5. incremental transport owing to heat conduction across the rods for the case of heat mixing.

TABLE 3.8. Experimental Data on Turbulent Intersubchannel Mixing in Rod Arrays

<u>Investigator^(a)</u>	<u>Mixing Technique</u>	<u>Channel Configuration</u>	<u>Pitch-to-diameter Ratio</u>	<u>Rod Diameter, mm</u>	<u>Axial Distance Involved in Mixing, mm</u>	<u>Remarks</u>
Rapier (1967, N4)	--	different inter-connected flow regions	1.12	41	6700	data here specified for single-gap channel Re 50,000
Walton et al. (1969, N4)	tracer injection	two adjacent triangular subchannels	1.27	8.2	457	
Voj et al. (1973, N4)	tracer injection	121-rod bundle with triangular array	1.27	8.2	525	Re = 10,000-20,000; spread of data $\pm$ 40%
Rowe and Angle (1969, N4)	tracer injection and heat input	adjacent triangular and rectangular	1.0355 1.15	14.3 14.3	1525	Re = 6600-320,000; spread of data $\pm$ 30%
Pietralla (1970, N4)	tracer injection	19-rod bundle with triangular array	1.20	15	1080	Re = 15,000-50,000; spread of data $\pm$ 30%
Rogers and Tarasuk (1968, N4)	tracer injection and heat input	Variety of bundles with triangular arrays	1.035	14		correlation based on experimental data of various investigators including those of Rowe and Angle (1969, N4)

(a) All references can be found in section N4 of the references--Appendix A.

TABLE 3.9 Correlations for $\beta(I,J)/\mu$ Based on Data

Investigator/Reference	Correlation
Rapier (1967, N4)	$\beta/\mu = 0.018 (p/d - 1) (R/de) Re^{0.875}$
Walton (1969, N4)	$= 0.041 S_{ij}/de Re^{0.783}$
Vaj et al. (1971, N4)	$= 0.0072 Re^{0.875}$
Rowe and Angle (1967, 1969, N4)	$= 0.0062 Re^{0.9}$
Pietralla (1970, N4)	$= 0.00845 Re^{0.857}$
Rogers and Tarasuk (1968, N4)	$= 0.0503 (p/d - 1)^{-0.57} Re^{0.68}$

FIGURE 3.45. Turbulent Intersubchannel Mixing Coefficients in a Triangular Rod Array, a Comparison with Experimental Data

Bundles with Spacers

The presence of spacer grids can significantly affect global hydrodynamics and therefore global hydrothermodynamics. The effect of the presence of spacer grids on the hydrodynamics is discussed in Section 3.2.2 on global hydrodynamics. Spacer grids enhance the effect of turbulent mixing between channels, the degree of enhancement depending upon the shape of the grid device. With increasing distance behind the spacer grid, there is a decay of turbulence toward an asymptotic value. Because they obstruct the axial coolant flow, spacer grids induce macroscopic interactions - diversion cross flow - between channels. This is especially true near the wall region where the grid resistance may vary from one subchannel to another, as a result of which lateral pressure differences develop, leading to intersubchannel cross-flow over relatively large axial distances.

The average mixing coefficient for a rod bundle with spacers is determined experimentally by varying β until the computational results match the measured temperature distribution at a specified axial location (refer to Figure 3.46). Measurements in heated prototypic bundles are the most reliable method of determining the effective mixing coefficient in the presence of spacer grids (Hoffman, Baumgartner and Frey 1977, EN4; Hoffman and Baumgartner 1974, EN4; Cornet et al. 1976, EN4; Moller et al. 1977, EN4).

A number of spacer-grid design have been investigated. These include the honeycomb spacer grid and the spark-eroded grid. The honeycomb grid is fabricated from steel straps adequately shaped and spot welded together. A typical honeycomb grid, used in SNR 300 Mk IA assembly (Weber and Cornet 1979, IAEA) is shown in Figure 3.47. Figure 3.48 shows a grid that is fabricated by spark erosion. The spacer grid may be positioned in a bundle by tie rods or by the so-called "skirts". These skirts are small steel plates (six for one grid) which are spot welded to the grid and to the wrapper tube. Skirts can have a favorable effect on the bundle thermal-hydraulics, especially for bundles in which the grid is located eccentrically. This favorable effect of the skirts is discussed in the section on non-nominal bundle performance.

FIGURE 3.46. Mean Subchannel Temperatures: Comparison Between Computed and Experimental Results

Table 3.10 and 3.11 list a number of state-of-the-art subchannel analysis computer programs being used for rod bundle thermal-hydraulic analysis. Table 3.12^(a) and Table 3.13 list codes currently being used in the U.S. nuclear industry in thermal-hydraulics, including a number of subchannel analysis computer programs. The subchannel analysis computer programs for bare-rod, spacer-gridded, and wire-wrapped bundles are similar in structure and are discussed next.

The existing methods of thermal analysis for nominal rod bundles can be classified into two categories for convenience in analysis: 1) the porousbody model and 2) the subchannel analysis. For bare-rod and grid-spaced bundles, the subchannel analysis has been used for a long time. For wire-wrapped

pin diameter	6.0 mm
pin pitch	7.9 mm
number of pins	166 +3
wrapper tube	
inner size	105.05 mm

FIGURE 3.47. SNR 300 Mk Ia Reference Element

pin diameter	6.0 mm
pin pitch	7.9 mm
number of pins	166 +3
wrapper tube	
inner size	105.05 mm

FIGURE 3.48. SNR 300 Mk Ia Sonder Element

TABLE 3.10. Computer Codes Presented at the IAEA Meeting
for Nominal Operation Conditions

(a) in development

TABLE 3.11. Table of LMFBR Core Subchannel and Local Thermal-Hydraulic Computer Codes (Grid-Spaced and Wire-Wrapped Bundles)

<u>Type of Code</u>	<u>U.S. Codes</u>	<u>French Codes</u>	<u>German Codes</u>	<u>British Codes</u>	<u>Comments</u>
Global (Subchannel)	ENERGY I,II,III COBRA-IV COTEC	TASNA FLICA	MISTRAL ARTIS THECA	SABRE HAMNA WIRCO	major British emphasis on grid-spaced bundles
Local (distributed- parameter)	VELASCO FATHOM-360	THESEE (finite element) HEATING-3D	VELASCO VERA-TERA TURBIT-3(a) VITESSE(a) (finite element)		German effort both on grid-spaced and wire- wrapped bundles
Transient Analysis	COBRA-IV CORTAN		TURBIT-3(a)		French and U.S. efforts on wire- wrapped bundles
Distorted Bundles Distortion Known Distortion Calculated	COBRA-IV	DISTAY FLICA(a)	IAMBIC/TRAMBIC(a)		
Mixed and Natural Convection	COBRA-WC CORTAN COMIX	THERNAT(a)	TURBIT-3(a) and NOTUNG		

(a) to be done or under development

**TABLE 3.12. Summary of U.S. Core Thermal-Hydraulic Computer Codes
(Due to R. A. Markley, WARD)**

Name	Originator	Purpose	Key Features	Status	Documentation
THI-3D	Argonne National Laboratory (ANL)	Coolant and rod temperature distribution in assembly	Solves momentum equations; has wire wrap model; detailed control assembly model	Operational; little experience	User's manual
TCLUST	ANL	Detailed whole-core coolant temperature distribution	Transient; no flow modeling; calibrated and verified	Operational; used in EBR-II analyses	In progress
NATCON	ANL	Plant transient analysis	Similar to DEMO, for EBR-II Design	Operational	Very limited
COMMIX	ANL	General thermal-hydraulics of rod bundle or other components	3-D, transient, single- and two-phase	Operational	User's manual
BODYFIT	ANL	General thermal-hydraulics of rod bundle or other components	Next generation to COMMIX; based purely on theoretical models; no calibration required	In development	In progress
TRITON	Westinghouse Advanced Reactor Division (ARD)	Detailed, core-wide duct, channel, and rod temperature distribution	Steady state; emphasizes duct temperature calculations; accounts for uncertainties; design oriented	Operational; used in CRBPR, FFTF design	User's manual; papers
COTECT	ARD	Coolant and cladding temperature distribution in assembly	Empirical code; fast running; extensively verified/calibrated; design oriented	Operating; extensive design experience (FFTF, CRBRP)	Unpublished user's manual
DEMO	ARD	Plant transient analysis	Models entire primary system; CRBRP design oriented	Operational	User's manual
OCTOPUS	ARD	Core orificing optimization	Core orificed to satisfy constraints on lifetime burnup, transients, upper internals, etc.; accounts for uncertainties; design oriented	Operational; extensive design experience (CRBRP, advanced LMFBRs)	User's manual; papers
CATFISH	ARD	Overall primary system hydraulics	Optimizes flow allocation and orifice sizing accounting for orificing requirements, pump characteristics; detailed P calculation	Operational	User's manual in progress; papers
FATHOM-360 tempera-	ARD/WRL Operational; used in		Detailed rod temperature distribution User's manual; articles ture distribution in subchannel and temperature in rod; solves momentum/energy equations for inboard channels; wire-wrapped rods	CRBRP design	Local 2-D velocity and

TABLE 3.12. (Cont'd)

Name	Originator	Purpose	Key Features	Status	Documentation
FATHOM-360S	ARD/WRL	Detailed rod temperature distribution	Same as FATHOM-360 for side channels; accounts for heat transfer through duct	Operational	User's manual in progress
CORINTH	ARD	Interassembly flow redistribution	Extended version of DYNAMIC FLODISC	Operational	User's manual
FLONET	ARD	Removable shield design	Detailed analysis of shield assemblies and interstitial gaps between core assemblies	Operational	User's manual
FLODISC	ARD	Interassembly flow redistribution	Steady state; simple, design-oriented; updated ΔP correlations	Operational	Limited
SSC	Brookhaven National Laboratory	Plant transient analysis	Generalized system code; detailed models of components	Development in progress	Papers
FORE-2M	GE/Advanced Reactor Division	Rod transient analysis	Point-kinetics and thermal-hydraulics; detailed nodalization; accounts for uncertainties	Operational; extensive design experience (FFTF, CRBR)	User's manual
FULMIX II	GE	Coolant and rod temperature distribution	Solves momentum equations, including wire wrap model	Operational; little experience	User's manual; papers
PACT II	GE	Probabilistic analysis of coolant and cladding temperatures	Thermal-hydraulics in coolant channels and Monte Carlo sampling of uncertainties in thermal-hydraulic calculations	Operational	User's manual; papers
CORTAC	GE	Transient ring-wise duct temperature for core restraint	Integrated thermal-hydraulic/nuclear reactivity feedback	Operational	User's manual; papers
DYNAMIC FLODISC	HEDL	Interassembly flow redistribution	Transient version of FLODISC	Operational	User's manual
CORA	HEDL	Detailed whole-core coolant temperature distribution	Transient; flow redistribution, models from energy	Operational; used in FFTF design	In progress
CONROD	HEDL	Control rod behavior	Thermal-hydraulics; structural analysis of control rods	Operational; used in FFTF design	Very limited
ENERGY	MIT	Coolant temperature distribution in assembly	Empirical; simple model; flexible to use and modify	Operational	User's manual; reports articles; papers
SUPERENERGY	MIT	Detailed, core-wide duct and channel temperature distribution	Steady state; fast running; simple model	Operational	Articles; User's manual
FLOW-3D	MIT	Detailed subchannel coolant distribution	Similar to FATHOM-360, but 3-D; only wire wraps; single channel	In development	In progress

TABLE 3.12. (Cont'd)

Name	Originator	Purpose	Key Features	Status	Documentation
COBRA-WC	PNL	Detailed whole-core coolant temperature distribution	Transient; flow redistribution; same features as COBRA-IV	Operational, development still in progress	In progress
CORTAN	PNL	Detailed whole-core coolant temperature distribution	Transient, flow redistribution; empirical models based on ENERGY	In development	In progress
COBRA-IV	PNL	Coolant temperature distribution in assembly	Transient capability; solves momentum equations; can be used outside calibration range; analyzes grid-spaced assemblies	Operational; extensive experience throughout U.S.	User's manual; several articles and papers
TEMPEST	PNL	3-D hydrodynamics	3-D version of VARR-II	In development	In progress
VARR-II	SAI	2-D hydrodynamics	General purpose code; for plena and internals design	Operational; used in CRBR design	User's manual
LIFE-IV	U.S. National Effort	Fuel and blanket rod behavior	Most updated models of irradiation behavior; continuously updated with in-pile data; steady state	Operational	User's manual; articles
LIFE-T	U.S. National Effort	Fuel and blanket rod behavior	Same as LIFE-IV for transient conditions	In development	In progress
FLOW-3D	MIT	Detailed subchannel coolant distribution	Similar to FATHOM-360, but 3-D, only wire wraps	in development	In progress

TABLE 3.13. Models for Subchannel Analysis Codes

Code Name	Author and Reference ^(a)	Model	Formulation		Limitations	
					Bundle Type	Convection Regime
ENERGY I	Khan et al. 1974-1975	Porous body and subchannel	C and E ^(b)	Marching	Tested for grid-spaced and wire-wrapped bundles	Forced convection only
ENERGY II		Porous body and subchannel	C,AM(-) ^(c) ,E	Marching	Tested for grid-spaced and wire-wrapped bundles	Forced and mixed convection
ENERGY III		Porous body and subchannel	C,AM,E	Marching	Tested for grid-spaced and wire-wrapped bundles	Forced and mixed convection
COTEC	Novendstern 1972	Subchannel	C and E	Marching	Wire-wrapped only	Forced convection
FULMIX	Magee 1972	Subchannel	C and E	Marching	Wire-wrapped only	Forced convection
FULMIX II	Wei 1977	Subchannel	C,AM,TM,E	Boundary value problem solved	Wire-wrapped only	Forced convection
ORRIBLE	Wantland 1974	Subchannel	C,AM,TM,E	Marching	Wire-wrapped only	Forced convection
SIMPLE	Jones 1977	Subchannel	C and E;	Marching	Wire-wrapped only	Forced convection
THI-3D	Sha 1976	Subchannel	C,AM,TM,E	Boundary Value problem solved	Wire-wrapped only	Forced and mixed convection
COBRA-IV	Wheeler 1977	Subchannel	C,AM,TM,E	Boundary value problem solved	Bare, grid-spaced wire-wrapped	Forced and mixed convection
COMMIX	Domanus 1977	Porous body	C,AM,TM,E	Boundary value problem solved	Bare, grid-spaced wire-wrapped	Forced and mixed convection

(a) All these references can be found in section N4 of the references - Appendix A.

(b) C,AM,TM and E refer to continuity, axial momentum, transverse momentum, and energy equations.

(c) AM(-) indicates no inertial terms in the axial momentum equation.

bundles, the porous-body model can save storage and computation time compared to the subchannel analysis. The degree of empiricism of both methods when applied to rod bundle thermal analysis is the same.

The porous-body model of Khan et al. (1975, N4) treats the rod assembly as a continuous porous media. The energy generation by the rods is modeled by a continuous volumetric heat source distribution. The energy transfer in the transverse direction is modeled by an empirically determined effective eddy diffusivity. For bare-rod bundles and grid-spaced bundles, the eddy diffusivity field models the turbulence generated by the presence of rods and spacer grids. In wire-wrapped bundles the mean flow appears to oscillate around each rod, progressing in an axial direction. For modeling purposes, this flow oscillation may be imagined to contribute an enhanced eddy diffusivity superimposed on the normal eddy diffusivity associated with turbulence. The porous-body model is similar in principle to the one which has been used to study heat and mass transfer in porous media (Khan 1975, N4). The packed bed is analogous to the bundle of rods in a rod assembly. The main difference between the two cases is that the Reynolds number at which the packed beds operate is 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 formulation of the governing equations 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 the continuity, momentum, and energy equations is required. The formulation of the equations for the forced convection model are given in the following discussions. The formulation of equations for the mixed-convection regime of bundle operation is developed by Khan et al. (1975, N4) and Domanus et al. (1974, N4).

In the porous-body model of Khan et al. (1975, N4) the presence of the rods is taken into account by including the bundle axial and transverse porosities in the governing equations. The porosity does not change axially but

(a) Information obtained in personal communications with R. A. Markley, Westinghouse, Advanced Reactor Division.

does change in the transverse direction. Generally, the minimum gap between the rods is taken as the characteristic dimension for defining transverse porosity. With the presence of the rods so included, differential governing equations for mass, momentum, and energy transport are developed for the continuum within the duct surrounding the wire-wrapped rod bundle. The bundle has two predominant regions. Region I is the inner core, which has a uniform superficial axial velocity $(U_I)_z$ (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 a 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) in Figure 3.49 may be written as follows:

$$\rho C_p (U_I)_z \frac{\partial T}{\partial z} = (\rho C_p \epsilon_I + \kappa K) \left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right] + Q(x,y,z) \quad (48)$$

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 of K

The coefficient, κ , of the thermal conductivity, K , in Equation 48 is included in order to account for the distortion of heat flow lines as they curve around the rods, which are assumed not to transmit heat.

In Region II the energy transport equation is

$$\rho C_p (U)_s \frac{\partial T}{\partial s} + \rho C_p (U_{II})_z \frac{\partial T}{\partial z} = (\rho C_p \epsilon_n + \kappa K) \frac{\partial^2 T}{\partial n^2} + (\rho C_p \epsilon_s + \kappa K) \frac{\partial^2 T}{\partial s^2} + Q(s,n,z) \quad (49)$$

Axial conduction has been neglected. The second term on the right side of Equation 49 represents energy transport in the s-direction (parallel to duct wall) by conduction and by eddy diffusivity. It can be shown that this term is an order of magnitude smaller than the first term on the left side, which represents energy transport by convection in the s-direction, and therefore it can be neglected. However, this second term has been retained in this analysis for completeness and for analysis of bundles with no wire wraps.

Since the energy transport by convection is 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 to be 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 assumed to be equal to ϵ in Region I. This was later justified by excellent match with data.

The boundary conditions coupling Equations 48 and 49 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 the present formulation, the bounding wall is assumed to be insulated and the temperature distribution at $z = 0$ is assumed to be uniform.

The differential Equations 48 and 49 may be written in a dimensionless form, from which useful, generalized dimensionless parameters α^* and C are obtained.

Region I:

$$\frac{(U_I)_z}{\bar{U}} \frac{\partial h^*}{\partial z^*} = (\epsilon + \kappa \alpha^*) \left(\frac{\partial^2 h^*}{\partial x^{*2}} + \frac{\partial^2 h^*}{\partial y^{*2}} \right) + Q^* (x^*, y^*, z^*) \quad (50)$$

Region II:

$$C \frac{\partial h^*}{\partial s^*} + \frac{(U_{II})_z}{\bar{U}} \frac{\partial h^*}{\partial z^*} = (\epsilon^* + \kappa \alpha^*) \frac{\partial^2 h^*}{\partial s^{*2}} + (\epsilon^* + \kappa \alpha^*) \frac{\partial^2 h^*}{\partial n^{*2}} + Q^*(s^*, n^*, z^*) \quad (51)$$

where

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

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

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

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

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

$z^* = z/d_e$

$s^* = s/d_e$

$n^* = n/d_e$

$\bar{U}$ = 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, Equations 50 and 51 may be solved in a differential form or in a finite-difference form.

Equations 50 and 51 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 Figure 3.49, with equal lengths η_T^* between nodes. Since we are solving a set of differential equations, the mesh η_T^*

can be of any convenient size. Unlike in subchannel analysis methods, the spacing between nodes does not have to be assumed 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 two-dimensional temperature gradients in the wall Region II can be obtained.

To solve Equations 50 and 51, the following quantities are known:

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 flow split model of Chiu et al. (1978, N2) (also in Appendix I) and from experimental data (Lorenz 1984, EN4; Pederson 1974, EN2; Chen 1974, EN1).

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" in literature. In order to take into account this two-dimensional effect, the molecular conductivity term was multiplied by a coefficient κ ($\kappa \leq 1$). Although Tortuosity also affects eddy diffusivity in a similar manner as it affects conductivity, this effect was included 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 Figure 3.50. The presence of a large number of rods, contained between the two boundaries at T_1 and T_2 , minimizes the effect of end

FIGURE 3.50. Numerical Solution of $\nabla^2 T = 0$ To Determine the Shape Factor

conditions on the 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 to 1.4 times the minimum lateral porosity, λ_L , 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 50 and 51 were programmed for computer solutions. This program, ENERGY-I, was for assemblies in forced convection only. 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 analyzed, a uniform ϵ^* was assumed in the bundle.

For all sets of data analyzed, 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 (Khan et al. 1974, N4), 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 fuel bundle. Because of

assuming that the wire wrap provides 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. The computer codes, Energy II and Energy III (Khan et al. 1975, N4), include the axial momentum equation, thereby accounting for flow redistribution within an assembly due to buoyancy. The parameters ϵ^* and C are bundle-size dependent. The corresponding bundle-size-independent parameters ϵ_L^* and C_L are given in Table 3.14 (Khan et al. 1979, N4). These are obtained by taking into account the local channel velocity in ϵ^* rather than the bundle-size velocity, which requires introducing the flow-split (Appendix B) parameter into the definition of ϵ_L^* and C_L . The corresponding parameters for subchannel analysis, ϵ_{1L}^* and C_{1L} , are also given in Table 3.14. Figures 3.51 and 3.52 give the recommended values for the parameters ϵ_{1L}^* and C_{1L} for various values of p/d and h/p , where p/d is the rod pitch-to-diameter ratio and h/p is the wire wrap lead-to-rod pitch ratio. The magnitude of ϵ_L^* and C_L can be determined from ϵ_{1L}^* and C_{1L} by the relationship shown in Table 3.14 with the use of the axial and lateral porosities. Table 3.15 presents geometrical parameters for the MIT salt injection experiments called out in Figures 3.51 and 3.52.

The porous-body model has been used for grid-spaced assemblies, which also requires determining ϵ^* empirically.

The subchannel analysis method for predicting flow and temperature fields in rod bundles has been used by a number of investigators (Table 3.10-3.13) for bare-rod, grid-spaced and wire-wrapped bundles. The degree of empiricism in the porous-body and the subchannel analyses is the same, but one may be more advantageous than the other, depending upon applications (Appendix C). The basic principle of the subchannel analysis method is described here using the COBRA-IV (Wheeler et al. 1976, N4) model as an example. The COBRA-IV code as originally written was a multipurpose code with capabilities to handle single- and two-phase flows in LMFBRs and LWRs. The COBRA-WC (whole core) code (George, Masterson, and Basehore 1979, N4) is a streamlined version of the original COBRA-IV code and has been specifically developed for the thermal-hydraulic analyses of LMFBR assemblies either on an assembly basis or

TABLE 3.14. Parameters for Turbulent Bundle Analysis

	Representation	
	Porous Body	Subchannel
Bundle-size dependent	$\epsilon^* = \frac{\epsilon_{HNI}}{U_A D_{e1}}$	$\epsilon^*_{11} = \frac{\lambda_A}{\lambda_{LI}} \epsilon^*$
Parameters	$C = \frac{U_{SII}}{U_A}$	$C_1 = \frac{\lambda_A}{\lambda_{LII}} C$
Bundle-size- independent	$\epsilon^*_L = \frac{1}{S^X_{11}} \frac{\lambda_A}{\lambda_{AI}} \epsilon^*$	$\epsilon^*_{IL} = \frac{1}{S^X_{11}} \frac{A}{LI} \epsilon^*$
Parameters	$C_L = \frac{1}{S^X_{UU}} \frac{\lambda_A}{\lambda_{AII}} C$	$C_{IL} = \frac{C_1}{S^X_{II}}$

TABLE 3.15 Geometrical parameters for MIT Salt Injection Experiment

p/d	h/d	h, in.	Depth, in.	Injection # Leads ^(a)	C ₁
1.25	24	6	12	2.00	0.355
1.25	24	6	25	4.17	0.225
1.25	48	12	24	2.00	0.190
1.25	48	12	26	2.17	0.160
1.25	24	6	26	4.33	0.190

(a) Number of wire wrap leads from the injection at which data was taken.

FIGURE 3.51. Variation of Enhanced Mixing Coefficient with Bundle Geometrical Parameters

FIGURE 3.52. Variation of Swirl Flow With Bundle Geometrical Parameters

on a whole-core basis: i.e., adjacent assemblies are coupled by a heat-transfer boundary condition. The governing equations for mass, momentum, and energy conservation for a subchannel, as they appear in the code, are derived in a finite-difference form (not differential) directly from control volume balances. In order to clearly show the various thermal-hydraulic phenomena modeled by the COBRA equations, they are presented here in a differential form for the subchannel of Figure 3.53. The equation for the conservation of mass is

$$A \frac{\partial \rho}{\partial t} = \frac{\partial}{\partial x} (\rho A v) - \sum \rho s w \quad (52)$$

where ρ is the fluid density, A is the subchannel area, v is the fluid velocity in the axial direction (x), w is the transverse velocity between two adjacent subchannels connected by a gap of width s , and $\sum s w$ is the net crossflow from the subchannel.

FIGURE 3.53. Subchannel Control Volume

The momentum balance in the axial direction gives

$$A \frac{\partial \rho v}{\partial t} = - \frac{\partial (AP)}{\partial x} - K v^2 \rho A - \frac{\partial (v v \rho A)}{\partial x} - \Sigma s w \rho v - \Sigma w' \Delta v - g A \rho \quad (53)$$

where p is the fluid pressure, g is the gravitational acceleration constant, and K is a generalized friction coefficient that includes the effect of the bulk friction factor and pressure loss coefficients, which are both due to orifices or grid spacers. The term $w' \Delta v$ represents the contribution due to turbulent momentum exchange where w' is the turbulent mass exchange and Δv is the difference in axial velocities between adjacent subchannels.

The transverse momentum equation is

$$A \frac{\partial \rho v}{\partial t} = \frac{S}{\ell} \Delta P - \frac{S}{\ell} C w^2 \rho - \frac{\partial (V \rho w A')}{\partial x} \quad (54)$$

where C represents the pressure loss coefficient for flow between adjacent channels and ℓ is the length of the transverse momentum control volume. Note in the above equation that the momentum contribution from neighboring gaps has been neglected. Also, for arbitrary geometries, it is difficult to choose the length ℓ so that all the transverse momentum is accounted for, and the loss coefficient C must be chosen by matching data, since no general correlations exist at this time. However, the essential feature of the transverse velocity is to react to a transverse pressure gradient. This allows the calculation of buoyancy-dominated flows where the transverse pressure gradient resulting from the differences in static head and axial frictional pressure drop is the driving force.

The fluid energy equation is

$$A \frac{\partial \rho h}{\partial t} = - \frac{\partial \rho V A h}{\partial x} - \Sigma \rho w s h - \Sigma C' \Delta T - \Sigma w' \Delta h - \Sigma w U \Delta T_w - \Sigma R H \Delta T_R \quad (55)$$

where h is the fluid enthalpy, C' is the effective fluid conduction coefficient, Δh is the enthalpy difference between adjacent channels for turbulent

energy exchange, U is the effective conductance for heat conducting wall, H is the effective conductance for heat transfer between a rod and a subchannel, and ΔT_w and ΔT_R are temperature differences between a conducting wall and the fluid, and the rod clad and the fluid, respectively. The operators Σ_w and Σ_R have definitions analogous to Σ except that they are for the adjacent walls and rods, respectively.

The forced crossflow induced by the wire wraps in LMFBR assemblies are modeled in COBRA-WC in one of two ways. The forced diversion crossflow may be modeled by employing the wire-wrap model, which is the same as that found in previous versions of COBRA (Wheeler et al. 1976, N4). This particular model sets the crossflow at each point where the wire wrap passes through a gap to a specified percent of the axial flow rate. The nonforced crossflows at any level are then solved to satisfy the momentum equations. Alternatively, the user may choose to model the mixing due to the wire wrap by specifying a percent of the axial flow rate to be forced through the gaps at the assembly periphery at all axial levels (swirl model). The code accounts for additional mixing in the bundle interior by using an enhanced turbulent mixing model similar to that used in the ENERGY code model (Khan et al. 1975, N4). The second alternative has the advantage in performing multiassembly analyses in that the standard subchannels may be lumped together and it does not restrict the maximum node length as the wire-wrap model does. The wire-wrap model is described in more detail later.

Axial conduction in the rod is not included, leaving a one-dimensional equation for the fuel

$$\rho c \frac{\partial T}{\partial t} = \frac{1}{R^2} \frac{k}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T}{\partial r} \right) - Hg \Delta T_c \quad (56)$$

where R is the outer fuel radius, r is the normalized fuel radius, Hg is the effective fuel-to-cladding gap conductance, and ΔT_c is the fuel-to-cladding temperature difference.

A lumped-parameter model is used for both the rod cladding and conducting walls of the form

$$\rho c \frac{\partial T_{c,w}}{\partial t} = H_1 \Delta T_1 + H_2 \Delta T_2 \quad (57)$$

Depending on whether a cladding or wall temperature is being solved for, $T_{c,w}$ is either the cladding or the wall temperature, H_1 is either the fuel-to-cladding gap or wall-to-fluid conductance, H_2 is either the cladding-to-fluid or wall-to-fluid conductance, and ΔT_1 and ΔT_2 are the appropriate temperature differences.

The flow boundary conditions accepted by COBRA-WC include a specified flow distribution at the core or assembly inlet with a uniform pressure profile at the top or, alternatively, a specified pressure drop across the core or assembly. Additionally, a specified inlet temperature distribution and rod power profiles are required. A solution of the steady-state equations provide initial conditions for a transient problem while the user supplies transient information for flow or pressure drop, inlet temperature, and rod powers.

These equations, together with an equation of state and boundary conditions, form the basis of the COBRA model and, while they do not model completely all the phenomena in a reactor core, they do contain the terms most important for obtaining reasonable estimates of flow and temperature distributions for both forced- and natural-convection problems.

The basic solution procedure is Gauss-Seidel with under or over relaxation, depending on the particular equation, for a boundary value solution to the finite-difference analogs to the equations. The equations are iteratively solved at each axial level before proceeding to the next level, the level-by-level solution is repeated until convergence is obtained in all cells.

At each level the energy equations for the coolant and rods are solved first using an orthogonal collocation technique to solve the rod energy equation. The orthogonal collocation matrix is reduced so that the rod temperatures are given in terms of the fluid temperatures. The wall temperatures are

also eliminated so that a combined rod, wall, and coolant energy equation can be solved for the fluid enthalpy. This is done on a core-wide basis since the interassembly heat transfer couples all the subchannel energy equations.

Following the solution for temperatures (enthalpies), the fluid density is obtained from the equation of state. The combined continuity and momentum equations for the subchannels within each assembly are then solved. The momentum and continuity equations may be solved on an assembly-by-assembly basis since there are no hydraulic connections between assemblies other than at the top and bottom of the core. The method for solving the momentum equations at each level is either direct inversion or iterative, depending upon the number and arrangement of the subchannels within an assembly.

For very low flow rates, where recirculation zones might occur within an assembly, an alternative solution method to the momentum equations is available. This method employs the Chorin-Hirt algorithm to solve a Poisson equation in pressure resulting from the combination of the momentum and continuity equations. This method is somewhat slower and is therefore used only when the flow and power distribution warrant it.

The equations are solved implicitly in both time and space. Upwind differencing is used for all the convected quantities which, in some instances, leads to significant flattening of temperature or salt concentration gradients due to numerical diffusion but adds to the stability of the solution scheme.

For most reactor steady-state and transient operating conditions, including the natural circulation transient, the flow field in the reactor assemblies is predominantly axial (parabolic flows) and further simplifications in the COBRA-WC code are possible and also desirable to reduce the computation and storage requirements. Parabolic flow situations allow marching solutions which require less computation time than required for solution of boundary value problems. The marching procedure has led to reduction in computation time in a number of calculational methods shown in Table 3.13 as compared to COBRA-WC, which solves a boundary value problem.

However, marching solutions do not allow downstream effects to propagate upstream, so marching codes cannot handle flow blockages within the bundle or

the formation of recirculation zones (local flow reversals such as behind a blockage or formation of a recirculation zone due to buoyancy-induced flow redistribution). The four empirical constants that need to be determined by comparison with data in the COBRA-WC code are the mixing coefficient, β , the crossflow resistance coefficient K_{ij} , the s/l parameters in the transverse momentum equation and the wire-wrap mixing parameter, δ , discussed below. Tang et al. (1978, Book) have discussed other subchannel analysis codes developed in the United States and abroad.

If the wire-wrap sweeping effect is not modeled as an enhanced eddy diffusivity of heat, a wire-wrap model is needed in subchannel codes to provide forced flow mixing between channels. A number of wire-wrap flow sweeping models have been reported by Rogers and Todreas (1968, survey). They showed that the most general form of the basic flow sweeping equation should be

$$\frac{W_{ij}}{G_i d_w} = \frac{\pi(d + d_w)}{h} \alpha_f \quad (58)$$

where W_{ij} is the forced-flow mixing rate, d is the rod diameter, d_w is the wire-wrap diameter, h is the axial pitch of the wire wrap, and α_f is the fraction of the total subchannel area (which contains the wire wrap) that is free-flow area. Ginsburg (1972, N4) recommended a force-flow mixing model using the following expression for the forced-flow mixing rate

$$\frac{W_{ij} h}{6m_i} = f(p/d, h/d, Re) \cdot \frac{\pi d_w}{6A_i} (d + d_w) \delta_{ij} = \phi_{ij} \delta_{ij} \quad (59)$$

where the correction factor, function f , must be obtained from experiment; m_i is the mass flow rate, $\delta_{ij} = +1, -1$, or 0 when flow is transported from subchannel i to j , from j to i , and for no wire crossing the i - j boundary, respectively. The dimensionless forced-flow mixing rate is ϕ_{ij} . This model specifies a pulse-type axial distribution of interchannel mixing rate. The model has been tested with only very limited experimental data.

The COBRA-WC wire-wrap model (Donovan et al. 1979, EN4), is based on the assumption that the crossflow induced by the wire wrap passing through a gap is some fraction of the subchannel flow. The expression developed for the fractional split is

$$W_{\text{forced}} = \pi(d + d_w) \left(\frac{s}{A_i} \right) \left(\frac{\delta}{\Delta X} \right) m_i \quad (60)$$

where

s is the gap width

A_i is the area of subchannel contributing the forced cross flow

ΔX is the axial node length

δ is the arbitrary parameter, usually set to $\Delta X/h$.

Subchannel flow areas and wetted perimeter are adjusted to account for the presence of a wire wrap in the subchannel. The forced cross flows are then held constant while solving the transverse momentum equations for the remaining cross flows (Donovan et al. 1979 EN4).

An improvement in the wire-wrap sweep flow model was made (Donovan et al. 1979, EN4) by taking the reduction in the channel area due to the wire wrap as only half of the wire-wrap cross-sectional area. The value of δ was set to $1.25 \Delta X/h$, and a modification was made to effectively transmit some of the pressure drop required for forced flow to the nodes upstream of the forced gap.

With these improvements considerable success has been met in predicting data (Donovan et al. 1979, EN4). Figure 3.54 shows a comparison of the prediction and data at the exit of a 19-heated-pin bundle (Fontana et al. 1974, EN4). Figure 3.55 shows that the code predicts the circumferential pressure distribution in a wire-wrapped bundle well. The data shown was obtained by Lafay et al. (1975, EN4) in a 19-pin bundle. The ability of the COBRA-IV code to predict the peripheral channel pressure distribution is encouraging. However the code was unable to predict the Argonne National Laboratory (ANL) point-injection, salt-tracer experiment (Lorenz et al. 1974, EN4), which was performed in a model 91-pin fuel assembly. Figures 3.56 a, b and c show the

FIGURE 3.54. Bundle Exit Temperatures ($X = 21$ in.)

axial variation of the measured and predicted salt concentration. The code predicts that the salt concentration profiles tend to spread faster than the experimental. Because turbulent mixing was not allowed in these calculations, this flattening effect is attributed to either numerical diffusion inherent to the upwind enthalpy advection scheme or due to excessive forced-flow prediction by the wire-wrap model or both. This effect becomes especially important in regions of low heat generation such as the upper plenum region of a reactor bundle.

FIGURE 3.55. Cyclic Pressure Variation in a Peripheral Subchannel

A number of other wire-wrap models have been proposed which include the models of Sha (1976, N4) and Miyaguchi (IAEA 1979). Miyaguchi describes a wire-wrap model used in the analytical code SWIRL. The code predictions are compared with data obtained in a 19-heated-pin blanket assembly and in 91- and 169-heated-pin fuel assemblies. The data compares well with predictions even when only two, three, and five pins are heated in the bundle. The wire-wrap model was calibrated with data for bundles of three different sizes--no bundle size effect was observed. However, no comparison of code predictions with data were reported in the unheated region above the bundle, which appears to

TABLE 3.16. Mixing Experiments in Wire-Wrapped Bundles of Various Sizes

Bundle Size (Number of Pins)	Literature Reference Categorized by Experiment Type			
	Hot Water Injection	Salt Injection	Heated Pin	Laser-Doppler Velocimeter
7	Skok 1973(a)	--	--	Wheeler et al. 1974
19	--	--	Hoffman and Baumgartner 1974 Miyaguchi and Takahashi; 1979 IAEA	
37	--	--	--	--
61	--	Chiu et al. 1977 and 1978 Hanson and Todreas 1977	--	--
91	Lorenz et al. 1974 Lorenz and Ginsberg 1977	--	--	--
127	Hines 1971	--	--	--
169	--	--	Miyaguchi and Takahashi 1979 IAEA	
217	--	Collingham et al. 1971(b) Fuchs and Faesch 1973	Graeber and Rieger 1973;	--

(a) All references refer to Section EN4 in the list of references (Appendix A) unless otherwise specified.

(b) Ongoing work in this area at MIT.

be the region of "true test" for the wire-wrap model. Based on present experience, it appears that further improvements in the wire-wrap model are needed.

Purely analytical solutions for predicting temperature distributions in wire-wrapped assemblies are few. Mikic, Khan, and Todreas (1977, N4) describe an approximate analytical method for predicting temperature distributions in wire-wrapped assemblies operating in forced convection.

TABLE 3.17. Mixing Experiments in Wire-Wrapped Rod Bundles With Different p/d and h/d Ratios

	Literature References Categorized by h/d Ratio From 14 to 55												
p/d	<u>14</u>	<u>17</u>	<u>19</u>	<u>21</u>	<u>24</u>	<u>26.5</u>	<u>28</u>	<u>34</u>	<u>43</u>	<u>48</u>	<u>50</u>	<u>52</u>	<u>55</u>
1.065	1,2		3										
1.09				9									
1.1													
1.14	3			3(a)			3						
1.20									9				
1.24										11,12		13,14	
1.25					(a)					(a)			
1.28					10								15
1.315		4,5,6,						4,5,6,			4,5,6,		
1.355		7,8						7,8			7,8		

1. Chiu et al. (1977, EN4)	9. Miyaguchi and Takahashi (1979, IAEA)
2. Chiu et al. (1978, EN4)	10. Hines (1971, EN4)
3. Skok (1973, EN4)	11. Lorenz et al. (1974, EN4)
4. Baumann and Moller (1969, EN4)	12. Lorenz and Ginsburg (1977, EN4)
5. Hoffmann (1973a)	13. Collingham et al. (1971, EN4)
6. Hoffmann et al. (1977, EN4)	14. Fontana et al. (1974, EN4)
7. Hoffmann and Baumgaertner (194, EN4)	15. Wheeler et al. (1974, EN4)
8. Hoffmann (1973b)	

(a) Ongoing work at MIT

A large number of experiments have been conducted (Tables 3.16 and 3.17) in wire-wrapped bundles in order to improve the understanding of the thermal-hydraulic phenomena within the bundle and for calibration of existing computer programs--both porous body and subchannel analysis codes. Khan et al. (1974, 1975, N4) and Chiu et al. (1978, N4) discuss the detailed analysis of data for calibration of the porous body and subchannel analysis models in the ENERGY code. Functional relationships for ϵ^* and C, dependent upon p/d, h/d, Re, and Pr, have been developed. Basehore et al. (1979, N4) described their efforts for calibration of the COBRA-WC code. Tang et al. (1978, Book) discuss some of the major findings from the earlier experimental data. A discussion of the

ORIENTATION A
(TOP VIEW)

FIGURE 3.56a. Subchannel Concentrations

FIGURE 3.56b.

FIGURE 3.56c.

accuracy of the different types of experimental techniques used (heated-pin, hot-water-injection, and salt-injection experiments) has been presented by Khan et al. (1975a, N4). It appears that heated-pin experiments not only provide the right boundary conditions, but the data is more reliable than hot-water-injection and salt-injection experiments, especially if integral, axially averaged mixing coefficients are needed (as required for the porous body models). One reason for this is that the dominance of numerical diffusion effects can prevent adequate code calibration in point-injection experiments, but the heat source term dominates in the heated-pin experiments and code calibration is more reliable.

In spite of the large number of experimental data available, and in spite of code calibration with data, predictions of temperature in regions of steep temperature gradients are poor, primarily due to the inherent assumptions in the subchannel code. This is especially true near the wall regions of grid-spaced and wire-wrapped assemblies. Figures 3.57 and 3.58 from Hofmann et al. (1979, IAEA) show that major differences exist between code predictions in the wall regions of a grid-spaced and a wire-wrapped assembly, respectively. Thus circumferential coolant and cladding temperatures in the wall region could be greatly under-estimated (Hofmann et al.) by subchannel computer programs (Figure 3.59).

Assessment of the Existing Verified Analytical Methods and Computer Programs for Design Calculations

Assessment of computer programs and analytical methods used for design calculations have been made by comparison with data and by comparison between the methods themselves. Comparison of the subchannel average temperature, the duct wall circumferential temperature distribution, cladding circumferential temperature distribution, flow distribution, pressure drop, and peak channel temperature have been made both in the U.S. and abroad (Hofmann et al. 1979, IAEA-5). The results of the comparison between the methods themselves and with existing data clearly showed that further improvements are required in the subchannel analysis methods and that additional data is needed to validate the code. The following is a brief summary of the trends observed in the forced-convection regime of bundle operation:

- The pressure drops using existing correlations differed from data by up to 15%.
- The temperature distributions between the codes and between the codes and data are similar.
- Peak temperatures are predicted within $\pm 10\%$ of the mixed mean coolant temperature rise.
- Duct wall circumferential temperature distribution predicted by the codes differed significantly, especially above the active core region of the bundle.
- Subchannel analysis codes underpredict the cladding circumferential temperature gradient, whereas local distributed parameter codes appear to overpredict it.
- Subchannel flow distribution behind grid spacer is not predicted well.
- For wire-wrapped bundles the predicted inlet flow split for fuel assemblies operating in forced convection does not match data.

Multiassembly Computational Methods

A number of steady-state and transient multiassembly subchannel analysis computer programs have been developed (Table 3.12). The design of an LMFBR core-restraint system requires a knowledge of the detailed three-dimensional temperature distribution in the assembly duct wall. This is because swelling and irradiation creep are temperature dependent and may significantly enhance duct wall bowing due to a temperature gradient across the duct. Figure 3.60 (Tang et al. 1978, Book) shows the limited free-bow core-restraint concept used in the FFTF and CRBR core designs. In order to determine the forces transmitted through the load pads onto the core-restraint system, it is necessary to have a knowledge of the duct wall temperature distribution, not only in the active core region but in the entire length of the assembly.

Interassembly heat transfer can significantly affect duct wall temperatures, especially for heterogeneous cores where blanket assemblies are mixed with fuel assemblies and in the vicinity of regions of large power gradients

FIGURE 3.57. Mean Subchannel Temperatures: Comparison Between Computed and Experimental Results

FIGURE 3.58. Mean Subchannel Temperatures and Mass Flow Rates: Comparison Between Computed Results

- EXPERIMENT TE A[], CORRECTED TO CLADDING SURFACE
- MEAN SUBCHANNEL TEMPERATURE (ARTIS-CODE)
- MEAN ROD SURFACE TEMPERATURE $Nu = 12$ ① ② ③
- MEAN ROD SURFACE TEMPERATURE $\begin{cases} Nu = 7 & \text{SIDE CHANNEL ①} \\ Nu = 14 & \text{CENTRAL CHANNEL ① ② ③} \end{cases}$

FIGURE 3.59. Azimuthal Temperature Variation of the Side Rod Cladding: Comparison Between Local Measurements - Global Calculations

FIGURE 3.60. Limited Free Bow Core-Restraint Concept

such as near control assemblies and at the interface of fuel blanket assemblies. Figure 3.61 shows the orificing pattern for a heterogeneous core (Khan et al. 1976, N2) with two rows of internal blanket assemblies. The flow rate to the internal blanket assembly, in the absence of variable orificing, does not change with time, although there is a significant change in the blanket power generation from beginning of life (BOL) to end of life (EOL). Consequently, the internal blanket assembly is initially overcooled. The resulting temperatures and temperature gradients between adjacent fuel and blanket will result in the duct wall temperatures and temperature gradients being changed continuously due to interassembly heat transfer.

The SUPERENERGY (Chen and Todreas 1975, N4), TRITON (Carelli and Bach 1978, N2) and CORE-3D (Beitel and Khan 1977, N4) codes have been developed to take into account the effect of interassembly heat transfer on assembly duct temperatures. These codes assume that the interstitial fluid between adjacent assembly duct walls is stagnant and that the mode of heat transfer between assemblies is conduction. Thermal resistances due to the duct wall and stagnant sodium are added to the convective thermal resistances on the inside of

FIGURE 3.61. Orificing Pattern for 1/6'th Core

the assemblies. Figure 3.62 (from Khan et al. 1975, N4) shows the variation of cross-duct temperature differences with bundle size for two cases: where mixing is allowed and where no mixing is allowed. No interassembly heat transfer effects are included. The cross-duct temperature differentials as predicted by the TRITON (Tang et al. 1978, Book) code, which uses the COTEC (Novendstern 1972, N4) model, are significantly reduced in the radial blanket assembly by interassembly heat transfer at the core periphery. This is

important for core restraint system design. However it is important to note that cross-duct temperature distributions are affected not only by interassembly heat transfer but by the flow and temperature distributions within an assembly. Consequently, duct wall temperatures are sensitive to thermal-hydraulic models such as the wire-wrap mixing (Figure 3.62), flow split between interior and wall channels, and bundle tolerance considerations.

Considerable difference currently exists in the predictions for peak circumferential coolant temperature gradients within an isolated (insulated) assembly, as shown in Figure 3.63. The differences are primarily due to the mixing models within assemblies and, to a smaller extent, to "numerical diffusion". In the presence of interassembly heat transfer, the deviation in the predictions could be more.

Mixed Convection Within Reactor Assemblies

An interesting phenomenon was observed during the steady-state operation of EBR-II reactor assemblies at a low flow rate. It was found that at low flows the temperature distribution is significantly flattened across the core (Figure 3.64). It was also observed that assembly temperature distribution was significantly flattened (Figure 3.65). A number of thermal-hydraulic phenomena are responsible for this. These include interassembly heat transfer, interassembly flow redistribution, intra-assembly heat transfer and intra-assembly flow redistribution. These phenomena are coupled and cannot be accounted for separately. Multiassembly effect, such as interassembly heat transfer have been discussed earlier. Interassembly flow redistribution is discussed in more detail, later, in the section on unsteady-state operation of a reactor core. Here the intra-assembly effects at low flow rates are discussed.

Even in the absence of power skews, significant temperature gradients can exist within a reactor assembly because the wall coolant channels have twice as much flow as the interior channels but the same rate of heat generation. In the presence of power skew, the temperature gradient is further increased. In forced convection, the buoyancy-induced axial flows are negligible as compared to the average flow in a channel. This is in contrast to behavior at low flow rates where flows are of the same order.

FIGURE 3.62. Variation of Maximum Cross-Duct Temperature Difference with Bundle Size

At low flow rates, the hotter channels receive fluid from the colder channels due to differences in densities. This buoyancy-induced flow redistribution effect within the bundle can significantly reduce temperature gradients. The mixed-convection regime of bundle operation is where buoyancy-induced flow redistribution superimposed upon the forced-inlet flow into the bundle results in mitigation of lateral coolant temperature gradients within the bundle. At low flows the two modes of energy redistribution within an assembly are thermal conduction and buoyancy-induced flow redistribution.

The effect of energy redistribution by wire-wrap mixing can be an order of magnitude smaller at Reynolds numbers of bundle operation below about 700 (Khan 1976, N4). The effect of buoyancy-induced flow redistribution and thermal conduction on mitigating temperature profiles can be seen in Figures 3.66,

FIGURE 3.63. Variation of Maximum Cross-Duct Temperature Difference with Axial Distance in the Upper Plenum Region (Khan et al. 1975, N4)

3.67, and 3.68 from Juneau and Khan (1979, 04), which compare the predictions of codes with the experimental data of Engel et al. (1978, E04). Engel et al. performed heat transfer experiments in a prototypical 61-pin blanket assembly (Figure 3.69) over a wide range of operating conditions, which included a power skew range from flat to 2 (maximum to average) and a Reynolds number range from 490 to 7300. Temperature measurements were reported at several axial planes both within and above the heated zone. Figures 3.66, 3.67, and 3.68 show the flattening of the bundle radial temperature profile due to a reduction in the bundle total flow rate for the same power skew and power-to-flow ratio. Similar results were observed (Juneau et al. 1979, 04) in the analysis of the HEDL 217-pin natural-convection test (Millburg et al. 1978, E04) and in the analysis of the Battelle 2X6 rod assembly (Quigley et al. 1977, E04). Table 3.18 gives the geometrical and operating regions of the various mixed-convection tests.

FIGURE 3.64. Variation of Subassembly Temperature Rise with Subassembly Row Number in EBR-II Instrumented Assembly

An interesting result obtained in the analysis of the 217-pin heated bundle test is that heat loss at the bundle duct wall penetrates significantly into the bundle. This is especially true below a Reynolds number of 600. Millburg found that a heat loss at the duct wall, as low as 2% of the total bundle heat input, can cause significant flow redistribution to the interior channels, significantly reducing their temperatures. At lower Reynolds numbers this flow redistribution increases. This result indicates that the flattening of temperature profiles in experiments at a low Reynolds number is not only due to flow redistribution and thermal conduction but could also be due to a radial heat loss at the duct wall. At Reynolds numbers below 200 flow redistribution can be very sensitive to geometric and boundary conditions.

FIGURE 3.65. Variation of Fuel Temperature Rise with Pin Bundle Row Number in EBR-II Instrumental Assembly

Khan et al. (June 1975, N4) have developed the following criterion to predict the inception of the mixed-convection regime within a wire-wrapped bundle. When the modified Grashof number, Gr^* is greater than 0.025, the assembly is assumed to operate with mixed-convection. Gr^* is defined as follows:

$$Gr^* = \frac{Gr}{MfRe^2} \frac{(Qp - 1)}{2v_1L} \quad (61)$$

where

$$Gr = g (T_M - T_{IN}) de^3 / \nu^2$$

f = friction factor

M = Novendstern's (1972, N2) multiplier for estimating pressure drop in wire-wrapped bundles

$$Q_p = \text{bundle peak to average power skew} = \frac{2Q_1}{Q_1 + Q_2}$$

Q_1, Q_2 = maximum and minimum volumetric heat generate rates

$(T_M - T_{IN})$ = bundle mixed mean temperature rise

de = hydraulic diameter of bundle interior channel

ν = dynamic viscosity

$$\nu_1 = \text{mixing parameter} = \frac{(P/d-1)}{D} \frac{16}{\pi} \left(\epsilon^* \frac{\bar{\lambda}_a}{\lambda_L} + \kappa \alpha^* \bar{\lambda}_a \right)$$

$$\epsilon^* = \text{dimensionless eddy diffusivity} = \frac{\epsilon_H}{Vde}$$

$$\alpha^* = \text{dimensionless thermal diffusivity} = \frac{\alpha}{Vde}$$

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

λ_L = bundle minimum lateral porosity

κ = Tortuosity factor for transverse two-dimensional flow of heat in a bundle

D = duct flat-to-flat dimension

The above criterion does not hold good for a power skew greater than 1.1 across the bundle. Yet, as mentioned earlier, buoyancy-induced flow redistribution can occur in such bundles. For flat power skews Chiu et al. (1978, N4) have developed a criterion for predicting the inception of mixed convection in a wire-wrapped bundle.

A number of other models that need to be mentioned are those by Tsai et al. (1973, 04), the analytical method of Ramm et al. (1977, 04) and Sha et al. (1976, 04). Tsai neglects the frictional drag due to the presence of wire wrap in his strip model and probably overpredicts the buoyance-induced flow redistribution.

FIGURE 3.66. Variation of Normalized Temperature $(T-T_{in})/\bar{\Delta T}$ Across the WARD Assembly

FIGURE 3.67. Variation of Normalized Temperature $(T-T_{in})/\bar{\Delta T}$ Across the WARD Assembly

FIGURE 3.68. Variations of Normalized Temperature $(T - T_{in}) / \Delta T$ Across the WARD Assembly

TABLE 3.18. Geometric and Operating Conditions for Mixed-Convection Data Analysis

Data Set	Geometrical Description	Operating Characteristics	Comments
1. BNWL - 2x6 rod array tests Quigley et al. (1977, E04)	2x6 electrically heated rod bundle; on a square pitch; $d = 0.475$ in., $p/d = 1.21$; instrumented length = 48 in.; rod-to-wall spacing = 0.350 in.; water coolant; no wire wrap	Reynolds number; 480 to 960; 2:1 and 1:0 power skew; $T_{inlet} = 60^{\circ}\text{F}$ $\Delta T = 30 - 33^{\circ}\text{F}$	Velocity distribution measured by LDV; temperature distribution measured by thermocouple rake
2. HEDL - 217-pin tests Millburg et al. (1977, E04)	217-pin, electrically heated rod bundle, $d = 0.23$ in.; $p/d = 1.24$; heated length = 36 in.; $h/d = 52$; sodium coolant	Reynolds number: 200 to 1000; 4 different power skews; uniform peak average = 1.19 and 1.65 and step change in power; $T_{inlet} = 400^{\circ}\text{F}$	Thermocouples used for temperature measurement at a number of planes
3. WARD - 61-pin tests refer to Engel et al. (1978, E04)	61-pin, electrically heated rod bundle; $d = 0.5$ pin; $p/d = 1.082$; $h/d = 7.7$; sodium coolant	Reynolds number: 490 to 8000; uniform power skew and several non-uniform skews	A number of runs allowed for radial heat transfer from duct; these runs were not analyzed.
4. FFM-II - 19-pin tests Fontana et al. (1976, E04)	19-pin, electrically heated rod bundle; $d = 0.23$ in.; $p/d = 1.24$; heated length = 21 in.; $h/d = 52$; sodium content	Reynolds number varied from 1000 to 7000; a wide range of power skews	Only mixed-convection data results presented here

FIGURE 3.69. WARD Test Section Cross Section

3.3 NON-NOMINAL BUNDLE AND NOMINAL CONDITIONS INCLUDING MIXED CONVECTION

3.3.1 Identification of Bundle Distortions

Figure 3.61 shows a layout for a heterogeneous core with inner and radial blanket assemblies, fuel, and radial shield assemblies. Each assembly is surrounded by a hexagonal duct. Existing duct designs call for an unvented duct. Vented duct designs and ductless cores (Dubberly et al. 1977, 04; Ziegler 1976, 04) will not be addressed here, although considerable effort is being made to estimate their beneficial effects on LMFBR core design.

At the beginning of life (BOL) the reactor bundles consist of nearly uniformly spaced, straight rods supported by a spacer system such as a grid or wire wrap. The initial manufacturing and assembling tolerances may introduce some nonuniformity in rod spacing. The manner in which the assembly tolerances are accommodated within the bundle affects temperature distribution. This is seen in Figure 3.70, which shows a comparison of the measured temperature profile in a central Rhapsodie assembly after 50 days of irradiation with two different predictions--one for the case where the bundle tolerance is fully accommodated within the bundle (fully compacted) and another for the case where the clearance is distributed uniformly within the bundle. The skewed temperature profile suggests a nonuniform accommodation of tolerance, i.e., a half-compacted bundle, especially since it rotated when the assembly was rotated. The temperature profile at BOL would also be affected by pin bowing within the assembly due to a steep temperature gradient in the vicinity of the wall (Figure 3.70), which subjects the pin to a differential thermal expansion and bow.

The effect of initial manufacturing tolerances and thermal bowing on the temperature field within a bundle may well be overridden by irradiation-(fast neutron spectrum) induced distortions within a bundle. These distortions are due to dilation of duct walls and pin diameters caused by voidage swelling, irradiation creep, pin and duct wall bowing due to differential swelling (caused by a temperature differential across the pin and the duct wall), and pin bundle deformation (dispersion, corkscrewing) due to bundle-duct interaction. Bundle-duct interaction occurs because the "hotter" bundle swells more

FIGURE 3.70. Skewed Temperature Profile in Central Raphsodie Assembly

than the "colder" duct wall. Table 3.19 lists the various bundle distortion mechanisms that are assumed to occur in an LMFBR bundle. Because of interaction between the various modes of distortions (Table 3.19) the bundle geometry becomes irregular, continually changing in the direction of flow. Thermal-hydraulic problems then become essentially three-dimensional. Physical modeling becomes much more speculative, and mathematical difficulties, which increase significantly, are further compounded by the strong link between mechanical and thermal calculations.

It is not now known if distortions lead to a reduction of assembly life within the core. Assembly life in the core is determined by a number of factors, which includes strain/CDF (cumulative damage function) constraints on pin life, interference constraints between bundle and duct, and allowable load constraint on duct-to-duct contact. For example, for the fuel pin, the principal structural design limits ensuring structural integrity may include the following (Tang et al. 1978, Book, pp. 78-79). Thermal creep strain and plastic strain values during steady-state and transient operation should be maintained below certain limits. Criteria for cumulative mechanical damage function ($CDF = \int_0^t t_R dt$ where t_R is the time to rupture and depends upon cladding loading, fluence, and temperature), including cyclic fatigue due to transient and seismic event, vibration, handling, and part-load operation, should be ≤ 1.0 . Other design guidelines on fuel element include 1) linear power limitations in the peak fuel element to avoid inception of fuel melting at a prescribed over-power condition and 2) maximum cladding mid-wall temperature limits for anticipated transients and for unlikely events. Similar limits pertain to LMFBR radial blanket and control assembly elements. The interference constraint between the bundle and duct pertains to potentially damaging pinch loads along the duct, intrabundle dispersion and increased cladding hot-spot temperatures due to interior pin-to-pin closure, and edge pin-to-duct closure/contact. Thermal-hydraulic conditions in fuel and blanket assemblies should be such that duct dilation resulting from the combined effects of duct temperatures, differential pressure, and irradiation swelling and creep on the duct will not prevent assembly removal during refueling.

TABLE 3.19. Bundle Distortions Mechanisms for Nominal Conditions

1. Initial built-in manufacturing and assembly tolerances
2. a) Pin bowing due to thermal gradient at beginning of life
b) Pin bowing due to a combined effect of differential temperature and swelling
3. Bulging of the duct wall due to irradiation-induced creep and swelling
4. Movement or shift of wire wrap spacer due to relaxation of wire wrap tension resulting from irradiation-induced creep
5. Bundle-duct interaction causing a corkscrewing of the assembly
6. Assembly bowing due to differential swelling of the duct walls and pins
7. Combination of above

3.3.2 Design Innovations

The enormous complexity of defining the distorted geometry and therefore the inability to predict temperatures with some degree of confidence is one of the difficulties that has inspired designers toward improvements in bundle designs in attempts to reduce bundle distortions. Such designs include the use of half wire (smaller-diameter wire) on rods adjacent to the wall or the use of flow obstructions in the wall channels (Weber and Cornet 1979, IAEA, Miyaguchi 1979, IAEA), which would reduce the temperature differentials across the rod at the wall and between the bundle and duct. Reduction in the temperature differentials would reduce rod bowing and the differential bundle-duct growth. However, the half wire does reduce the edge clearance and the amount of overall bundle distortion which can be accommodated. Also to be considered is the fact that the reduction in bundle-to-duct interference has the net result of increased duct growth because of higher duct temperatures. This increase must be considered in relation to the design and performance of the overall core system.

End-of-life loads and bundle interference can be decreased by increasing the amount of as-built porosity. This does provide room for relative growth, but in-pile data has indicated that an upper porosity limit exists at or near the reference FTR (FFTF) porosity for precluding excessive fretting and wear.^(a) A possible method of reducing the effects of interference is to improve spacing techniques. For wire-wrapped assemblies this would mean using a different wire-wrap configuration than the single straight-start wire-wrap configuration used in FTR- and CRBR-type designs. The principal advantage of the approach is the ability of an assembly to accommodate swelling internally without the negative aspects of large as-built porosity. It would thus be possible to construct a tight bundle (reduced porosity) for improved vibration control while still providing room for relative growth. Kaplan and Punches (1977, E04) have performed experiments in a 37- and 127-pin bundle on a number of improved spacing techniques such as staggered starts, partially wireless bundles, etc., in order to determine the compression characteristics of these bundles. The various configurations tested include the straight start, straight start with wireless pins, 0° - 45° - 90° and 0° - 30° - 60° starts, and the locked wrap (Figure 3.71). Figure 3.72 shows the observed maximum bundle-to-duct gap reduction with across-flats interference, taken directly from test data. The across-flats interference is the differential bundle-to-duct growth after assembly clearance (initial porosity) has closed. If pin-to-pin nesting, i.e., dispersion, did not occur, the 0.056-in. gap wire diameter) would have closed at an across-flats interference of 0.056 in. because of the bundle helical bending without relative pin-to-pin displacements. The locked-wire wrap exhibits the best clearance behavior because of its nesting ability. Figure 3.73 shows the absolute minimum pin-to-pin gap with cross-flats interference. Also shown is the limiting pin-to-pin gap which would cause excessive coolant temperature in an FTR-type design. The softened bundles do provide superior compression characteristics compared with the reference straight-start wire under isothermal conditions. Kaplan and Punches prefer the straight-start wire wrap bundle softened by wireless pins bundle. This bundle not only

(a) The author obtained this information in personal communications with R. Jackson of the Hanford Engineering Development Laboratory.

FIGURE 3.71. Locked-Wrap Design Wire Contact Locations

showed favorable characteristics during the compression tests but also showed other potential economic advantages such as reduced pressure drop and a slight gain in breeding ratio.

For grid-spaced bundles, bundle-duct interaction may not be a major factor limiting assembly life. However a concern is the flow maldistribution

FIGURE 3.72. Bundle-Duct Observed Minimum Clearance Versus Across-Flats Compression Displacement

FIGURE 3.73. Pin-Pin Observed Minimum Clearance Versus Across-Flats Compression Displacement

that might occur because of an eccentric positioning of the grid within the bundle. Weber and Cornet (1979, IAEA) compare the effect of skirt on flow resistance in the wall channels of a bundle with an eccentric grid. The SNR-300 Mk Ia grid elements do not contain skirts, whereas the SNR-2 grid elements are attached with skirts to the wrappers. A schematic illustration of the influence of the skirt on the flow resistance of the wall channels is given in Figure 3.74. The grid without the skirt showed a decreased total flow resistance with eccentricity due to the reduction of both friction and grid resistance. A lower total flow resistance in the wall subchannel means higher mass flow rate and therefore lower temperature rise--this increases the nonuniformity in the bundle temperature. For the grid with skirts it is possible to design for uniform total flow resistance with eccentricity--an obvious benefit.

3.3.3 In-Pile Experience

Considerable in-pile experience has been gained in Rapsodie (Leteinturier and Carter 1979, IAEA), Phenix (Marback, Millet et al. 1979, E04) and EBR-II^(a) irradiation examination of the Phenix bundle (Marback et al. 1979, E04) containing 217 wire-wrapped fuel rods located in the first ring of the reactor for 6 cycles (6.6 atom% burnup) showed a number of interesting bundle features including the following (based on translation of Marbeck et al.)

- a bell-shaped deformation of the duct face the amplitude of which is accentuated for the levels corresponding to a possible contact between the cladding and the duct
- bending of the cladding under the effect of tension of its own wire
- ovalization of the cladding
- local cladding to cladding contact which is generally located close to fuel rods having an angular displacement of the wire
- a total cross-section for passage of sodium of about 10% less than that of the initial bundle, giving about 2% reduction in flow
- It appears that the hydraulics of the bundle are not disturbed by the dispersion of the areas of the channels and by the displacement of the fuel rods.

FIGURE 3.74. Influence of the Skirt on the Flow Resistance

No breaches occurred until 6.6 atom% burnup. A possible reason for absence of pins breaches could be due to the solution-annealed 316 stainless steel used for the cladding material. Peak swelling in stainless steel alloy occurs at a lower temperature than in CW-304. Consequently, peak swelling occurs below the center line of the active fuel. If local hot spots occurred in the neighborhood of the region of peak swelling, it would be in a region of much lower (design) temperature than if it occurred well above the midplane of the core. Thus the cladding hot-spot temperatures can be expected to be lower for SA-316 stainless steel cladding material than for CW-304.

3.3.4 Summary of the Present Understanding on Anomalous Pin Temperatures in Bundles Tested in EBR-II

HEDL fuel pins irradiated in EBR-II are posutlated to have failed because of global overheating (P41-A3R, P41-B10, P42-B71), global and local (pin-to-pin contact or near pin-to-pin contact) overheating (P-12AA-63K, P12AB-11B, N/E-122), local heating due to pin/insulated duct contact (P23A-37, P23B-1A, P14-26), global overheating and breach in cladding wear mark (PNL 10-14, PNL 11-39), pin-to-pin contact (P5A-1, P5B-17, P14-29, P23B-73E), and fuel-cladding mechanical interaction (P42R-D1, P42R-D2, P43-D77). The cause of the breach in pin N/E-054 is not known. Some of the breached pins (P23B-12A, P23B-31B, P23C-7A, P23C-12A, P52-C31, P60-A3) are undergoing post-irradiation examination. Pin P40-D91 had an interior end-of-life breach (Table 3.20)

As discussed by Weber et al. (1979, 04) global heating was quite severe in the P41 and P42 subassemblies. Weber et al. suggested that the general coolant over-temperature is related to assembly hardware design--namely EBR-II hardware with segmented liners (Figure 3.75). They postulated that leakage through the segmented liner caused a reduction in bundle flow rate which led to global overheating. Failure in these assemblies occurred in the interior pins, as would be expected if gross global overheating (300⁰F) led to failure since overcooling of the wall channels would keep the rods at the wall at

(a) The author obtained this information is personal communications with R. Jackson of the Hanford Engineering Development Laboratory.

TABLE 3.20. Postulated Mode of Breach for HEDL
Mixed-Oxide Fuel Pin

- LOCAL CLADDING HOT SPOT
 - PIN-TO-INSULATED DUCT CONTACT
 - P23A-37(271)
 - P23B-1A(297)
 - P14-26(271)
 - PIN-TO-PIN CONTACT
 - P5A-1
 - P5B-17
 - P14-29(144)
 - P23B-73E(74)
- LOCAL CLADDING HOT SPOT PLUS GLOBAL OVERHEATING
 - P-12AA-63K(166⁰F)
 - P-12AB-11B(365⁰F)
 - N/E-122(396⁰F)
- GLOBAL OVERHEATING
 - P41-A3R(318)
 - P41-B10
 - P42-B71(214)
- FUEL CLADDING MECHANICAL INTERACTION
 - P42R-D1 91%
 - P42R-D2 SMEAR
 - P43-D77 DENSITY
- CLADDING/WIRE-WRAP WEAR PLUS GLOBAL OVERHEATING
 - PNL10-14(60⁰F) **
 - PNL11-39(90⁰F)
- CAUSE NOT SURE
 - N/E-054
- IN PROGRESS
 - P23B-1A
 - ETC.

*PREPARED WITH THE HELP OF R. JACKSON, D. CHAN
AND J. WEBER (HEDL)

**IRRADIATED IN 61-PIN ASSEMBLY WITHOUT PROBLEMS,
BUT THINNING AND WEAR OF CLADDING OCCURRED

FIGURE 3.75. Cross Section of MKE-190 Assembly

a lower average temperature. Other assemblies irradiated in similar EBR-II hardware were PNL-10, PNL-11, P-12AA, and N/E. Global overheating of pins PNL-10-14, PNL-11-39 has been postulated with the breach at the cladding wear mark. It is important that the excess cladding temperature (overpredicted temperature) for PNL 10-14 and PNL 11-39 was estimated to be 60⁰F and 90⁰F. Failure may have occurred because pins PNL 10-14 and PNL 11-39 were irradiated in a 61-pin assembly before reconstitution in a 37-pin assembly there showing moderate to severe wear and thinning of cladding because of flow-induced vibrations. The 61-pin assembly was contained in a solid duct.

In the case of pins P-12AA-63K and N/E-122, Weber et al. postulated that restrictions of flow in the interior of the subassembly resulted in global overheating as suggested by the high-temperature microstructure observed around the breached pins and their neighbors. The excess cladding temperatures for these pins were 166⁰F and 396⁰F, respectively. Weber et al. write, "The condition of local cladding hot spot found for these two pins is not prototypic of that found in most of the other pins examined in these subassemblies. The severe pin displacements resulting in pin-to-pin and pin-to-duct contact appear to be related to the reconstitution of these subassemblies." In addition they found that the wires in assembly P-12AA were out of position on pins adjacent to P-12AA-63K. The same may have been true for pin P12AB-11B, which showed an excess cladding temperature of 365⁰F. This pin was in an assembly with an associated segmented liner which may have been responsible for the

global overheating. Weber et al. postulated that local heating of the pin occurs because of pin-to-pin contact over a relatively long distance. This was possible, as it was found during post-irradiation examination that the wire was straight over a large section of the pin.

Pins 41-A3R, 41-B10, and 42-B71 breached because of fuel-cladding mechanical interaction (FCMI) which occurred relatively early in life due to a higher smear density of the fuel. These breaches are distinguishable from breaches caused by hot spotting since they show up as longitudinal cracks around the circumference of the rod.

An interesting observation is the hot spot excess cladding temperatures of 271⁰F, 297⁰F, and 270⁰F found in pins P23A-37, P23B-1A, P14-26. The hot spot faced the duct, which indicated possible pin-duct contact. Failure is postulated to have occurred because of pin insulated duct contact. The associated hardware with the first two pins was a bypass liner. The hot spot excess temperatures are large and of the same order of magnitude as for the pin-to-pin contact. Possible reasons for this are discussed later.

The excess cladding hot spot temperatures for interior pins P14-29 and P23B-73E were 144⁰F and 74⁰F, respectively.

Local cladding overheating that leads to a hot spot probably occurs when the wire wrap does not prevent pin-to-pin, near pin-to-pin, or pin-to-duct contact. As indicated by Weber et al. (1979, 04) "as this condition exists at many points along the fuel pin a driving force must have been present to cause the contact. Radial temperature gradients, as much as 212⁰F on corner pins, result in postirradiation pin bowing of several centimeters in exterior and to a lesser extent in interior pins when the restraint of the duct is removed. During reconstitution and the subsequent irradiation, pin-to-pin forces resulting from the bow can cause pins to be displaced from their normal positions while still retaining the basic hexagonal arrangement." When adjacent pin-to-pin or pin-to-duct contact occurs, local starvation of flow can occur with the subsequent formation of a relatively large laminar sublayer which results in a significant loss of cooling. Displacement of pins from their normal positions can significantly increase the radial temperature gradients across the pins and induce further pin bow.

The combined effect of loss of global and local cooling are responsible for the large excess cladding temperature (200 to 300⁰F) found in pins irradiated in EBR-II. Unreconstituted bundles within solid ducts in which wires and rods are not displaced from their normal positions may still show excess temperatures due to pin-to-duct contact, nonuniform accommodation of bundle built-in porosity (Leteinturier et al. (1979, IAEA) etc.) but these are expected to be much less than 300⁰F. Based on parametric studies and improved modeling, an upper boundary to the excess cladding temperature can be found. An investigation should be initiated to establish the important parameters governing hot spot excess cladding temperature and to find an upper boundary to this excess cladding temperature. It is important to note that EBR-II driver assemblies and the Phenix assemblies have had few breaches and have gone to relatively high burnups. One of the parameters significantly different in these assemblies, in contrast to the HEDL assemblies tested in EBR-II, is the 6-in. wire-wrap pitch. Parametric studies of the type mentioned earlier would show the beneficial effect of the reduced wire-wrap pitch in reducing hot spot cladding excess temperature.

A Possible Reason for Observed High Cladding-to-Wall Contact Temperatures

A higher temperature might be expected at the point of contact between two heat-generating rods than at the point of contact between a heat-generating rod and a relatively cold, non-heat-generating wall. However, the explanation below plausibly argues why contact temperatures might be of the same order of magnitude for the two types of contact. This needs to be experimentally or analytically verified.

When two surfaces contact, the contact temperature depends on several parameters, which include the initial temperatures of the two surfaces, their physical properties, the contact area, the heat removal around the area of contact, and the heat generation rate. The contact area depends on the force acting between the two surfaces among other parameters. The heat removal capability would depend upon the thickness of the laminar sublayer in the immediate vicinity of the contact. Within this laminar sublayer the heat transfer

is purely by conduction. Parametric calculations performed by Khairallah et al., (1979, IAEA) show a high sensitivity of the cladding surface hot spot to the following two parameters:

- contact width between claddings (or cladding and wrapper)
- thickness of the assumed laminar layer.

"This sensitivity is illustrated in Figure 3.76, corresponding to a cladding-to-cladding contact and to Phenix pin diameter and maximum linear power. It is presently postulated that the more representative case for Phenix is the "small laminar layer" and contact width of 1/2 to 1 wire diameter. In these conditions, hot spots of the order of 100°C (above the nominal temperature) at the central (interior rods) positions and core midplane could be expected." Three-dimensional calculations show that axial conduction effects reduce this by 15 to 20% for a contact height of five wire diameters. If the contact height is less than five wire diameters, the reduction in hot-spot temperature due to axial conduction is larger. Thus hot spots created by cladding-to-cladding contact of the order of 50 to 80°C (80-144°F) in the central regions of the bundles can be expected. This temperature increase from nominal at the hot spot is of the same order of magnitude as found at the location of the breached cladding in the interior pins of the HEDL bundles tested in EBR-II (when global overheating was not a problem).

For a cladding-to-wall contact, the same assumptions of laminar layer thickness and contact width may not hold good as for interior rods. In the central regions of the bundle considerable turbulence, mixing due to wire wrap, etc. would reduce the size of the laminar sublayer as compared to the wall region. Moreover, temperature gradients across the pins at the wall are higher, and the subsequent larger pin bowing may make the cladding-to-wall contact loads larger, resulting in a larger contact area. A larger laminar sublayer and an increased contact area can compensate for the fact that the wall does not generate heat. Thus cladding-to-wall contacts might lead to local hot-spot excess cladding temperatures in the same range as those for cladding-to-cladding contact.

FIGURE 3.76. Hot Spot Due to Infinite Length

It is recommended that experiments be carried out to evaluate the extent of the assumed stagnant or laminar sublayer around the contact using a laser-Doppler velocimeter. Subsequently fine measurements of temperature distribution are required around simulated contacts.

3.3.5 Out-of-Pile Tests

Several out-of-pile tests that have recently been reported are described below.

Figure 3.77 shows a heated 7-pin wire-wrapped bundle with one corner pin larger than the rest which has been tested in sodium at Casaccia-Laboratory Valutazione Nocciolo e Canali (CNEN) (Falzetti et al. 1979, IAEA). The figure shows the temperature distribution in the assembly with and without the distortions. It appears that the effect of the distortion was to increase the

FIGURE 3.77. Deformed Pin Effects on Temperature Map vs Velocity

pressure drop across the bundle (Figure 3.78) significantly. This effect was explained by the authors as being due to a "reduction in the peripheral gaps near the deformed pin and the consequent disturbance induced in the helical recirculation around the duct walls." The temperature distribution shown in Figure 3.77 agrees with this hypothesis. Following the rotation of the wire,

FIGURE 3.78. Pressure Losses (Experimental) in 7 Pin Bundles

a temperature decrease was observed in the subchannel before the obstacle, where the axial velocity is expected to increase. In channel 13 an increase in temperature of the deformed bundle was observed, showing that there is flow from the inner, hotter regions to the cooler wall channel 13. This perhaps is the path taken by the swirl flow due to the reduction in crossflow area near the swollen pin.

Figure 3.79 (Hofmann et al. 1979, IAEA) shows the change in temperature distribution in a heated 19-pin grid-spaced bundle when one of the rods was intentionally displaced from its normal position, as can be expected for a bowed rod in an assembly. Rod 12 showed a peak temperature rise of about 13°C (23°F) above its nominal value when the adjacent rod 4 was displaced

FIGURE 3.79. Cladding Temperatures for Different Bowing Positions

toward the middle of the channel formed by rods 3, 4, and 12. Surprisingly, rod 3 did not show a temperature distortion of the same magnitude. This could have been partly due to the position of the wire wrap at that axial level.

Figure 3.80 shows the circumferential temperature distribution measured around a peripheral rod in a 19-pin assembly cooled by liquid sodium. The circles show the measured rod surface temperatures. Figure 3.80 also shows, in dashed lines, the predictions of the global subchannel code ARTIS for a uniform Nusselt number around the rod. The temperature difference predicted across the pin is much smaller (by a factor of 2) than measured. When the local Nusselt number around the rod was used ($Nu = 7$ in the wall channel and $Nu = 14$ in the central channels) it was found that the maximum temperature difference predicted across the rod further decreased. This clearly shows the limitations of subchannel (global) analysis codes in predicting local temperatures. This result was not expected since the global codes are unable to predict local temperatures for several reasons; the main reason is that subchannel analysis only defines an average value and not its variation within a subchannel.

- EXPERIMENT TE A[], CORRECTED TO CLADDING SURFACE
- MEAN SUBCHANNEL TEMPERATURE (ARTIS-CODE)
- MEAN ROD SURFACE TEMPERATURE $Nu = 12$ ① ② ③
- MEAN ROD SURFACE TEMPERATURE $\begin{cases} Nu = 7 & \text{SIDE CHANNEL ①} \\ Nu = 14 & \text{CENTRAL CHANNEL ① ② ③} \end{cases}$

FIGURE 3.80. Aximuthal Temperature Variation of the Side Rod Cladding; Comparison Between Local Measurements--Global Calculations

The current practice for predicting local cladding or coolant temperatures is to follow two steps: 1) the subchannel analysis codes are used to determine the average subchannel flow rate and temperature, 2) a two-dimensional local prediction within the isolated subchannel is made by a distributed parameter code. This technique would work if the flow is fully developed locally in the subchannel at the elevation where the localized predictions are made and if the subchannel mixing (turbulence and conduction) parameters are adjusted to give the same averaged results as the more detailed analysis.

Pin contact experiments have been conducted in a 7-pin heated bundle by Haga et al. (1974, E04) and Collingham et al. (1972, E04). Measured hot-spot cladding temperatures were found to increase by 30 to 50⁰F at the point of contact. For three and seven pin contacts (Miyaguchi 1979, IAEA) the contact temperatures can be 100⁰F to 212⁰F, respectively, which will still be well below the boiling point of the coolant. An experiment has been carried out by Bloeman et al. (1976, E02) to verify the BOBL subchannel analysis code (Boggard et al. 1976, 04) for pin bowing and grids.

3.3.6 Analytical Methods

A number of subchannel analysis methods have been developed for predicting flow and temperature distributions in bundles of known distortions. These include the methods of Ohtsubo (1974, 04), Boggard (1976, 04), Leteinturier (1978,04). Three-dimensional methods of determining the deformation of fuel pins within a wire-wrapped assembly have been reported by Miki (1979, 04) and Sakai (1978, 04), who perform an iterative mechanical thermal-hydraulic calculation using the subchannel analysis COBRA code. They have found by analysis of a 169-pin bundle that there is no direct contact between cladding surfaces in the fuel assembly; the engineering hot-channel factor accounting for fuel pin deflection is around 1.023 to 1.035 at zero irradiation, depending upon power gradient and further increases to 1.045 at 500-days of irradiation. An interesting result is that the peak coolant temperature in the inner flow channels of a deformed bundle is below the peak coolant temperature in a completely tight bundle; i.e., a bundle in which all the fuel pins are compressed throughout their entire length. The use of a subchannel analysis method as used by Miki and Sakai may have led to optimistic results for temperature gradients across pins.

3.3.7 Limitations of Subchannel Computer Programs

The subchannel analysis codes have certain inherent weaknesses due to the assumptions and approximations made. These are described in detail in the section on global hydrothermodynamics and in Section 2.0 of this report. Even for bundles with uniform geometries the subchannel codes do not predict the temperatures well in regions of large temperature gradients--such as in the vicinity of the duct wall. The mixing coefficients are dependent upon geometrical and thermal boundary conditions and corrections have to be made (Ginsberg and France 1973, N4) in order to adequately model the lateral transfer of heat between channels. The normal assumption of complete mixing of incoming enthalpy within a calculation step could lead to serious errors which will be especially important in the analysis of deformed bundles. As expected, circumferential cladding temperature gradients are underpredicted by subchannel analysis (Figure 3.80), even for nominal geometries and this could be worse for distorted geometries. This is not unexpected but needs to be emphasized.

As a result of these and other approximations discussed previously, the validity of subchannel analysis for thermal-hydraulic analysis of distorted bundles is doubtful. However, since the distributed-parameter methods have excessive computational and storage requirements, the mixed approach suggested by Nijssing and Eifler (1978, 03) may perhaps be the gap between subchannel and distributed-parameter analysis. Figure 3.81 shows some of the major advantages and disadvantages of the Nijssing and Eifler approach as compared to the global and local approaches. Figure 3.82 shows a typical sub-subchannel division for a 60° sector of a 7-pin bundle that is used in the mixed approach. This method is relatively fast. However, it is important to model the spacer system in the method, and the method needs to be further developed. The method seems to be easily adaptable for use on computers and adaptable to an iterative thermal-hydraulic/mechanical design procedure for simultaneously determining the flow and temperature fields as well as the distorted geometry in the rod bundles. The advantage of the method appears straightforward. The same wire-wrap mixing coefficient used in the global analysis can perhaps be used for this method (in distorted bundles this will be a conservative assumption because mixing will be enhanced by the distortions). Yet the assumption

FIGURE 3.81. Mixed Approach for Analysis of Distorted Bundles

of complete mixing of enthalpy in a calculation step (numerical diffusion) will not lead to errors as large as it does for global codes.

Any code for analysis of distorted bundles should include the following additional features for the reasons discussed below. The inertial terms in the axial momentum equation should be included in the formulation. This is especially important for distorted bundles since neglecting the inertial terms

FIGURE 3.82. 60° Sector of 7 Rod Bundle

would lead to greater extremes of flow starvation and overcooling. Hydraulic diameter flow splits for determining flow distributions is out of the question. A transverse momentum equation should be included in the formulation, otherwise the axial pressure drops could be underestimated. A model for fluid-to-fluid shear should be included. A boundary value form of solution of the equations is recommended although not necessary. This is to allow downstream effects to propagate upstream. The reason this is not a necessary requisite is because the downstream effects will propagate upstream by one- or two-calculation steps for most distorted geometries (except for large distortions such as severe corkscrewing of the bundle due to bundle-duct interaction). In view of the other approximations in the method, this may be a tolerable assumption and a marching method may be acceptable. It may be possible to modify existing subchannel codes such as COBRA-IV to a form similar to Nijssing's method by incorporating a two-dimensional wire-wrap model and by further subdividing the channel.

Other major problems in predicting distortions in bundles is the large error bars associated with the swelling and irradiation-induced creep curves. Unless these are determined more accurately, predicted shapes of the distorted bundle can be widely different, and the accuracy of the results would be determined not by mechanical or thermal-hydraulic considerations but by materials properties. It is recommended that as the accuracy of determining material properties improves, improvements in mechanical and thermal-hydraulic methods

should also be made. Techniques to confirm the analytical predictions experimentally in an operating reactor assembly or soon after the removal of the assembly from the reactor should also be developed since the distorted shape may change while the assembly is cooling down for geometrical measurements.

3.4 NOMINAL BUNDLES, NON-NOMINAL (TRANSIENT) CONDITIONS INCLUDING MIXED AND NATURAL CONVECTION

3.4.1 Umbrella Transients for Reactor Design

Different types of transients are considered in the design of an LMFBR power-generating system. Some of these transients may not have a serious effect on the core and are not within the scope of this report. Hypothetical, unprotected, safety-related transients which may lead to fuel melting, fuel-coolant interaction, core meltdown, etc. are also not within that scope. Protected transients that can have a direct impact on fuel performance and on core design will be addressed here. Other transient reactor design events are categorized as normal, upset, emergency, and faulted (Tang et al. 1978, Book) according to their likelihood of occurrence. The frequency of occurrence and classification of events is established by the designer, based on industrial and nuclear experience and the special characteristics of LMFBR design.

As indicated by Tang et al. (1978, Book), core design limits are used to evaluate the consequences of various transient events: "If a criterion such as maximum cladding temperature is violated, design changes must be made, e.g., lowering plant protection system (PPS) settings or increasing scram insertion rates to attain acceptable design margins."

Two basic transients could cause dynamic loading of the cladding as opposed to long-term creep effects associated with steady-state operations. The first transient results from undercooling or slow overpower transients. The second involves fast overpower transients of the type that could produce fuel cladding interaction. All upset, emergency, and faulted events that result in cladding temperature increases can be categorized as one or the other of these transients.

3.4.2 Analytical Methods

Table 3.21 presents the upset, emergency, and faulted transients that umbrella all other events in the respective design categories. Three of the 15% overpower events per year of design life should envelope the damage due to the frequency of all the anticipated transients. Only a single occurrence at the worst time in assembly life needs to be considered for the emergency and faulted transients. Table 3.21 shows that during the transients the assemblies are subjected to a wide range of varying power and flow conditions, ranging from laminar to turbulent flow and natural-, and forced-, and mixed-convection regimes of operation.

A number of methods have been developed for the transient global (sub-channel) analysis of single assemblies within an LMFBR core. These include the work of Domanus et al. (1977, N4), Sha et al. (1977, N4) Wheeler et al. (1976, N4), Yao et al. (1977, N4), Yung et al. (1976, N4), Chen et al. (1978, 04), Rowe (1972, 1973, 04), and Saito (1977, 04). These computational methods subdivide the single assembly being examined into a large number of channels and solve the governing equations with prescribed boundary and initial conditions to solve for the transient thermal-hydraulic field within the assembly. However, the power and flow into the core are prescribed functions of time. The single-channel method of Miller et al. (Tang et al. 1978, Book, Ch. 6) includes the reactivity feedback effects and thus calculates the power variation with time but requires a transient flow-forcing function that is obtained from plant codes. Since the computational capability is desired over a wide range of transient conditions--laminar, transition, turbulent flows, and natural, forced, and mixed convection--it is necessary to assure that the constitutive equations or equivalent correlations are available for use in the codes. For example, one needs to know if the steady-state friction factor and Nusselt number can be used during transients or if the friction factor or the shear stress distribution is significantly affected by the simultaneous effect of transient and buoyancy conditions.

Kawamura (1976, 03) has numerically solved for the transient turbulent flow and temperature field in a circular tube subjected to a step change of pressure gradient. Transient variation of the friction factor and the heat

TABLE 3.21. Definition of Umbrella Transients

UPSET EVENT

U-2b^(a) Uncontrolled Rod Withdrawal from 100% Power

An uncontrolled withdrawal of one control rod causes the reactor power to increase instantaneously from 100% to 115% (just below the high-flux trip point). A manual reactor trip is assumed after 5 min. Sodium flows are maintained at initial values until the trip occurs, and the feedwater and steam flows are increased appropriately for the increase in reactor power. Three U-2b events per year of operation should contain the anticipated frequency of transients.

EMERGENCY EVENT

E-16^(a) Three-Loop Natural Circulation

From initial conditions of full-power operation, complete loss of forced-sodium circulation in all loops is assumed. A reactor/turbine trip is initiated by primary pump under-voltage relays. Steam pressures increase, causing some relief of steam through the power-operated relief and safety valves. Sodium pumps coast down and stop, and natural circulation flow is established in all sodium loops. Auxiliary feedwater flow is established from the auxiliary feedwater portion of the steam generator auxiliary heat removal system based on low-drum level signals. The turbine-driven auxiliary feed pump takes suction from the protected water storage tank to maintain drum levels. Terminal conditions include decay heat removal through the steam generator auxiliary heat removal system (SGAHRs). For a component with less than a 4-year design lifetime, only one three-loop natural circulation event needs to be considered since it is an emergency event.

FAULTED EVENT

SSE^(a) Safe Shutdown Earthquake

At time equals zero, a loss of offsite power is assumed to occur due to the seismic disturbance, and the pumps start coasting down to pony motor flow. Relative to set points of the Plant Protection System trip functions, a step reactivity insertion is assumed to occur at the worst time after power loss (due to core compaction/stick slip effects). The delay in control system insertion speed is factored into the shutdown performance. Components need consider only a single SSE occurrence, since it is a faulted event.

transfer coefficient are obtained. Figure 3.83 shows the variation of the friction factor ratio f/f_{st} for various initial and final Reynolds numbers. In case of acceleration, f/f_{st} increases remarkably with increase of the ratio Re_1/Re_0 , becoming as high as 7 for a severe transient. At Reynolds number lower than 10^4 , this ratio could be higher. This can be important during a natural-circulation mode of core cooling where fuel temperatures are becoming increasingly large and the pumps are turned on to reduce the rate of temperature increase. In deceleration, f/f_{st} does not differ much from unity. Figure 3.84 illustrates the transient velocity profiles calculated by Kawamura in a circular tube in accelerated and decelerated flows. Z is a non-dimensional time defined as

$$Z = 4 \left(\frac{f_1}{D} \right) \bar{u}_1 t \quad (62)$$

where D is the tube diameter, f_1 and $\bar{u}_1$ are the final steady-state friction factor and mean coolant velocity, respectively, and f_{st} is the steady-state friction factor. Kawamura found the momentum equation as used in the subchannel analysis codes to be approximately applicable to both the accelerated and the decelerated flow. He also found, for the simplified circular tube model he used, that the transient heat transfer coefficient in decelerated flow is slightly larger than its steady-state value, whereas in accelerated flow the transient heat transfer coefficient decreases well below its steady-state value if the heating wall has no heat capacity. When the wall has a large heat capacity, the heat transfer coefficient does not decrease much. Minkowycz and Stein (1973, 04) have also developed an improved method for calculating liquid-metal turbulent heat transfer with time-dependent heat fluxes. They propose the relationship:

FIGURE 3.83. Variation of the Friction Factor

FIGURE 3.84. Transient Velocity Profile Compared With the Steady-State Profile

$$T - T_c = \frac{q''}{h} + (\text{a correlation factor for transients})$$

$$\begin{aligned} \text{correction factor} = & \frac{D^2 R_1}{2k\beta} \left(\frac{\partial q''}{\partial \ell} + \frac{1}{\bar{u}} \frac{\partial q''}{\partial t} \right) \\ & + \frac{D^3 R_2}{2k\beta^2} \left(\frac{\partial^2 q''}{\partial \ell^2} + \frac{2}{\bar{u}} \frac{\partial^2 q''}{\partial \ell \partial t} + \frac{1}{\bar{u}^2} \frac{\partial^2 q''}{\partial t^2} \right) + \dots \end{aligned} \quad (63)$$

where

$\partial q''/\partial \ell$ = axial gradient of heat flux

$\partial q''/\partial t$ = change of heat flux with time

D = hydraulic equivalent diameter

$\bar{u}$ = flow velocity

R_1, R_2, β = dimensionless coefficients that are functions of Reynolds and Prandtl numbers and duct shape but are independent of space and time.

For the design category of transients, the correction term can be neglected (Tang et al. 1978, Book).

3.4.3 Time Constants Associated with Umbrella Transients

In order to understand the response of the fuel, cladding, and coolant during an umbrella transient and to understand the important geometrical and physical parameters on which this response is dependent, it is necessary to examine the various time constants associated with the transients. The various time constants associated with the transients include the following:

τ - time constant of the fuel pin

p - pump half time, i.e., time required for total core flow to reach half its initial value

τ_p - power period, i.e., assuming for simplicity the power production is given by $P = P_o e^{t/\tau_p}$

τ_{in} - period of inlet temperature change assuming inlet temperature change is given by $T_{in} = T_{in(o)} e^{t/\tau_{in}}$

During a U-2b transient (Table 3.21), the power increases instantaneously from 100 to 115%, but the coolant flow rate does not change. For the first 300 sec of the transient, τ_p does not enter the picture, whereas the effect of inlet temperature is not important initially because it changes slowly. If the power increases instantaneously, τ_p is very small relative to τ , the time constant or response time of the fuel pin. In this case the fuel heats up almost adiabatically since the pin cannot respond fast enough. On the other hand, if the power changed slowly ($\tau_p \gg \tau$), the fuel would be in thermal equilibrium because the increase in power would be easily accommodated by a quick response of the fuel, i.e., by a quick release of heat to the coolant.

Of greater interest is the E-16 transient (Table 3.21), which has been discussed in detail by Coffield et al. (1979, IAEA). The normal mode of core heat removal upon reactor shutdown is by forced circulation from AC-powered pony motors which have emergency backup power from batteries and/or diesel generators driving the primary pumps. However, most LMFBR plants have been designed with the added capability of adequate cooling by means of natural circulation. The thermal driving head developed by the thermal center of the intermediate heat exchanger (IHX) is elevated above that of the core; that plus the respective elevation differences in the intermediate loops and steam generator system provides an inherent emergency coolant flow.

During the early transition into the core natural-convection cooling mode, the effect of dynamically approaching low flow with worst-case decay heat load results in a power-to-flow ratio greater than one. Consequently, core thermal temperatures increase and the natural-convection phenomena associated with inter- and intra-assembly flow redistribution due to different thermal heads and hydraulic characteristics of the core assemblies become significant. In general, the core thermal head is as large as the form and friction loss across the core at about 3 to 4% of full flow. Coupled with the flow redistribution, significant heat redistribution on an inter- and intra-assembly basis occurs throughout the core because of large temperature differentials and a significant heat transport time (low power assemblies can have a transport time of over 20 sec). These effects (i.e., natural-convection flow and heat

redistribution) were found to significantly reduce maximum core temperatures in the EBR-II natural-circulation experiments (Singer et al. 1978, E04; also 1976, 1977, 04).

This significant attenuation of maximum core temperatures has important core design implications. The natural-circulation transient is a worst-case, duty-cycle event in the emergency category and limits LMFBR fuel and blanket rod lifetime capability. Any decrease in the predicted maximum core temperatures during the natural-circulation transient, will provide significant economic benefits by allowing increased fuel and blanket burnups. In addition, the flattening of the radial core temperature profile will greatly reduce assembly cross-duct temperature gradients (and thus reduce bowing), thereby reducing core transverse forces on the core restraint system during the natural-circulation transient. These aspects, coupled with the fact that lower maximum core temperatures always enhance reliability by providing greater margins for design and safety limitations, help emphasize the importance of taking into account all the coupled phenomena that occur in the core while operating in the natural-convection cooling mode.

Figure 3.85 shows a typical core power and core total flow variation during an E-16 event. The coolant temperature response will depend upon τ , t_p , τ_{in} , and the local power-to-flow ratio at any time. Since the rate of change of inlet coolant temperature is small and occurs several minutes into the transient, it is not considered here. For a typical LMFBR core subjected to an E-16 event at full power.

$$\frac{d}{dt}(\Delta T_{coolant}) \sim f[\tau, t_p, \frac{P(t)}{W(t)}, \tau_{in}] \quad (64)$$

If the fuel time constant is very small compared to pump half time, which is the case for fuel assemblies, the assembly coolant temperature rise would follow the trend of the power-to-flow ratio, $P(t)/W(t)$, which causes an initial decrease in cladding temperature followed by a subsequent increase to a peak value and then a further decrease (Figure 3.86). For a blanket assembly, τ is of the same order as t_p ($\tau \sim t_p$). As the coolant flow decreases, the heat

FIGURE 3.85. Power and Flow Rate for Natural-Circulation Transient

flux out of the rod may or may not decrease at the same rate, and so the cladding temperature may go up or go down (Figure 3.87). The $P(t)/W(t)$ ratio includes the effect of interassembly heat transfer, interassembly flow redistribution, and any other phenomenon which may affect $W(t)$ or $P(t)$. For example, the control system shutdown worth affects $P(t)$ (Figure 3.88). Thus the coolant and, therefore, the cladding temperatures would depend upon the parameters that affect τ , t_p , and $P(t)/W(t)$. These parameters are defined as follows, to study the thermal response of an assembly:

FIGURE 3.86. Natural-Circulation Temperatures for Fuel Assembly Hot Rod

$$\tau = f(R, \rho cV)$$

$$R = f(k_{fuel}, h_{gap}, d)$$

$t_p \sim I_{pump}$, system fluid inertia, loop frictional characteristics

$P(t) \sim$ control system shutdown worth, interassembly heat transfer, decay power

$W(t) \sim$ interassembly flow redistribution, I_{pump} , system fluid inertia, pump--rotor damping forces, residence time of fluid

where R , the thermal resistance of the fuel pin, depends primarily upon fuel thermal conductivity (k_{fuel}), gap conductance (h_{gap}), fuel rod diameter (d),

FIGURE 3.87. Natural-Circulation Temperatures for Radial Blanket Assembly Hot Rod

FIGURE 3.88. Effect of Shutdown System Worth on Fuel Hot Rod Maximum Temperatures for Three-Loop Natural-Circulation Transient

fuel rod density (ρ), heat capacity (c), and volume (V). Coffield et al. (1979, IAEA) have described the sensitivity of the hot-channel temperatures to these parameters in detail. It is important to understand that during the transient for times much less than t_p , buoyancy effects and interassembly heat transfer effects are not important, and during this period the total core flow rate depends largely upon the pump inertia, the loop frictional characteristics, and to some extent on the fluid inertia in the primary system. At

times $t \sim t_p$, buoyant head in the primary loop may start to become important, whereas at $t \gg t_p$, buoyancy controls the flow through the core. At $t \gg t_p$, the flow in the hot assembly is affected by buoyancy-induced flow redistribution within the core and by interassembly heat transfer. For a typical LMFBR core (CRBR) the fuel time constant is approximately 2 sec, the blanket time constant is approximately 15 sec, and the pump half time is about 7 sec.

The sensitivity of the hot-channel temperatures to the various parameters in Equation 65 can be seen more explicitly as described below.

Protected duty-cycle transients affect fuel and blanket lifetime within the core. Design experience^(a) has shown that if the cladding temperature is held below 1500°F (maximum transient, 2 temperature) for the upset transient (U-2b) and below 1600°F for the emergency and faulted events (E-16 and SSE), failure has a small probability of occurring when the rod structural integrity is evaluated by various modeling approaches (e.g., CDF^(b)).

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- (a) The author obtained this information in personal communication with R. Markley of Westinghouse, Advanced Reactor Division.
 (b) Roth et al. (1978, N4)

$$CDF = \int \frac{dt}{t_R(t)}$$

The combined steady-state and umbrella transient contributions to CDF should not exceed 1.0 during assembly life in the core. CDF = 1.0 implies cladding failure. The value $t_R(t)$, the time to cladding rupture in hours, is given by a Larson Miller correlation

$$LMP = T(20 + \log_{10} t_R)$$

where T is the cladding temperature in kilo-°R and

$$LMP = 45.522 - 4.349(\log_{10} \sigma)^2 - 0.185(\log_{10} \phi t) \sigma - 0.101(\log_{10} \phi t)^2 \sigma \\ + 1.448 \times 10^{-3} (\log_{10} \phi t) \sigma^2$$

where σ is the cladding loop stress in ksi and ϕt is the fast neutron fluence in 10^{12} n/cm².

Figure 3.89 (Coffield IAEA-11, 1979) shows the effect of the various parameters discussed earlier, such as fuel thermal conductivity and rod diameter, on the maximum steady-state operating temperatures such that the 1600°F limit is not exceeded during an E-16 event. As with flow coastdown, rod inertial effects influence the transient temperatures. The degradation of the low-thermal-conductivity fuel and of blanket rods above rod diameters of 0.46 in. OD is due to stored heat effects (coupled with flow coastdown half time, t_p). For the high-thermal-conductivity fuels (e.g., carbide and nitrides), the performance is somewhat worsened over that for the oxides in the diameter range less than 0.46 in. This is because the rod more closely follows the power-to-flow ratio and because the lack of inertial effects delay the temperature rise to values indicative of early shutdown powers. Figures 3.90 and 3.91 from Coffield and Miller (1978, 04) show the overall transient limit envelope of allowable operating temperatures for low-thermal-conductivity and high-thermal-conductivity fuel rods for specified reactor operating conditions.

Based on previous discussions, it is clear that the rate of pump coastdown is an important parameter for the umbrella transients. The total pump coastdown time is also empirical data that is of great importance with respect to maximum core temperature (Additon et al. 1977, 04). "The principal explanation for the sensitivity is that reactor decay power decreases significantly in the first 2 to 3 min and thus the earlier the pumps stop, the higher is the decay power level which natural convection must accommodate. The early portion of the pump coastdown (100% to 10%) can be calculated readily by available stored energy, the pump operating point, and the affinity laws applicable to the pump's characteristic hydrodynamic curves. The tail end of the coastdown, however, is dominated by relatively unquantified frictional effects. The coastdown time for FFTF is sufficiently long that the driver fuel stored heat is removed from the core while forced-circulation transient temperatures are insensitive to parameters affecting stored heat such as gap conductance or fuel conductivity. If the initial rate of flow coastdown is large, the total coastdown time may have to be increased."

The flow coastdown rate is determined by a compromise between the improved performance of fuel and blanket assemblies (Figures 3.86 and 3.87), which

FIGURE 3.89. Transient Operating Limit Curve for Low and High Thermal Conductivity Fuels at 16 kW in Peak Power Generation for Natural-Circulation Event

FIGURE 3.90. Overall Transient Limit Envelope of Allowable Operating Temperatures for Low Thermal Conductivity Fuel Rods at 16 kW/ft Peak Power Generation (650°F Inlet Coolant Temperature and Steady-State Hot Channel Coolant $\Delta T \leq 650^\circ\text{F}$)

FIGURE 3.91. Overall Transient Limit Envelope of Allowable Operating Temperatures for High Thermal Conductivity Fuel Rods at Peak Power Generation (650°F Inlet Coolant Temperature and Steady-State Hot Channel Coolant $\Delta T \leq 650^\circ\text{F}$)

requires a smaller flow coastdown rate, and the consideration of thermal shock to the upper internals, which requires a larger flow coastdown rate. The blanket assemblies are particularly sensitive to the flow coastdown rate.

Existing methods (Tang et al. 1978, Book) of estimating hot-channel temperature in the core during a duty-cycle transient do not take advantage of the various thermal-hydraulic phenomena which tend to mitigate core-wide transverse temperature gradients. Figures 3.86, 3.87, 3.92, and 3.93 do not include the effect of the various thermal-hydraulic phenomena which mitigate peak coolant temperature. These phenomena include interassembly heat transfer and flow redistribution due to buoyancy and intra-assembly heat transfer and flow redistribution. The predicted temperatures in Figures 3.86, 3.87, 3.92, and 3.93 are therefore much higher than if the aforementioned thermal-hydraulic phenomena were taken into account. Comparison of predicted temperature without the inclusion of these phenomenon to EBR-II peak transient temperature measured during an E-16 type of event, shows that the single-channel predictions can overpredict peak coolant temperature in the core by as much as 20 to 30% of the core coolant mixed-mean temperature rise at steady state.^(a) For this reason the COBRA-WC and CORTAN (Khan, George, and Wheeler 1979, IAEA) computer programs have been developed to take into account on an assembly-to-assembly basis, the effect of interassembly flow redistribution, interassembly heat transfer, intra-assembly flow, and temperature distribution. These thermal-hydraulic phenomena are all coupled and cannot be taken into account on an individual basis. Figure 3.92 from Coffield et al. (1979, IAEA) shows the calculated typical magnitude of interassembly flow redistribution in fuel and blanket assemblies, neglecting interassembly heat transfer and intra-assembly flow and heat redistribution. It is significant to note that the hot fuel and blanket assemblies (which would contain hot channels) receive enhanced cooling during their peak temperature period as can be seen by examining the timing of the peak temperatures in Figure 3.93, (Coffield et al. 1979, IAEA), which predicts the maximum temperatures with no assistance from flow and heat redistribution.

FIGURE 3.92. Typical Interassembly Flow Redistribution During Natural-Circulation Transient

FIGURE 3.93. Typical Maximum Hot Rod Cladding Temperatures for Three-Loop Natural Circulation for No Flow Redistribution Case

3.4.4 Transient Mixed/Natural Convection Data

A considerable amount of work has been done in combined natural- and forced-convection coolant flow in channels (Chato 1969, Survey) but very little experimental data is available for transient mixed/natural convection in prototypic LMFBR geometries for low-Prandtl-number fluids. An experimental study of forced and natural convection in a vertical rectangular test section during flow transients was performed by Kerian et al. (1972, E04) using NaK-78 as the working fluid. The transients investigated initiated during the forced-convection regime and terminated in a natural-convection regime. The maximum temperatures occurred during the transient intermediate regime of combined forced and natural convection. Figure 3.94 shows the test section geometry and dimensions. Thermocouples were located on the inside wall of the test section at distances of 2.68 and 7.68 in. from the start of the heated region and in the liquid at the channel centerline at distances of -0.7, 3.52, 5.55, 8.05, and 10.82 in. from the start of the heated region. Flow rates were measured at the inlet and outlet of the test section. Several rates of flow coastdown shown in Figure 3.95 were investigated. For each of the planned rates of coastdown and each initial Reynolds number, a zero heater power (isothermal) coastdown was first run in order to describe the forced-flow transient velocities.

The transient velocities in the presence of heat flux applied to the vertical test section are shown in Figures 3.96 and 3.97 for two different initial Reynolds numbers of 10100 and 5730 and flow coastdown rate "B1." The region of combined flow begins at departure from the curve $Gr = 0$, and purely natural convective flow occurs when the curves approach zero slope at about 100 to 200 sec.

The width of the combined flow region was found to increase with decreasing initial velocity and increasing Grashof number as would be expected from steady-state correlations. Velocity overshoots observed at large Grashof numbers indicate a natural-convection response to large peaking temperatures.

(a) The author obtained this information in personal communications with R. Singer of Argonne National Laboratory.

FIGURE 3.94. Test Section Geometry and Dimensions

Typical temperature data for flow-initiated transients are shown in Figure 3.98. As the Grashof number is increased, the magnitude of the temperature overshoot increases and occurs earlier. Temperature fluctuations occur during flow transient B1 (where the flow reduces to near zero in about 10 sec) in the absence of buoyancy forces ($Gr = 0$). However, the temperature fluctuations do not occur for flow transient B4, where a final steady-state forced

FIGURE 3.95. Transient Relative Velocities Without Heating

flow is maintained. Temperature fluctuations before the temperature maximum are apparently caused by Be'nard-type convection cells, produced by a temperature inversion above the test section that is caused by the rapidly increasing temperatures in the test section.

Axial temperature profiles during a typical flow transient are shown in Figure 3.99. During the initial decrease in flow, the temperature profile assumes a convex shape as the entering fluid spends relatively more time in the heated section than the downstream fluid. The thermal entrance length is continually varying as flow rate changes. Near the exit there are strong, local natural-convection effects which may or may not include the formation of Be'nard convection cells.

FIGURE 3.96. Transient Relative Velocities with Initial Reynolds Number of 10,100 and with Heating

The parameters of interest in the transient combined flow include the geometry of the test section, test section material properties, Rayleigh number, rate of flow coastdown, and initial and final Reynolds numbers.

The PNL 2x6 rod bundle test section (Quigley et al. 1977, E04) has been used to study (Bates and Khan 1979, E03) the mixed-convection thermal-hydraulic phenomena for controlled transients where the final steady-state flow rate is non-zero (i.e., forced flow is maintained). Figure 3.100 shows a cross section of the 2x6 rod bundle, and Figure 3.101 shows the elevation of the test section. Also shown in Figure 3.101 are the locations of windows at which laser-Doppler velocimeter measurements were taken. Nine windows allow velocity measurements using a laser-Doppler velocimeter. Coolant temperatures at

FIGURE 3.97. Transient Relative Velocities with Initial Reynolds Number of 5730 and with Heating

various axial levels are measured by inserting a thermocouple rake. The bundle heated length is 48 in. During mixed and natural convection, recirculation zones or Be'nard cells can form not only at the bundle exit, but also within the bundle. This can occur if there is a power skew across the bundle, which can cause significant flow redistribution and possibly flow reversal in parts of the bundle (i.e., formation of a recirculation zone).

Table 3.22 shows four typical transients for which detailed velocity and temperature distribution data was obtained within the channels of the rod bundle (Bates and Khan 1979, E03). The controlled flow rate decreased linearly during the transient. Initial and final steady-state velocity and temperature

FIGURE 3.98. Temperature Transients During Flow Transient B1 (See Figure 3.95)

distributions were measured. Figure 3.102 shows the isothermal and heated-bundle (1:0 power skew) steady-state velocity profiles at window 9 (the heated length) for a Reynolds number of 480. Significant flow redistribution occurs towards the hotter side of the bundle because of buoyancy effects. The isothermal velocity profile in the various channels (at window 9) is similar to the parabolic velocity profile at a Reynolds number of 200. There is considerably more peaking of the velocity profile toward the center of the channel at a Reynolds number of 200 than at a Reynolds number of 480. Figure 3.103 also shows the velocity profile for a heated bundle at a Reynolds number of 300. The power skew across the bundle is 1:0; i.e., only the six pins on the

FIGURE 3.99. Typical Axial Temperature Profiles During Flow Transient B1

right half of the bundle are heated. Negative flow was observed in parts of the colder channel since buoyancy drives flow toward the heated side of the bundle.

Figure 3.104 shows a comparison of data and predicted velocity profiles as function of time for channels 2, 4, and 6 for case 4 of Table 3.22. A constant power to the rods was maintained throughout the transient but the flow rate decreased linearly. The initial and final Reynolds numbers range from 730 to 320. As the bundle inlet flow decreased, buoyancy-induced flow redistribution between channels occurred within the bundle. As a result of the flow redistribution, negative flows started in channel 2 at window 8 in about 30 sec. At windows 1 through 7, the flow was still positive in channel 2.

SUBCHANNEL DIMENSIONS			
TYPE	AREA (in. ²)	WETTED PERIMETER (in.)	HEATED PERIMETER (in.)
CORNER	0.0782	1.073	0.3731
SIDE	0.1127	1.321	0.7461
CENTRAL	0.1534	1.492	1.492

FIGURE 3.100. Cross-Section of the 2 x 6 Rod Bundle Test Section

Due to recirculation, the temperature of channel 2 increased, causing buoyancy forces to retard the downward acceleration of fluid in channel 2. At about 60 sec the negative velocity in channel 2 attained a peak, and eventually in the steady-state it recovered to a small positive value. The rate of recovery of the velocity from a negative to a small positive velocity in the steady state is rather slow.

Three different models were used to predict the data using the COBRA-WC (George et al. 1979, N4) computer program. Using a zero mixing coefficient ($\beta = w' = 0$), the two models one with and the other without a cold wall model, predict close to zero velocities for channel 4 at about 50 sec. In fact channel 4 velocities were well above that. A number of possible reasons to explain this were investigated. The most plausible reason appears to be the interaction of turbulent thermal plumes superimposed on a predominantly laminar flow

FIGURE 3.101. Axial Elevation of the 2 x 6 Rod Bundle Test Section (all measurements in inches)

TABLE 3.22. Operating Conditions for Four Transient Runs

Case	Initial/Final Flow, gpm	Transient Time, sec	Power Gradient	Power Per Hot Rod, kW	Reynolds Number Initial/Final	Average Temperature Rise (steady-state), °C
1	3.08/1.08	150	1:0	0.91	1290/520	12.1
2	3.08/1.08	45	1:0	0.91	1290/520	12.1
3	3.08/1.08	45	2:1	0.91	1340/560	18.2
4	1.63/0.55	45	1:0	0.91	730/320	22.9

field, which causes mixing between channels to reduce temperature gradients between channels. The resulting channel 4 temperatures are high enough to generate buoyancy forces which will prevent the velocity from approaching zero. The thermal plumes have been visually observed in the vicinity of the rods. As the Reynolds number decreases, the intensity of the turbulent plumes increases. The plumes spread radially outwards at higher axial levels and, as adjacent plumes interact, they generate turbulence which probably is responsible for the reduction in the temperature gradients across the bundle. As mentioned earlier, the intensity of the turbulent plumes increases as the flow is reduced, given a constant heat input. Decreasing flow would mean an increase in turbulent mixing cross flow rate, w' . A constant value of β would result in a decrease of w' over time due to a decrease in flow rate ($\beta sG = w'$). Consequently, a constant value of β was not used to match the predictions with data. Instead, w' was correlated with the ratio of the bundle Grashof number to the square of the bundle average Gr/Re^2 . This ratio was multiplied by the local to average heat flux in the bundle since no thermal plumes will be present if heat generation rate is low or zero. Cases 1 to 3 did not require this correlation for w' ($w' = 0$ was used), although the predictions matched data equally well with this correlation as well as with $\beta = 0$. This was because the turbulent mixing effect caused by thermal plumes apparently becomes effective below a Reynolds number of about 400. More low-Reynolds-number data is being generated in order to confirm this.

FIGURE 3.102. Comparison of Isothermal and Heated Bundle Velocity Profiles (steady-state flow)

The results of this study permitted the following conclusions:

- Transient velocity and temperature predictions obtained with the COBRA-WC computer program using default input parameters were in satisfactory agreement with experimental data above Reynolds numbers of about 450.
- An improvement in the turbulent cross flow mixing model in COBRA-WC resulted in acceptable predictions of velocity and temperature histories at Reynolds numbers below 450.
- Recirculation was detected during the transient induced by the flow coastdown and not during steady-state flow conditions. A hypothesis that adequately described this behavior was formulated.

FIGURE 3.103. Comparison of Isothermal and Heated Bundle Velocity Profiles (steady-state flow)

- A preliminary criterion for the inception of recirculation in the mixed-convection transient flow regime was found to be $Gr/Re^2 (\hat{Q}_p - 1) \geq \sim 0.2$ where Gr is the bundle Grashof number, Re is the Reynolds number and $\hat{Q}_p$ is the radial local to bundle average power skew.

Singer et al. (1979, E04) have reported in-pile data in EBR-II related to emergency decay heat removal by natural circulation. Reactor flow and temperature response to three general categories of natural circulation plant transients are discussed. These categories include the following: 1) loss of forced flow from decay power and low initial flow rates; 2) reactor scram with a delayed loss of forced flow; and 3) loss of forced flow with a scram

FIGURE 3.104. Measured (channel centerline) and COBRA Predicted (channel average) Velocity Histories

activated by the plant protective system. In all cases, the transition from forced to natural convective flow was smooth and the peak in-core temperature rises were small to moderate.

The first category of transient was run from a decay power level of 1.6% and a forced flow rate of 5.3% of the full flow rate. The transient is initiated by a loss of pumping power and a subsequent transition to natural circulation. Figure 3.105 shows the resulting response of the reactor. The coolant flow rate in an instrumented assembly drops from 5.3% to 0.7% in 20 sec, and then gradually increases to 1.2% to 1.3% as buoyancy forces develop. The corresponding coolant temperature at the top-of-core location rises to a peak of 513°C (inlet temperature = 371°C) at 60 sec and slowly declines thereafter.

FIGURE 3.105. Thermal and Hydraulic Core Response to a Transition to Natural Convection from Low Power and Flow Conditions

Figure 3.106 shows the results of a series of tests where the effect of reactor decay power upon peak in-core temperature were measured. In contrast to forced-convection transients, in which a similar plot of the peak temperature rise versus reactor power would be linear, these natural-circulation transients exhibit a strong nonlinear peak transient temperature versus power relationship. This nonlinearity is due to the increased power, which generates higher temperatures and buoyancy forces. Thus, in natural-circulation transients, the system inherently tends to mitigate the effects of power changes. A delayed loss-of-forced-flow transient with reactor scram results in flow rates that drop to 0.23% of the rated condition. This is because the core has cooled down after scram, which results in a reduced rate of buildup of buoyancy forces and consequently lower flow rates. The top of the active core and subassembly outlet coolant temperatures indicate significantly slower rise times than were experienced in the category 1) transients, but higher peak values are reached because of adverse convective flow conditions (Figure 3.107).

FIGURE 3.106. Dependence of the Peak Transient Coolant Temperatures at Core Top and Subassembly Outlet Upon the Reactor Decay Power

FIGURE 3.107. Thermal and Hydraulic Core Response to a Transition to Natural Convection with Reactor Scram Preceding Loss of Forced Flow

APPENDIX A
REFERENCES

APPENDIX A

REFERENCES

The references presented here are grouped alphabetically within 19 independent sections. The first three sections cover books, general survey papers, and papers presented at a key International Atomic Energy Agency (IAEA) meeting in 1979. The remaining 16 sections divide the reference by topic: analytical studies or experimental studies; hydrodynamics or hydrothermodynamics; local or global approaches; nominal rod bundle geometry and nominal operation conditions or non-nominal rod bundle geometry and/or non-nominal operation conditions. The 19 reference sections are coded as shown below. For example, the reference (Dwyer 1966, N1) can be found listed alphabetically by author in Reference Section N1.

<u>Code</u>	<u>Reference Source or Topic</u>
Book	Books
Survey	General Survey Papers
IAEA	1979 Meeting at Karlsruhe
N1	Analytical Studies: Local Hydrodynamics--Nominal Geometry/Nominal Conditions
N2	Analytical Studies: Global Hydrodynamics--Nominal Geometry/Nominal Conditions
N3	Analytical Studies: Local Hydrodynamics--Nominal Geometry/Nominal Conditions
N4	Analytical Studies: Global Hydrodynamics--Nominal Geometry/Nominal Conditions
O1	Analytical Studies: Local Hydrodynamics--Non-Nominal Geometry/Non-Nominal Conditions
O2	Analytical Studies: Global Hydrodynamics--Non-Nominal Geometry/Non-Nominal Conditions
O3	Analytical Studies: Local Hydrodynamics--Non-Nominal Geometry/Non-Nominal Conditions
O4	Analytical Studies: Global Hydrodynamics--Non-Nominal Geometry/Non-Nominal Conditions
EN1	Experimental Studies: Local Hydrodynamics--Nominal Geometry/Nominal Conditions
EN2	Experimental Studies: Global Hydrodynamics--Nominal Geometry/Nominal Conditions
EN3	Experimental Studies: Local Hydrothermodynamics--Nominal Geometry/Nominal Conditions
EN3	Experimental Studies: Global Hydrothermodynamics--Nominal Geometry/Nominal Conditions
E01	Experimental Studies: Local Hydrodynamics--Non-Nominal Geometry/Non-Nominal Conditions
E02	Experimental Studies: Global Hydrodynamics--Non-Nominal Geometry/Non-Nominal Conditions
E03	Experimental Studies: Local Hydrothermodynamics--Non-Nominal Geometry/Non-Nominal Conditions
E04	Experimental Studies: Global Hydrothermodynamics--Non-Nominal Geometry/Non-Nominal Conditions

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APPENDIX B

SUBCHANNEL AND BUNDLE FRICTION FACTORS AND FLOW SPLIT
PARAMETERS FOR LAMINAR, TRANSITION AND
TURBULENT LONGITUDINAL FLOWS IN WIRE-WRAP-SPACED
HEXAGONAL ARRAYS

APPENDIX B

Subchannel and Bundle Friction Factors and Flow Split Parameters for Laminar, Transition and Turbulent Longitudinal Flows in Wire-Wrap-Spaced Hexagonal Arrays

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ABSTRACT

Correlations are presented for subchannel and bundle friction factors and flow split parameters for laminar, transition and turbulent longitudinal flows in wire-wrap-spaced hexagonal arrays. These results are obtained from pressure drop models of flow in individual subchannels. For laminar flow, models are developed for pressure drops of individual subchannels. From these models, expressions for the subchannel friction factors are identified and expressions for the flow split parameters are derived. Bundle friction factors are found by the equal pressure drop assumption and the preceding expressions. For turbulent flow, an existing pressure drop model (Chiu et al. 1979) for flow in edge subchannels is modified, and the resulting edge subchannel friction factor is identified. Using the expressions for flow split parameters of Chiu et al. and the equal pressure drop assumption, the interior subchannel and bundle friction factors are obtained. For transition flow, a model is proposed which superposes any one of the three pairs of the friction factor components obtained earlier. Values for the flow split parameters are obtained by using the transition flow values for the subchannel friction factors in the equal pressure drop assumption. The various constants which appear in the final expressions are calibrated against the available literature on bundle friction factor data. Results of this procedure show that for friction factors, 97% of the data points fall within $\pm 16\%$ of the predicted values while for flow split parameters, 100% of the data points are within $\pm 10\%$ of the predicted values. An improved formulation for friction factor for the laminar case will be presented at the ANS topical meeting at Saratoga NY, Oct. 1980.

1.0 INTRODUCTION

Liquid metal fast breeder reactors (LMFBRs) currently employ hexagonal assemblies of wire-wrapped fuel rods. The wires are wrapped around the rods in a helix of lead length, H , all in the same direction, and consequently, at any vertical cross section they are all at the same angular position relative to the same arbitrarily chosen axis of each rod. The upward flow of coolant through both the interior and edge type of subchannels results in a pressure drop which is primarily due to the form drag of the wires and the skin friction of the rods and wall.

Various empirical methods for correlating the bundle average friction factors as functions of the wire-wrapped sub-assembly geometries have been suggested (Novendstern 1972; Rehme 1973; Engel et al. 1979). Significantly, none of these authors extended their efforts to correlate subchannel based friction factors, which are needed as input to LMFBR design codes such as COBRA-IIIC or IV (Rowe 1973; Wheeler et al. 1976). These codes solve coupled conservation equations and can be used to derive flow split parameters for use as input to those codes (Glazer et al. 1979) in which only the energy equation is employed to calculate the sub-assembly temperature distribution. To date, the only work on subchannel friction factors has been by (Chiu et al. 1979) in a study which was restricted to turbulent flow. Thus a need exists for phenomenologically based subchannel and bundle average friction factor correlations.

In this paper, models for subchannel pressure drop in rod bundles in purely laminar and turbulent flows are proposed. From these models, expressions for subchannel friction factors, flow split parameters and bundle friction factors are derived. A correlation model is then proposed which superposes the laminar and turbulent friction factors in order to derive their respective transition region values, and from these the transition region values of the flow split parameters are obtained. The corner subchannel friction factor is not predicted in this paper because of the small effect this parameter has on the distribution of flow in the remainder of the bundle.

2.0 LAMINAR FLOW

In the laminar flow regime no phenomenological models for subchannel pressure drops exist in the literature. Consequently, pressure drop models for each subchannel type must be formulated in order to obtain the respective friction factors and flow split parameters. The bundle friction factor may then be obtained by the equal pressure drop assumption.

2.1 SUBCHANNEL PRESSURE DROP MODEL

The general form of the laminar friction factors, f_{1L} , f_{2L} and f_{bL} ^(a) can be expressed as:

$$f_{iL} = \frac{C_{iL}}{Re_i} \quad (1)$$

The quantity C_{iL} is a function of the dimensionless geometrical parameters P/D and H/D . To discover the exact nature of the dependence of C_{iL} on P/D and H/D , the overall subchannel pressure drop in laminar flow must be formulated and compared to the Darcy pressure drop formula, $\Delta P = fL/D_e \rho V^2/2$. The total subchannel pressure drop has two components: form loss and skin friction loss. The form pressure drop, ΔP_{iLF} , is found from the definition of the drag coefficient:^(b)

$$A'_{si} \Delta P_{iLF} = A_{ri} C_D \frac{\rho V_w^2}{2} \quad (2)$$

According to the results of Eisner, as presented by Rohsenow and Choi (1961), the drag coefficient at very low Reynolds number is approximately inversely proportional to a Reynolds number based on the wire diameter and V_w , i.e.

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- (a) A complete nomenclature for the symbols used in this appendix can be found on pages B.31 and B.32.
(b) Primed quantities refer to equivalent bare rod bundle values.

$$C_D = \frac{C_Y}{\frac{v_w D_w}{v}} \quad (3)$$

where C depends only on the shape of the obstacle. The behavior of C_D at very low flow rates is chosen in order to ensure that the expression to be obtained corresponds to purely laminar flow. Such a choice also ensures that the resulting flow split parameters are independent of Reynolds number, as is currently assumed. This assumption is made because there has not yet been any flow split data measured below a Reynolds number of 500. Inserting Equation (3) into (2) we have:

$$\Delta P_{iLF} = \left(\frac{A_{ri}}{A_{si}} \right) \frac{C_Y}{v_i De_i / v} \frac{\rho v_i v_w}{2} \left(\frac{De_i}{D_w} \right) \quad (4)$$

Assuming that a parabolic velocity distribution exists in a subchannel in laminar flow (i.e., Hagen-Poiseuille flow), it can be shown that v_w is related to the subchannel average velocity v_i in the following manner:

$$v_w = C_{\beta i} v_i \left(\frac{D_w}{De_i} \right) \left(1 - \frac{D_w}{2De_i} \right)$$

Inserting Equation (5) into (4) yields:

$$\Delta P_{iLF} = \frac{A_{ri}}{A_{si}} \frac{C_{\alpha i}}{Re_i} \frac{\rho v_i^2}{2} \left(1 - \frac{D_w}{2De_i} \right) \quad (6)$$

where $C_{\beta i}$, C_Y has been compressed into $C_{\alpha i}$. This result expresses the form loss for flow passing over a single wire. Over a subchannel length, L_b , however, there will be (L_b/H) such wires and consequently the above result must be multiplied by this factor to give the overall form loss.

The total pressure drop, ΔP_{iL} , may now be expressed as the sum of the above result and the skin friction loss (which is given by the Darcy formula):

$$\begin{aligned}\Delta P_{iL} &= \Delta P_{iLS} \text{ (skin friction)} + P_{iLF} \text{ (total form)} \\ &= \left[\frac{C_i}{Re_i} + \frac{C_{\alpha i}}{Re_i} \left(\frac{A_{ri}}{A_{si}'} \right) \left(\frac{De_i}{H} \right) \left(1 - \frac{D_w}{2De_i'} \right) \right] \frac{L_b}{De_i} \frac{\rho v_i^2}{2}\end{aligned}\quad (7)$$

C_1 has been analytically evaluated by Sparrow and Loeffler (1959) who obtained the following approximate correlation for an infinite triangular array of fuel rods:

$$C_1 = \left(-336(P/D - 1)^2 + 340(P/D - 1) + 40 \right) \frac{A'}{A} \frac{De_1}{De_1'} \quad (8)$$

C_2 is presented in graphical form by Rehme (1971). In order to present numerical values, a best fit was made to this graph with the result that:

$$C_2 = \left(-318.2(P/D-1)^2 + 290.9(P/D-1) + 37.4 \right) \frac{A'}{A} \frac{De_2}{De_2'} \quad (9)$$

Equation (7) is valid only for the case that $\frac{W}{D} = \frac{P}{D}$.

2.2 SUBCHANNEL FRICTION FACTORS, FLOW SPLIT PARAMETERS AND BUNDLE FRICTION FACTOR

From Equation (9) it can be seen that the expression for laminar flow, subchannel friction factors is:

$$f_{iL} = \frac{C_i}{Re_i} + \left(\frac{A_{ri}}{A_{si}'} \right) \left(\frac{De_i}{H} \right) \left(1 - \frac{D_w}{2De_i'} \right) \frac{C_{\alpha i}}{Re_i} \quad (10)$$

The laminar flow, flow split parameters, which may be derived from the assumption of equal subchannel pressure drops, mass continuity, and the above expressions for friction factors, are:

$$x_{1L} = \frac{N_1 A_1 + N_2 A_2 + N_3 A_3}{N_1 A_1 + (N_2 A_2 + N_3 A_3) \left[\frac{C_1 + C_{\alpha 1} \left(\frac{A_{r1}}{A'_{s1}} \right) \left(\frac{De_1}{H} \right) \left(1 - \frac{D_w}{2De'_1} \right)}{C_2 + C_{\alpha 2} \left(\frac{A_{r2}}{A'_{s2}} \right) \left(\frac{De_2}{H} \right) \left(1 - \frac{D_w}{2De'_2} \right)} \right] \left(\frac{De_2}{De_1} \right)^2} \quad (11)$$

$$x_{2L} = \frac{N_1 A_1 + N_2 A_2 + N_3 A_3}{N_1 A_1 \left[\frac{C_2 + C_{\alpha 2} \left(\frac{A_{r2}}{A'_{s2}} \right) \left(\frac{De_2}{H} \right) \left(1 - \frac{D_w}{2De'_2} \right)}{C_1 + C_{\alpha 1} \left(\frac{A_{r1}}{A'_{s1}} \right) \left(\frac{De_1}{H} \right) \left(1 - \frac{D_w}{2De'_1} \right)} \right] \left(\frac{De_1}{De_2} \right)^2 + (N_2 A_2 + N_3 A_3)} \quad (12)$$

From the assumption of equal subchannel and bundle pressure drops, and the above expressions, (Equations (11), (12)), the laminar flow bundle friction factor is found from:

$$f_{bL} = f_{iL} \frac{De_b}{De_i} x_{iL}^2 = F(C_{\alpha 1}, C_{\alpha 2}, \text{geometry}) \quad (13)$$

The method by which the constants in Equations (10), (11), (12) and (13) were calibrated and the results are presented in the discussion (Section 6.0).

3.0 TURBULENT FLOW

In the turbulent flow regime, phenomenological models for subchannel pressure drops exist in the literature (Chiu et al. 1979). Expressions for flow split parameters were derived from these models, and calibrated against

the available literature data; analogous expressions for subchannel friction factors were not developed fully (See Chiu et al. 1979). Consequently, the pressure drop model for one subchannel type (either edge or interior) must be completed in order to obtain an expression for that subchannel friction factor. The other subchannel friction factor and the bundle friction factor may then be obtained from the equal pressure drop assumption and the known expressions for the flow split parameters.

3.1 EDGE SUBCHANNEL PRESSURE DROP MODEL

As mentioned in the introduction and above, Chiu et al. (1979) presented a detailed phenomenological model for the pressure drop in subchannels in turbulent flow. In this model, the form loss due to flow past the wire and the skin friction due to the actual subchannel flow direction were taken into account. A simplification in the final results was achieved by considering only the dominant components of pressure drop in each subchannel type: for edge subchannels the dominant component is skin friction due to the actual flow direction, while for interior subchannels the dominant components are skin friction due to axial flow and form loss. The results which were obtained are:

$$P_{1T} = \left[1 + 2200 \left(\frac{A_{r1}}{A_{s1}} \right) \left(\frac{De_1}{H} \right) \left(\frac{p^2}{\pi^2 p^2 + H^2} \right) \right] \frac{C_{f1}}{Re_1^m} \frac{L_b}{De_1} \frac{\rho V_1^2}{2} \quad (14)$$

$$P_{2T} = \left\{ 1 + \left[1.9n \frac{V_t}{V_A} \text{gap} \right]^2 \right\}^{1.375} \frac{C_{f2}}{Re_2^m} \frac{L_b}{De_2} \frac{\rho V_2^2}{2} \quad (15)$$

The geometrical dependences of the factors C_{f1} and C_{f2} were not considered by Chiu et al. (1979) because the authors were only interested in their slowly varying ratio. For the goals of this paper, however, determination of the geometrical dependence of C_{f2} is necessary.

The behavior of C_{f2} may be determined by comparing the available literature on bundle friction factor data with the expressions for the bundle friction factor which are obtained from the equal pressure drop assumption, Equations (14) and (15), and the expression for the edge flow split parameters derived in the paper by Chiu et al. (1979), i.e.,

$$f_{bT} = f_{2T} \frac{De}{De_i}^2 x_{2T} \quad (16)$$

This was done by the authors and it was found that no simple power curves (e.g. $C(P/D)^X (H/D)^Y$) adequately predicted C_{f2} . Consideration was then given to deriving an expression for the axial flow friction factor based on a detailed examination of the boundary layer behavior as it passed each wire; this effort was dropped as it became apparent that the resulting expressions for the various possibilities would be extremely complicated (See Table 1 for a list of the various possibilities of the boundary layer behavior). Consequently a simple phenomenological model was developed and this was found to satisfactorily predict the geometrical dependence of C_{f2} .

It was found that the geometrical dependence of C_{f2} could be found by considering the additional pressure drop due to the velocity profile distortion which is caused by the presence of the wire on the rod and the fact that some edge subchannel flow slips over it. Characterizing this pressure drop by an empirical constant K , we have

$$\Delta P_{2p} = K \frac{\rho v^2}{2} \quad (17)$$

The velocity, V_p , whose direction is perpendicular to the axis of the wire, is related to the axial velocity V_2 in Chiu et al. (1979) by (see Figure (1)):

TABLE 1. Possible Length-Averaged Friction Factor Expressions Based on Boundary Layer Considerations

Boundary Layer Flow Regime	Becomes Fully-Developed Within H	Does not Become Fully-Developed
Laminar	Not Applicable	$f=f_{DL}$
Turbulent (negligible laminar length)	$f = f_{DT} \frac{L_{DT}}{H}$ $+ f_{DT} \frac{H-L_{DT}}{H}$	$f=f_{DT}$
Laminar, becomes turbulent within H	$f = f_{DL} \frac{L_{DL}}{H}$ $+ f_{DT} \frac{L_{DT}}{H}$ $+ f_{FD} \frac{H-L_{DTOT}}{H}$	$f = f_{DL} \frac{L_{DL}}{H}$ $+ f_{DT} \frac{H-L_{DL}}{H}$

Subscripts indicate:

Flow regime	{ L=laminar
	{ T=turbulent
Character of	{ D=developing
Boundary Layer	{ FD= fully-developed

$$\tan \theta = \frac{\pi D}{H}$$

$$V_P = \sin \theta V_A$$

FIGURE 1. Relationship Between Subchannel Axial Velocity and Its Component Perpendicular to the Wire

$$\frac{v_p}{v_2} = C_c (f_{s2})^{.5} \frac{p^2}{\pi^2 p^2 + H^2} \quad (18)$$

where f_{s2} is the skin friction factor for flow with a fully developed velocity profile. Inserting Equation (18) into (17), we obtain

$$\Delta P_{2p} = C_c^2 f_{s2} \frac{p^2}{\pi^2 p^2 + H^2} \quad (19)$$

Adding this additional pressure drop to the skin friction pressure drop for the case of no velocity profile distortion (i.e., no wire) the result is

$$\begin{aligned} \Delta P_{2s} &= f_{s2} \frac{L_b}{De_2} \frac{\rho v_2^2}{2} + \Delta P_{2p} \frac{L_b}{H} \\ &= \left[1 + C_c^2 \left(\frac{De_2}{H} \right) \left(\frac{p^2}{\pi^2 p^2 + H^2} \right) \right] f_{s2} \frac{L_b}{H} \frac{\rho v_2^2}{2} \end{aligned} \quad (20)$$

From this relationship, the subchannel friction factor for channels with many sudden velocity profile disturbances (i.e. an edge subchannel) may be identified as:

$$\frac{C_{f2}}{Re_2^m} = \left[1 + C_c^2 \left(\frac{De_2}{H} \right) \left(\frac{p^2}{\pi^2 p^2 + H^2} \right) \right] f_{s2} \quad (21)$$

The functional form of f_{s2} is assumed to be:

$$f_{s2} = \frac{C_T}{Re_2^{.15}} \quad (22)$$

where the exponent of 0.15 (instead of 0.20 which is usually used for smooth tubes), is assumed to apply because Ito's (1959) experimental data suggests that secondary flow (analogous to the sweeping flow in wire-wrapped rod bundles) will change the exponent from 0.2 to 0.15. Also, the very wide scatter in the available data makes it impossible to confidently predict this exponent's behavior with variations of geometry (see Figure 2). With this caveat then, the choice of 0.15 is reasonable on the physical grounds cited.

The total turbulent pressure drop for an edge subchannel may now be expressed as:

$$\Delta P_{2T} = \left\{ 1 + \left[1.9n \frac{V_t}{V_A} \text{ gap} \right]^2 \right\}^{1.375} \left[1 + C_c^2 \left(\frac{De_2}{H} \right) \left(\frac{p^2}{\pi^2 p^2 + H^2} \right) \right] \cdot \frac{L_b}{De_2} \frac{\rho V_2^2}{2} \quad (23)$$

3.2 SUBCHANNEL FRICTION FACTORS, FLOW SPLIT PARAMETERS AND BUNDLE FRICTION FACTOR

From the above result it can be seen that the expression for a turbulent edge subchannel friction factor is:

$$f_{2T} = \left\{ 1 + \left[1.9n \frac{V_t}{V_A} \text{ gap} \right]^2 \right\}^{1.375} \left\{ 1 + C_c^2 \left(\frac{De_2}{H} \right) \left(\frac{p^2}{\pi^2 p^2 + H^2} \right) \right\} \frac{C_T}{Re_2^{0.15}} \quad (24)$$

The turbulent flow, flow split parameters, which may be derived from the assumption of equal subchannel pressure drops, mass continuity, and the above friction factor expressions, are:

$$x_{1T} = \frac{A_{s1}N_1 + A_{s2}N_2 + A_{s3}N_3}{A_{s1}N_1 \left[\frac{c_1 \left(\frac{De_1}{H} \right) \left(\frac{A_{r1}}{A_{s1}} \right) \left(\frac{p}{\sqrt{\pi^2 p^2 + H^2}} \right)^{2+1}}{\left\{ 1 + \left[C_2^n \frac{V_T}{V_A} \text{ gap} \right]^2 \right\}^{1.375} C_3} \right]^{0.571} \left(\frac{De_2}{De_1} \right)^{0.714} [A_{s2}N_2 + A_{s3}N_3]} \quad (25)$$

GRAPH
KEY

$\Delta P/D$ \ $\Delta H/D$	0-20	21-40	41-60	61-80	81-100
1.05-1.15	○	⊙	○	⊕	⊗
1.16-1.25	△	△	△	△	△
1.26-1.35	□	□	□	□	□
1.36-1.45	◇	◇	◇	◇	◇

FIGURE 2. Demonstration of the Lack of a Simple Geometrical Dependence of Reynolds Number. Exponent in the General Friction Factor Expression, $f = c/Re^x$

$$x_{2T} = \frac{A_{s1}N_1 + A_{s2}N_2 + A_{s3}N_3}{A_{s1}N_1 \cdot \left[\frac{\left\{ 1 + \left[C_2 n \frac{V_T}{V_A \text{ gap}} \right]^2 \right\}^{1.375} C_3}{C_1 \left(\frac{De_1}{H} \right) \left(\frac{A_{r1}}{A_{s1}} \right) \left(\frac{P}{\sqrt{\pi^2 P^2 + H^2}} \right) + 1} \right]^{0.571} \left(\frac{De_1}{De_2} \right)^{0.714 + A_{s2}N_2 + A_{s3}N_3}} \quad (26)$$

The interior subchannel friction factor is found from:

$$f_{1T} = f_{2T} \left(\frac{De_1}{De_2} \right) x_{2T}^2 / x_{1T}^2 \quad (27)$$

Although the pressure drop models for interior and edge subchannels in turbulent flow have been modified, there are no changes in the expressions for x_{1T} and x_{2T} which were derived by Chiu et al. (1959). This is because the modifications, (i.e., determining the form of C_{f2} and changing the assumed Re exponent from 0.25 to 0.15) are absorbed by constants in these expressions. In the first case, since only the ratio of f_{1T} and f_{2T} are formed in determining the flow split parameters, only the ratios of C_{f1} and C_{f2} are important. But these ratios are very nearly constant and have been treated as such by Chiu et al. (1979) (C_3 in Equations (25) and (26)). In the latter case, the resulting changes in the exponents in the flow split expressions (e.g. .571) leads only to a small change in the predictions and even this small change would be accounted for by a recalibration of the constants C_1 and C_2 in Equations (25) and (26).

From the assumption of equal subchannel and bundle pressure drops, and the above expressions, the turbulent flow bundle friction factor is found from:

$$f_{bT} = f_{2T} \frac{De^2}{De_2^2} x_{2T}^2 \quad (28)$$

The method by which the constants in Equations (24) (25) (26) (27) and (28) were calibrated, and the results are presented in the discussion.

4.0 SIMPLIFIED LAMINAR AND TURBULENT FLOW, FLOW SPLIT PARAMETERS

A simplification in the evaluation of the laminar and turbulent flow split parameters, due to Symolon,^(a) comes about by the introduction of three factors, G_L , G_T , and S :

$$G_L \equiv \frac{x_{2L}}{x_{1L}} = \frac{C_1 + C_{\alpha 1} \left(\frac{A_{r1}}{A'_{s1}} \right) \left(\frac{De_1}{H} \right) \left(1 - \frac{D_w}{2De_1} \right)}{C_2 + C_{\alpha 2} \left(\frac{A_{r2}}{A'_{s2}} \right) \left(\frac{De_2}{H} \right) \left(1 - \frac{D_w}{2De_2} \right)} \left(\frac{De_2}{De_1} \right)^2 \quad (29)$$

$$G_T \equiv \frac{x_{2T}}{x_{1T}} = \left[\frac{c_1 \left(\frac{De_1}{H} \right) \left(\frac{A_{r1}}{A'_{s1}} \right) \left(\frac{P}{\sqrt{\pi^2 P^2 + H^2}} \right)^2 +}{\left[1 + \left[C_2^n \frac{V_T}{V_A} \mid \text{gap} \right]^2 \right]^{1.375} c_3} \right]^{0.571} \left(\frac{De_2}{De_1} \right)^{0.714} \quad (30)$$

$$S \equiv \left(\frac{A_{s2}}{A_{s1}} \right) \frac{1}{N_R} + \left(\frac{A_{s3}}{A_{s1}} \right) \frac{1}{N_R^2} \quad (31)$$

The G -factors ($\equiv \frac{x_2}{x_1}$) are solely geometry dependent because they arise from the equal pressure drops assumption, and subchannel pressure drops depend on geometry only. The S -factor is primarily size dependent because it arises from the continuity equation. From Figures (3) and (4) one can determine the G_T and S factors by inspection rather than by the tedious calculation of equations (30) and (31). Formulating Equations (11), (12), (25) and (26) in terms of the appropriate pair of the above parameters yields:

(a) Authors obtained this information in private communications with P. Symolon.

FIGURE 3. Geometrical Dependence of Turbulent Flow Split Ratio
 Factor Introduced to Simplify Turbulent Flow Split
 Expressions

FIGURE 4. Bundle Size Dependence of S-Factor Introduced to Simplify Flowsplit Expressions

$$x_1 = \frac{1 + S}{1 + GS} \quad (32)$$

$$x_2 = \frac{1 + S}{1/G + S} \quad (33)$$

for either laminar or turbulent flow.

5.0 TRANSITION REGION, SUBCHANNEL AND BUNDLE FRICTION FACTORS

In the preceding sections, the quantities of interest have been derived for purely laminar and turbulent flows. In order to obtain values for these quantities in the transition region, the friction factors are assumed to have the form:

$$f_i = \left[(f_{iL})^\ell + (f_{iT})^\ell \right]^{1/\ell} \quad (34)$$

where the subscripts L and T denote purely laminar and turbulent flow conditions, and the exponent ℓ dictates the pattern of the superposition of the laminar and turbulent components. Phenomenologically, the form of Equation (34) is dictated by the smooth and continuous transition which occurs between laminar and turbulent flows. As written, the assumption embodied in Equation (34) is that the pattern of superposition of the components (i.e., the transition region) is the same for the different subchannels and the bundle. This is not precisely correct as the data of Chiu et al. (1977) shows in Figure 5. However, since there is very little other subchannel friction factor data available, this assumption is necessary. Furthermore, since the shapes of the data curves in Figure 5 are not greatly different, this assumption will not introduce much uncertainty into subchannel friction factor calculations. Thus only one value for ℓ has to be found. The method by which is found and the result are presented in the discussion.

A word of caution concerning the use of Equation (34) is in order. When this equation is being used to determine subchannel friction factors, the appropriate Reynolds numbers must be used in the different component expressions. For example, in determining f_{iL} one must use a laminar subchannel Reynolds number, that is, one which is obtained from a known bundle Reynolds number by multiplying it by x_{iL} and De_i/De_b ; i.e., $Re_i = Re_b x_{iL} De_i/De_b$. The same caveat applies to f_{iT} . It is not permissible to choose a particular subchannel Reynolds number and proceed to use it in evaluating the components because such a procedure implicitly assumes that the subchannel flow split parameters are known at the bundle Reynolds number in question and this is not generally true. Only when the flow split parameters are known may such a procedure be used (i.e., $Re_i < 400$ or $Re_i > 100,000$).

5.1 TRANSITION REGION, FLOW SPLIT PARAMETERS

From the above expressions for subchannel and bundle friction factors, and the assumption of equal subchannel and bundle pressure drops, the transition region, interior and edge flow split parameters may be obtained from:

$$x_1 = \frac{f_b}{f_1} \left(\frac{De_1}{De} \right)^{.5} \quad (35)$$

$$x_2 = \frac{f_b}{f_2} \left(\frac{De_2}{De} \right)^{.5} \quad (36)$$

FIGURE 5. Comparison of Shapes of Interior (f_1), Edge (f_2) and Bundle (f_b) Friction Factor Data to Show Their Similarity for Laminar and Turbulent Bundle Flow

6.0 DISCUSSION

The primary focus in this paper has been on subchannel based quantities. Expressions have been obtained for laminar, subchannel friction factors and flow split parameters, as well as turbulent, subchannel friction factors. Unfortunately, there exists a very small subchannel friction factor data base, and consequently we chose to work with the much larger bundle friction factor data base in calibrating the various constants which have appeared in the preceding sections. The following comparisons between the friction factor correlation of Equation (34) and the data, therefore, apply to bundle average quantities only. It should be kept in mind, however, that the expressions for the subchannel friction factors are at the base of the bundle friction factor predictions (by equal pressure drops), and consequently that the predictions of the former quantities should be more accurate than the latter, derived quantity.

Calibration of the laminar friction factor constants, which appear in Equations (10), (11), (12) and (13) was performed by matching the predicted laminar bundle friction factor with the available literature data (Engel et al. 1979; Chiu et al. 1977; Thorne 1977). This consists of 3 data points over the ranges:

$$1.07 \leq P/D \leq 1.25$$

$$4.0 \leq H/D \leq 52$$

$$61 \leq N_p \leq 217.$$

Although the data base is very small, geometrical dependencies for both $C_{\alpha 1}$ and $C_{\alpha 2}$ were determined. No physical basis for their behaviors is offered here. The results are:

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

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

Calibration of the turbulent friction factor constants, which appear in Equations (24), (25), (26), (27) and (28), was performed by matching the

predicted turbulent friction factor, f_{bT} , with the available literature data (Rehme 1973; Baumann et al. 1968; Reihman 1969; Bartholet et al. 1975) at a very high Reynolds number (100,000). This consists of 45 data points over the ranges:

$$1.07 \leq P/D \leq 1.343$$

$$7.6 \leq H/D \leq 96$$

$$19 \leq N_p \leq 217$$

At this high Reynolds number, f_{bL} is very small compared to f_{bT} and consequently the experimental data may be used as f_{bT} . The results are:

$$C_T = .118$$

$$C_c^2 = 598$$

In order to find λ , the predicted transition region friction factor, f_i (see Equation (34)), was matched against the available literature data (Rehme 1973; Baumann et al. 1968; Riehmman 1969; Bartholet et al. 1975; Magee and Nilsson 1971) for various values of λ . This consists of 57 data points over the ranges:

$$1.07 \leq P/D \leq 1.343$$

$$7.6 \leq H/D \leq 96$$

$$19 \leq N_p \leq 217$$

The result is:

$$\lambda = 1.4$$

Tables 2 and 3 list the principal results of the preceding sections.

Comparisons between the predictions of Equations (34) and (36) and the data used to develop these equations are presented in Figures (6) and (7). In making such comparisons it is necessary to choose a level of accuracy to be used as a standard of reference. In the cases of interest the uncertainty consists of two components: random, experimental errors which occur during

TABLE 2. Summary of Friction Factor Results

	LAMINAR	TURBULENT	TRANSITION	P/D	H/D	N _p
INTERIOR	Expression given by Equation (10) $Ca1 = 8700/(H/D)^{.33}$ C_f given by Eqn. (8)	Expression given by Equation (27) Using Eqns. (24), (25) and (26)	Expression given by Equation (34) $\epsilon = 1.4$	1.06 to 1.34	7.6 to 96	19 to 217
EDGE	Expression given by Equation (10) $Ca2 = 8300/(H/D)^{.16}$ C_2 given by Eqn. (9)	Expression given by Equation (24) $C_T = .118$ $C_c^2 = 598$	Expression given by Equation (34) $\epsilon = 1.4$	1.06 to 1.34	7.6 to 96	19 to 217
BUNDLE	Expression given by Equation (13) Using either Eqns. (10) and (11) or (10) and (12)	Expression given by Equation (28) Using Eqns. (24) and (26)	Expression given by Equation (34) $\epsilon = 1.4$	1.06 to 1.34	7.6 to 96	19 to 217
	Re<400	Re>100,000	2000<Re<300,000			

TABLE 3. Summary of Flow Split Results

<u>INTERIOR</u>	<u>LAMINAR</u>	<p>Expression given by Eqn. (11) with: $C_{\alpha 2} = 8300/(H/D)^{.16}$ <u>or</u> $C_{\alpha 1} = 8700/(H/D)^{.33}$ C_1 and C_2 given by Eqn. (8), (9)</p>	<p>Expression given by Eqn. (32), where G_L is defined in Eqn. (29) and S is defined in Eqn. (31)</p>	<400	1.06 to 1.25	4 to 51.4
	<u>TURBULENT</u>	<p>Expression given by Eqn. (24) with: $C_1 = 2200$ <u>or</u> $C_2 = 1.9$ $C_3 = 1.2$</p>	<p>Expression given by Eqn. (32), where G_T is defined in Eqn. (30) and S is defined in Eqn. (31)</p>	>15k	1.06 to 1.27	4 to 52
	<u>TRANSITION</u>	$x_1 = (f_b De_1 / f_1 De)^{.5}$		400 to 3×10^5	1.06 to 1.34	4 to 96
<u>EDGE</u>	<u>LAMINAR</u>	<p>Expression given by Eqn. (12) with: $C_{\alpha 1} = 8700/(H/D)^{.33}$ <u>or</u> $C_{\alpha 2} = 8300/(H/D)^{.16}$ C_1 and C_2 given by Eqn. (8), (9)</p>	<p>Expression given by Eqn. (33), where G_L is defined in Eqn. (29) and S is defined in Eqn. (31)</p>	<400	1.06 to 1.25	4 to 51.4
	<u>TURBULENT</u>	<p>Expression given by Eqn. (25) with: $C_1 = 2200$ <u>or</u> $C_2 = 1.9$ $C_3 = 1.2$</p>	<p>Expression given by Eqn. (33), where G_T is defined in Eqn. (30) and S is defined in Eqn. (31)</p>	>15k	1.06 to 1.27	4 to 52
	<u>TRANSITION</u>	$x_2 = (f_b De_2 / f_2 De)^{.5}$		400 to 3×10^5	1.06 to 1.34	4 to 96

(57 Sets of Experimental Data are considered)

FIGURE 6. Comparison of Bundle Friction Factor Correlation with Available Experimental Data

FIGURE 7. Comparison of Edge Subchannel Flow Split Correlation with Available Experimental Data

the measurement process and systematic errors due to inherent deviations of the bundle geometry from nominal specifications. For friction factors, $\pm 16\%$ is a good measure of accuracy, relative to data uncertainty, while for flow split parameters $\pm 10\%$ is sufficient. From Figure (6) it can be seen that 97% of the reported bundle friction factor data fall into the $\pm 16\%$ range around the predicted values; from Figure (7) it can be seen that all of the available edge subchannel flow split data is within a $\pm 10\%$ interval of the predicted values.

From both figures it is evident that the available laminar data base is small and for this reason, results using Equations (10), (11), (12) and (13) should be considered preliminary.

The above correlations feature the ability to predict a curve for each combination of geometrical parameters over a wide range of Reynolds numbers. The predicted curves are all smooth and continuous with no breakpoints (e.g., between laminar and transition flows): this behavior is well established by experimental data for the various friction factors; however, it is not well established for the flow split parameters because of their unstable nature in the transition regime. More transition flow split data need to be taken to definitively establish the correct behavior.

Another important attribute of the correlations is that they reduce to the bare rod case. Thus with appropriate recalibration they should predict bare-rod bundle data.

Finally, a few words are in order comparing our bundle friction factor correlation with those of Novendstern (1972) and Rehme (1973). The data base used by Novendstern plus the recent results of Engel et al. (1979) were used to establish common grounds for comparison among the three correlations (57 cases). All correlations were tested at fifty or more Reynolds numbers over the Reynolds number range of 2000 to 150,000. Table (4) summarizes the statistical information.

TABLE 4. Comparison of Turbulent Bundle Average Friction Factor Correlations

<u>Reference</u>	<u>Accuracy at 95% Confidence Level</u>	<u>Percent of Predictions Within 16% Accuracy</u>	<u>Average Unsigned Error</u>
This paper	<u>+17.5%</u>	97%	7.6%
Novendstern (1972)	<u>+16.0%</u>	94%	6.3%
Rehme (1973)	<u>+15.6%</u>	93%	5.6%

Three points need to be made about Table (4);

- (a) An accuracy of +2 standard deviations constitute a 95% confidence level, and this is shown in the first column of Table (4) for the three correlations. This title can be misleading, however, because this approach tends to smear out large prediction errors. In other words, close prediction of many points tends to negate a relatively small number of bad predictions. Thus a correlation which misses several geometries badly and hits many fairly well may do better in this type of analysis than one which is more consistent in its predictions, if the latter has a higher average error.
- (b) A way to overcome the bias in (a) is to set an accuracy range and to count the number of points which fall outside of this range when a large number of comparisons is made. This is what is shown in column two of Table (4).
- (c) Column three of Table (4) shows the unsigned average errors of the three correlations. This quantity is the average of the absolute value of the relative errors of all the comparisons between the correlations and the data. Thus, this quantity is a measure of the average relative error that can be expected in using one of the correlations in predicting a data point.

Application of the bundle friction factor correlation proposed in this paper will yield about as accurate predictions of bundle pressure drops as do the correlations of Novendstern and Rehme. Additionally, the proposed subchannel friction factor and flow split correlations will finally allow users to confidently make predictions of these important subchannel based parameters.

CONCLUSION

In this paper the authors have:

1. modeled laminar flow, pressure drop for interior, edge and corner subchannels in wire-wrapped rod bundles by considering the effects of form loss and skin friction;
2. extended an existing model of turbulent flow, pressure drop in interior and edge subchannels in wire-wrapped rod bundles by considering the disturbance of the wire on the velocity profile;
3. obtained expressions for laminar and turbulent flow subchannel and bundle friction factors and flow split parameters;
4. proposed a model for superposing laminar and turbulent friction factor components in order to obtain expressions for transition flow friction factors and flow split parameters; and
5. showed that the correlations proposed in (3) and (4) yield accurate results.

NOMENCLATURE

A = bundle area (cm^2)
 A_{ri} = projected area of wires in i -th subchannel (cm^2)
 A_{si} = axial average subchannel area for a single i -th type of subchannel (cm^2)
 A'_{si} = i -th subchannel type area with no wire (cm^2)
 A_w = area of wire (cm^2)
 C_1, C_2, C_3 = calibration constants from Reference (13), introduced in Eqns. (25) and (26); $C_1 = 2200.0$, $C_2 = 1.9$, $C_3 = 1.2$

 $C_{\alpha i}$ = constant introduced in Eqn. (6)
 $C_{\beta i}$ = constant introduced in Eqn. (5)
 C_Y = constant introduced in Eqn. (3)
 C_d = drag coefficient for flow past cylinders
 $C_i = f_i Re_i$ for i -th subchannel type for bare rod bundles
 D = rod diameter (cm)
 De = overall equivalent hydraulic diameter (cm)
 De_i = equivalent hydraulic diameter for subchannels including wire (cm)
 f = overall bundle friction factor
 f_i = i -th subchannel type friction factor
 f_{si} = i -th subchannel friction factor for bare rods in turbulent flow
 F = subscript introduced in Eqn. (2) to denote form loss
 g = gap between rod and duct
 G = parameter defined in Eqns. (29) and (30)
 H = wire lead length (cm)
 $i = 1, 2, 3$ and 6 ; subscripts which denote the interior, edge, and corner subchannel types, and the bundle, respectively
 ℓ = parameter introduced in Eqn. (34) which determines the pattern of the superposition of the laminar and turbulent friction factor components
 L = subscript introduced in Eqn. (1) to denote the laminar flow regime
 L_b = bundle length (cm)
 n = parameter from Chiu et al. (1977) introduced in Eqn. (15)
 N_i = number of i -th-type of subchannels in the bundle
 P = rod pitch (cm)

ΔP = pressure drop (newtons/cm²)
 ΔP_p = pressure drop due to the velocity profile distortion of wire
 P_w = bundle equivalent wetted perimeter (cm)
 Re = Reynolds number = $\rho V \cdot D_e / \mu$
 S = parameter defined in Eqn. (31)
 T = subscript introduced in Eqn. (14) to denote the turbulent flow regime
 $V_A = V$ = bundle average axial velocity (cm/sec)
 $V_{Ai} = V_i$ = i-th subchannel type axial velocity averaged over one wire lead length (cm/sec)
 V_p = velocity perpendicular to wire wrap, see Fig. (2)
 V_T / V_A gap = ratio of transverse velocity (V_T) to axial velocity which is correlated in the work of Chiu et al. (1977)
 V_w = local velocity in a subchannel at a distance $D_w/2$ from fuel rod
 $sX_i = X_i$ = i-th subchannel type flow split factor = V_i / V
 ρ = coolant density (g/cm³)
 μ = dynamic viscosity ($\frac{\text{newton-sec}}{\text{m}^2}$)

Primed quantities refer to equivalent bare rod bundle values.

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APPENDIX C

A REVIEW OF POROUS MEDIA MODELLING
FOR ROD BUNDLE THERMAL-HYDRAULIC ANALYSIS

by

D.S. Rowe

SUMMARY

A review of porous media and subchannel analysis approaches indicate that they are alternate methods of considering the same class of problems^[9]. Both consider the average (global) nature of flow and temperature fields as defined by the overall boundary conditions of the bundle. The average values are defined over the size of the computational cell used for analysis. Neither method can consider the local details of flow and heat transfer on a scale smaller than the size of a computational cell which is typically no smaller than the size of a subchannel. Both methods use the concept of distributed resistance to define the shear between the fluid and solid surfaces--the internal rod structure. Both methods can and do use the advanced computational methods (ICE) developed for classical flow fields^[10]. These methods allow consideration of a wide class of problems involving a variety of flow and pressure boundary conditions. Some often stated claims for the superiority of the porous media approach^[4] are not justified. For example, frequent reference to the inability to evaluate terms such as s/l and β in COBRA-IV-I are not true. The same information is required in the porous media approach and involve cell aspect ratios and eddy diffusion coefficients, respectively. Claims regarding the superiority of the momentum equations formulation in the porous media approach are also often overstated^[4]. As an example, the value of including the second order viscous stress term is of little value because it is at least an order of magnitude smaller than the distributed resistance (fluid-to-solid shear stress).

The porous media approach merits attention as it can provide an easier computation format for analysis of more general problems. It can be formulated with a close analogy to continuum flow fields by change of the constitutive relationships used for closure of the differential equations. This allows the use of existing work on numerical methods used in continuum flow fields. The porous media approach is most applicable when

the computational cell is much larger than the size of a subchannel and computational cells can be selected without much concern of geometry dependence. When the computational cells approach the size of subchannels, a number of problems arise in using the porous media approach and this impacts its apparent generality. As in subchannel analysis, the relationships for closure become dependent on the rod geometry within the computational cells. The definition of these relationships requires experiments specific to the chosen nodal geometry.

POROUS MEDIA TECHNICAL APPROACH

This section reviews some of the basic features of the porous media concept applied to rod bundle analysis and makes appropriate comparisons to subchannel approach. This review includes some basic definitions, assumptions, equations and boundary conditions.

Definitions

Two important definitions in porous media are the porosity and permeability. The porosity is defined as the ratio of fluid volume to total volume of the medium under consideration. Sometimes called volume porosity, it can be expressed as

$$\gamma_v = \frac{V_f}{V} \quad (1)$$

The computational cell has volume V in the porous media formulation; thus, the volume porosity is required to get the correct fluid volume. The fluid volume is specified directly in the subchannel analysis approach so volume porosity is not used.

A surface porosity can also be defined to relate the relative area for flow through a surface in the direction of flow; thus,

$$\gamma_s = \left(\frac{A_f}{A} \right)_s \quad (2)$$

This has also been called the surface permeability vector;^[4] however, that terminology is not preferred since permeability has a special meaning related to flow resistance in porous media. Note that the surface and volume porosity need not be the same. The actual flow areas at computational cell boundaries are specified directly in the subchannel analysis approach so a surface porosity vector is not used.

Permeability is defined in porous media to provide a relationship between superficial velocity and pressure gradient. In general, following the approach of Butterworth^[2] for rod bundles, the pressure gradient due to distributed frictional resistance can be expressed as

$$\vec{F} = -\mu \vec{R} \vec{v} \quad (3)$$

where $\vec{R}$ is the flow resistance tensor and μ is the viscosity. The superficial velocity $\vec{v}$ is the "smeared" average velocity over the volume. The actual local velocity ($\vec{u}$) (through pores) is higher. These can be related through the porosity by

$$\vec{v} = \gamma \vec{u} \quad (4)$$

The flow resistance tensor is related to the flow conductivity tensor by

$$\vec{k} \vec{R} = \vec{\delta} \quad (5)$$

where $\vec{\delta}$ is the Kronecker delta. $\vec{k}$ is equivalent to the permeability tensor for anisotropic flow through porous media.

To relate the flow resistance to the standard friction factor, consider the case of one-dimensional flow in a pipe where

$$\frac{dp}{dx} = -\frac{f}{D} \frac{\rho v |v|}{2} \quad (6)$$

This can be rearranged as

$$\frac{dp}{dx} = -\mu \frac{f(\rho D |v|/\mu)}{2 D^2} v \quad (7)$$

therefore; the single value of the flow resistance tensor is

$$R = \frac{f R_e}{2 D^2} \quad (8)$$

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The resistance is seen to be directly related to the diameter, friction factor and Reynolds number. For laminar flow with

$$R = \frac{32}{D^3} \quad (9)$$

and for turbulent flow with $f = a R_e^b$

$$R = \frac{a R_e^{(1+b)}}{2 D^2} \quad (10)$$

By using the same analogy with rod bundle analysis, the specification of resistance is the same as the specification of friction factor in the subchannel analysis approach. The distributed resistance features of the two methods are equivalent except possibly for the anisotropic properties of $\bar{R}$. Butterworth^[2] has investigated the properties of $\bar{R}$ for tube bundles under the assumption that $\bar{R}$ only depends on the magnitude of the superficial velocity vector $\vec{v}$. The result is that there are three principle directions for flow resistance where one is parallel to the tubes. The other two directions are along lines of symmetry perpendicular to the bundle axis. For square and triangular arrays the two directions perpendicular to the tubes are nearly isotropic based on experimental data for pressure drop for flow across tubes banks. Data are presented from tube bundle flow tests to support the theoretical analysis. This result implies that R (or $\bar{R}$) is a diagonal matrix. The primary conclusion is that the pressure drop data for one-dimensional flow through tube bundles in the axial and lateral direction can be used to provide the frictional resistance for multi-dimensional analysis of the tube bundles.

Butterworth's results support the approach which has been taken in sub-channel analysis for many years whereby the axial and lateral friction factor are assigned independently by using the hydraulic diameter concept. His result provides additional confirmation of the approach and further suggests that the Reynolds number be evaluated by using the magnitude of the superficial velocity.

Assumptions

Two principle assumptions are required for the development of equations for the porous media approach. They are as follows:

1. Flow and temperature field variables are averaged over a computational cell.

This assumption allows the porous media approach to consider only the average aspects of flow and temperature fields on a scale which is of the same order as the computational cell. Details of local gradients and fine flow structure cannot be resolved. Gradients are expressed in terms of differences between average variables. Local gradients at walls and computational cell interfaces are not resolvable. This assumption is most valid when average gradients are not large and the local gradients are well distributed throughout the region of interest. This would be most appropriate for a large number of rods penetrating a computational cell. This would be similar to the basic concept of "real" porous media. Under those conditions cells could be selected rather arbitrarily without introducing geometry dependence from node selection. As the computational cells approach the size of subchannels, however, the gradients become more significant and nodes selection becomes dependent on the specifics of rod geometry in a computational cell.

2. The fluid-to-solid shear is defined by distributed resistances.

This basic assumption together with the averaging assumption eliminates the possibility of considering the fine structure of rod bundle flows. Furthermore, since the distributed resistances due to fluid shear at the rod surface is normally large compared to fluid-to-fluid shear, the flow is dominated by the distributed resistance. Inclusion of viscous or eddy shear usually has little, if any, importance to rod bundle.

applications. This assumption impacts the right side of the momentum equations by the choice of constitutive relationships for closure. The same assumption is used for subchannel analysis.

Equations

Several equation formulations have been published regarding the application of the porous media concept to rod bundles^[1-4, 19]. They all draw upon the concept of describing the rod bundle as a quasi-continuum where variables are defined by suitable averages. The differences come from various forms of porosity definitions and the constitutive relationships for closure. There are basically two approaches which can be taken to formulate porous media equations.

One approach is to start with the local differential equations for a fluid volume and then integrate the equations over a volume which includes the fluid as well as interpenetrating solid material. The integration involves both volume and surface integrals including integration of jump discontinuities at the interface between fluid and solid. This approach has been used to develop the equations used in the SCORE code and also follows the traditional concepts of porous media equation development^[1]. The resulting equations contain volume porosity for the purpose of converting the region volume to a fluid volume. Superficial velocity, which is the averaged local velocity over the surface of the region, can be used instead of actual fluid velocity.

The second approach uses the idea of volume and surface porosity. This approach considers balance equations applied to a computational cell. Volume porosity is used to convert from the cell volume to the fluid volume and surface porosity is used to convert the cell surface area to the fluid flow area. Thus, while porosities are used, the approach is really a control volume approach. This approach has been taken in the COMMIX code^[4] and is really not much different from the approach used in subchannel analysis. This is because surface porosities are highly dependent upon the selection of computational cells when the cells are of the same size as the subchannels. For example, the following grid

selections have the same volum porosity; however, the surface porosity are much different.

Traditional porous media approach would make no distinction between the two grid selections and the porosity (volume and surface) would be the same. The same discussion applies to larger region computational cells that involve more rods per cell. Finer subdivision is possible, but again the application and results are very dependent on the choice of the computational cell.

The equations developed for the SCORE code^[1] are summarized in the following sections illustrate the equations used in the porous media approach for rod bundles. Differences and similarities with the COMMIX code^[4] equations are identified as they arise. The subchannel equations used in the COBRA-IV-I code^[7] are also compared to show the analogy between the porous media and the subchannel formulation.

Mass Balance

The volume averaged mass balance equation is obtained from an integration of the local fluid mass balance equation over an arbitrary volume and the result is

$$\frac{\partial}{\partial t}(\gamma \rho) + \nabla \cdot (\gamma \rho \vec{u}) = 0 \quad (11)$$

where γ is the porosity and $\vec{u}$ is the volume averaged local fluid velocity. This equation is identical in form to that of conventional continuum fluid mechanics. It is common to use the superficial velocity

in porous media; thus,

$$\frac{\partial}{\partial t}(\gamma \rho) + \nabla \cdot \rho \vec{v} = 0 \quad (12)$$

where $\vec{v}$ is the product $\gamma \vec{u}$.

The use of surface porosity in COMMIX^[4] gives an alternate form (in cartesian coordinates)

$$\frac{\partial}{\partial t}(\gamma_v \rho) + \frac{\partial}{\partial x}(\gamma_x \rho u) + \frac{\partial}{\partial y}(\gamma_y \rho v) + \frac{\partial}{\partial z}(\gamma_z \rho w) = 0. \quad (13)$$

This equation originates from a control volume mass balance by using flux differences $\Delta(\rho u \gamma_x \Delta y \Delta z)$, etc. Note that the surface porosities convert the computational cell flow area to the fluid flow area.

The mass balance for the subchannel formulation in COBRA-IV-I is written as

$$A \frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho A u) + D^T \rho s v = 0 \quad (14)$$

where D^T is a difference operator for lateral divergence. The equation for COBRA and COMMIX are identical for subchannels on a rectangular grid since:

$$A \Delta x = \gamma_v \Delta x \Delta y \Delta z \quad (\text{Volume})$$

$$A = \gamma_x \Delta y \Delta z \quad (\text{x-area}) \quad (15)$$

$$s \Delta x = \gamma_y \Delta x \Delta z \quad (\text{y-area})$$

$$s \Delta x = \gamma_z \Delta x \Delta y \quad (\text{z-area})$$

The mass balance equations for the three formulations are equivalent since they differ only by the definitions used.

The following Table 1 summarizes the essential features of the mass balance formulations for two dimensions:

Table 1 Comparison of Mass Balance Terms for Two-Dimensions.

	Mass	Axial Flow (x)	Lateral Flow (y)
SCORE	$\gamma_v \Delta x \Delta y \Delta z \rho$	$\gamma_v \Delta y \Delta z \rho u$	$\gamma_v \Delta x \Delta z \rho v$
COMMIX	$\gamma_v \Delta x \Delta y \Delta z \rho$	$\gamma_x \Delta y \Delta z \rho u$	$\gamma_y \Delta x \Delta z \rho v$
COBRA-IV-I	$A \Delta x \rho$	$A \rho u$	$S \Delta x \rho v$

The formulations for SCORE and COMMIX differ by the use of volume and surface porosity. The COMMIX and COBRA formulations are identical when the definitions of Equation (15) are used.

Linear Momentum Balance

The volume averaged momentum balance as derived for the SCORE code^[1] can be written in the form

$$\frac{\partial (\gamma_v \rho \hat{u})}{\partial t} + \nabla \cdot \gamma_v \rho \hat{u} \hat{u} = -\gamma_v \nabla p + \gamma_v \nabla \cdot \hat{\tau} + \gamma_v \hat{F} + \gamma_v \rho \hat{g} \quad (16)$$

The force $\hat{F}$ is that due to fluid shear on solid surfaces and is represented as a distributed resistance. It can be defined in terms of permeability (friction factors) as discussed previously. The term $\nabla \cdot \hat{\tau}$ includes the fluid-to-fluid shear from viscosity and turbulence. Its contribution is generally small for rod bundles since the distributed resistance term is dominant.

x-Direction Momentum Equation

The momentum equation (x-direction) developed for the COMMIX code^[4] can be written in the form

$$\begin{aligned} \frac{\partial}{\partial t}(\gamma_v \rho u) + \frac{\partial}{\partial x}(\gamma_x \rho u u) + \frac{\partial}{\partial y}(\gamma_y \rho u v) + \frac{\partial}{\partial z}(\gamma_z \rho u w) = \\ - \gamma_v \frac{\partial p}{\partial x} + \gamma_v F_x + \gamma_v \rho g_x \\ + \mu \left(\gamma_x \frac{\partial^2 u}{\partial x^2} + \gamma_y \frac{\partial^2 u}{\partial y^2} + \gamma_z \frac{\partial^2 u}{\partial z^2} \right) \end{aligned} \quad (17)$$

This is identical in concept to the formulation for SCORE, except for the use of the surface porosity. As for the mass balance equation, the use of the surface porosity converts the computational cell area to a fluid flow area.

The use of γ_v as a multiplier on $\partial p / \partial x$ is not consistent with the concept of surface porosity because the pressure acts on fluid surfaces and not on the volume as a body force. While volume weighting is used in the porous media derivation for the SCORE code, it does not seem to be consistent here.

The formulation for SCORE and COMMIX include the $\nabla \cdot \vec{\tau}$ term to account for fluid-to-fluid shear. The applicability of including this term for rod bundle analysis is not clear because of its small magnitude and uncertain basis for application. The velocity gradients are very local and their characteristic distance is small compared to the size of computational cells. Only with the proper correction factor could the term have meaning. It may be advisable to drop this term for most rod bundle application because of its small effect and questionable value. Including it tends to overstate the capability of the modelling approach and leads to false impressions of rigor.

To quantify the relative importance of the distributed resistance and the fluid-to-fluid shear terms, consider a representative example. Let them be represented by

$$F = \frac{f}{D} \frac{\rho u^2}{2g_c} \quad (18)$$

and

$$F.T = \mu \frac{\partial^2 u}{\partial y^2} \quad (19)$$

To quantify the value of these two terms assume 100° F water, $u = 20$ ft/sec, $f = 0.02$, $D = 0.5$ in, $\rho = 62$ lb/ft³; then,

$$F = \frac{0.02}{(0.5/12)} \frac{(62.0)(20)^2}{(2)(32.2)} = 184.8 \text{ lb/ft}^3. \quad (20)$$

Normally, $\Delta^2 u$ would be at most on the order of u , however, for counter current flow it could be of order $4u$. Assuming $\Delta^2 u = 4u$, $\Delta y = 0.7$ in, and $\mu = 4.8 \times 10^{-4}$ lb/ft-sec; then,

$$\mu \frac{\Delta^2 u}{\Delta y^2} = \frac{(4.8 \times 10^{-4})(4)(20)}{(0.7/12)^2} = 11.2 \text{ lb/ft}^3 \quad (21)$$

The fluid-to-fluid shear is seen to be more than an order of magnitude smaller than the distributed resistance. It would be even smaller by application of the surface porosity, however, that could be off set by

an increased value for eddy viscosity. As the computational cell becomes larger, Δy increases; however, $\bar{F}$ does not change. Thus, the fluid-to-fluid shear contribution becomes even smaller as more rods are allowed to penetrate a computational cell.

It is instructive to compare the axial momentum equation formulation used in the COBRA-IV-I code:

$$\begin{aligned}
 A \frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial x}(\rho A u^2) + D^T(\rho s u v) = & \quad (22) \\
 - A \frac{\partial p}{\partial x} + A F_x + A \rho g_x \\
 + \int_T D^T w' D u
 \end{aligned}$$

Except for the last term, this equation is identical with the COMMIX formulation on a rectangular grid. The various flux terms can be equated to those of the COMMIX formulation by using Equation (15). The last term of Equation (22) approximates only the turbulent shear between subchannels ($\mu \partial^2 u / \partial y^2$). Shear in the other directions is not included. Table 2 summarizes the various terms of the x-direction momentum equation. Except for the use of volume porosity in SCORE the terms are equivalent. Additional differences occur in the equations for distributed resistance and viscous shear stress used to close the equations.

Table 2 Comparison of x-Direction Momentum Equation Terms
for Two Dimensions (x, y)

Formulation	Momentum	x-Momentum	y-Momentum	Pressure Force	Shear Forces	Body Forces
SCORE	$\delta_v \Delta x \Delta y \Delta z \rho u$	$\delta_v \Delta y \Delta z \rho u^2$	$\gamma_v \Delta x \Delta z \rho u v$	$\delta_v \Delta y \Delta z \Delta P$	Eqn (16)	$\delta_v \Delta x \Delta y \Delta z \rho g_x$
COMMIX-I	$\delta_v \Delta x \Delta y \Delta z \rho u$	$\gamma_x \Delta y \Delta z \rho u^2$	$\gamma_y \Delta x \Delta z \rho u v$	$\gamma_v \Delta y \Delta z \Delta P$	Eqn (17)	$\delta_v \Delta x \Delta y \Delta z \rho g_x$
COBRA-IV-I	$A \Delta x \rho u$	$A \rho u^2$	$S \Delta x \rho u v$	$A \Delta P$	Eqn (22)	$A \Delta x \rho g_x$

y-Direction Momentum Equation

The literature related to the development of COMMIX^[4] refers to the assumptions used in the formulation of the transverse momentum equation for subchannel analysis. To help define the true nature of those assumptions it is instructive to compare the COMMIX and COBRA-IV-I formulation. For COMMIX the y-direction momentum equation is written as (without $\nabla \cdot \frac{\rho}{\rho_0}$ terms)

$$\frac{\partial}{\partial t}(\rho v) + \frac{\partial}{\partial x}(\rho v u) + \frac{\partial}{\partial y}(\rho v v) + \frac{\partial}{\partial z}(\rho v w) = -\rho \frac{\partial p}{\partial y} + \rho F_y. \quad (23)$$

The corresponding equation for COBRA-IV-I is

$$s l \frac{\partial}{\partial t}(\rho v) + s l \frac{\partial}{\partial x}(\rho v u) + s S D^T \rho v v \cos \theta = -s D p + s F_y \quad (24)$$

where the third term approximates the $\rho v v$ flux term only. The $\rho v w$ term is not included. The computational volume (staggered grid) in COMMIX is $\Delta x \Delta y \Delta z$ whereas in COBRA it is $s l \Delta x$. The two equations are equivalent for two-dimensional flow (rectangular grid) since (for $l \Rightarrow \Delta y$, $s \Rightarrow \Delta y$):

$$\begin{aligned} s l \Delta x &= \bar{\rho} \Delta x \Delta y \Delta z & (\text{Volume}) \\ s l &= \bar{\rho} \Delta y \Delta z & (\text{x-area}) \\ s \Delta x &= \bar{\rho} \Delta x \Delta z & (\text{y-area}) \end{aligned} \quad (25)$$

where the bar (-) denotes an average because of the staggered grid. This comparison shows that the s and l dimensions can be directly related to the computational cell dimensions and surface porosities of COMMIX.

The differences between the approaches used in COMMIX and COBRA are due to assumptions resulting from the choice of the computational cell network and not fundamental differences between the COMMIX porous media concept and the subchannel formulation. To accomodate the arbitrary subchannel arrangement provision in COBRA-IV-I, assumptions are used to permit simplification of the numerical solution. For a rectangular grid system the subchannel formulation can be written the same as in COMMIX; thus, there are no fundamental differences other than the assumptions used for the momentum flux terms in a three-dimensional problem. This assumption is concerned with the components of momentum flux. The full momentum flux array ($\vec{v}\vec{v}$) cartesian coordinates consists of the following:

$$\vec{v}\vec{v} = \begin{bmatrix} uu & uv & uw \\ vu & \cancel{vv} & \cancel{vw} \\ wu & \cancel{wv} & \cancel{ww} \end{bmatrix}$$

where u, v and w are the velocity components in the x, y, and z directions, respectively. The cross product of lateral velocity components wv and vw are not included in COBRA-IV-I. Their omission removes the ability to consider the axial component of vorticity. The normal flux terms involving vv and ww are approximate for rod bundles. The approximation concerns the assignment of these flux components in the direction of the rod gap under consideration.

A comment made earlier regarding the use of Y_v in the lateral pressure gradient in COMMIX can be related to the COBRA-IV-I momentum equation formulation. For the axial direction the equations are the same because $Y_x = Y_v$. For the lateral direction they are not the same because for COBRA-IV-I the lateral pressure gradient term is DP/L whereas in COMMIX it is $Y_v \partial P / \partial y$. Since $Dp \Rightarrow \Delta p$ the relationship between them is $Y_v / \Delta y = 1/L$. The quantity s/L appears in COBRA-IV-I because the transverse momentum equation is multiplied by s to get the crossflow

per unit length (s/l). The term s/l is nothing more than $\gamma_v \Delta z / \Delta y$ in COMMIX; thus, s/l is a cell aspect ratio weighted by a porosity factor. The issue of using s/l , γ_v or alternately γ_y is related to the relative weighting of the stress versus inertial terms. To be consistent with the ideas of porous media, the pressure gradient should be weighted the same as the distributed resistance to preserve the pressure gradient and friction factor relationship for rod bundles. The weighting of the inertial terms depends on the assumptions for terms such as γ_v and s/l . This is also related to the choice of definitions for averaging surface porosity on the staggered grid.

Table 3 summarizes the various terms of the y-direction momentum equation. The essential differences are the use of volume versus surface porosity. Additional differences occur in the equation for distributed resistance and viscous shear.

In summary the use of porous media ideas to the formulation of momentum equations for bundle analysis is analogous to the subchannel formulation. The assumptions used in the COBRA-IV-I transverse momentum equations are related to the choice of the computational grid and not on fundamental differences with the porous media formulation.

Energy Balance

The thermal energy balance for a fluid in a porous media can be written in the form

$$\frac{\partial}{\partial x}(\gamma \rho I) + \nabla \cdot (\gamma \rho \vec{u} I) + P \nabla \cdot \vec{u} = \dot{Q}_w + \gamma \rho \dot{Q}_g - \gamma \rho \cdot \vec{q} \quad (26)$$

where $\dot{Q}_w$ is the heat addition from the solid to the fluid, $\dot{Q}_g$ is the internal heat generation per unit volume and $\vec{q}$ is the heat transfer in the fluid. Constitutive relations are needed to define these quantities. The above equation is similar to that used in the formulation of SCORE. COMMIX uses the same form (with enthalpy) except

Table 3. Comparison of y-Direction Momentum
Equation Terms for Two-Dimensions

Formulation	Momentum	x-Momentum	y-Momentum	Pressure Force	Shear Forces
SCORE	$\partial_v \Delta x \Delta y \Delta z \rho v$	$\partial_v \Delta y \Delta z \rho u v$	$\partial_v \Delta x \Delta z \rho v^2$	$\partial_v \Delta z \Delta x \Delta P$	Eqn (16)
COMMIX-I	$\partial_v \Delta x \Delta y \Delta z \rho v$	$\partial_x \Delta y \Delta z \rho u v$	$\partial_y \Delta x \Delta z \rho v^2$	$\partial_v \Delta z \Delta x \Delta P$	Eqn (23)
COBRA	$S_L \Delta x \rho v$	$S_L \rho u v$	$S \Delta x \rho v^2$	$S \Delta x \Delta P$	Eqn (24)

the energy flux terms are weighted by the surface porosity:

$$\gamma_v \frac{\partial \rho h}{\partial t} + \frac{\partial}{\partial x} (\gamma_x \rho u h) + \frac{\partial}{\partial y} (\gamma_y \rho v h) + \frac{\partial}{\partial z} (\gamma_z \rho w h) =$$

$$\gamma_v \frac{\partial p}{\partial t} + \dot{Q}_w - \frac{\partial}{\partial x} \gamma_x p_x - \frac{\partial}{\partial y} \gamma_y p_y - \frac{\partial}{\partial z} \gamma_z p_z \quad (27)$$

The corresponding subchannel form used in COBRA-IV-I for single phase flow is

$$A \frac{\partial \rho h}{\partial t} + \frac{\partial}{\partial x} (\rho A u h) + \partial^T (\rho s v h) = \dot{Q}_w + D^T w D h + \frac{\partial}{\partial x} A A \frac{\partial T}{\partial x} \quad (28)$$

The left side is identical with the COMMIX formulation for a rectangular grid by Equation (15). The comparison for all three code formulations are shown in Table 4. Again the differences are primarily the use of volume or surface porosity.

Table 4. Comparison of Energy Equation Transport Terms for Two-Dimensions

Formulation	Energy	x-Energy Flow	y-Energy Flow	Heat Sources and diffusion
SCORE	$\gamma_v \Delta x \Delta y \Delta z \rho h$	$\gamma_v \Delta y \Delta z \rho u h$	$\gamma_v \Delta x \Delta z \rho v h$	Eqn (26)
COMMIX-I	$\gamma_v \Delta x \Delta y \Delta z \rho h$	$\gamma_x \Delta y \Delta z \rho u h$	$\gamma_y \Delta x \Delta z \rho v h$	Eqn (27)
COBRA-IV-I	$A \Delta x \rho h$	$A \rho u h$	$S \Delta x \rho v h$	Eqn (28)

The term $\partial p / \partial t$ is neglected since it is small for subsonic flows. The last two terms are equivalent to $D \cdot \epsilon D T$ in classical flow fields. Similar exchange is included in COMMIX to define q .

Heat transfer from the solid $\dot{Q}_w$ is defined by the heat generation in the fuel rods for problems where fuel temperatures are needed. If the fuel temperatures are needed then the heat transfer is normally defined by $h_w(T_w - T)$ and an energy balance written for the solid to define T_w .

Constitutive Relationships

An equation of state is required to relate density, pressure and internal energy (or enthalpy);

$$\rho = \rho(P, I). \quad (29)$$

Additional relationships are needed to close the equations. Most of those required have been discussed with the equations. To summarize, relationships are needed for:

momentum equation

- distributed resistances
- fluid-to-fluid shear

energy equation

- heat transfer to solids
- heat transfer within the fluid
- internal heat generation

Porosity and thermal and physical properties are required for all materials.

Ideally it would be desirable to have a universal set of relationships to close the porous media formulation for use in rod bundles regardless of the selected nodal system of computational cells. Unfortunately this is not possible because some constitutive relations for closure are very dependent on the bundle nodalization. This is especially true of the diffusive exchange processes of heat and momentum which depend on eddy and molecular diffusivities. Such geometry dependence has led to the rather standardized concept of channel centered subchannels in rod bundle analysis. For core-wide analysis the nodal geometry dependence is less because the porous media concept is better represented; however, in that case the gradients are not well defined by using averaged values.

To illustrate the problem of nodal geometry dependence, consider the following nodal layouts for a rectangular array.

These arrays have the same volume porosity for the same p/d ; however, their surface porosities are very different. The rod centered node has $\gamma_s = 1$ and the subchannel centered node has $\gamma_s \ll 1$. The energy solutions for the two nodal systems are equivalent, however, if surface porosity logic is applied as in COMMIX, the heat flux between nodes is in proportion to the value of γ_s . Experience from rod bundle experiments does not support that proportionality. Exchange factors must be developed which are applicable to a specific node system. It might be argued that the above nodal selections are not appropriate, but this is precisely the point. The constitutive relations for closure depend on the node selection.

To illustrate the above point more fully, consider the lateral diffusive energy flux which can be written in the form

$$q_t = \epsilon \frac{\partial T}{\partial y} \quad (30)$$

The application of this term in the nodal structure of a computer code produces difference equations of the form

$$q_{j+1/2} = a_{j+1/2} (T_{j+1} - T_j) \quad (31)$$

where the coefficient $a_{j+1/2}$ depends upon an appropriate combination of a diffusion coefficient and diffusion length (Δy).

The following table presents representative form of the above equation for the three code formulations:

Code Formulation	Diffusive Energy Flux,
SCORE	$\frac{\epsilon_j \gamma_{j+1/2}}{\frac{1}{2}(\Delta y_j + \Delta y_{j+1})} (T_{j+1} - T_j)$
COMMIX	$\frac{\epsilon_{j+1/2} \gamma_{j+1/2}}{\frac{1}{2}(\gamma_{v_j} + \gamma_{v_{j+1}}) \Delta y} (T_{j+1} - T_j)$
COBRA*	$\left(\frac{\epsilon}{\Delta y} \right)_{j+1/2} (T_{j+1} - T_j)$

* $\epsilon/\Delta y = \beta \rho u C_p$

Note that in all cases there is a coefficient multiplying the temperature difference and that coefficient depends on the details of specifying the properties of the grid through the porosity. Note also that the flux cannot be preserved by simply transferring to another grid. For example, the coefficient is not the same when applied to the two computational grids shown above unless the eddy diffusion coefficient is redefined. The exchange coefficient depends on the selection of the computational grid. Experimental data are needed to define the proper correlation form of the coefficient.

The above considerations also apply to the fluid-to-fluid stress terms for the momentum equation, but in that case the effects are normally smaller because of the small value of those terms.

The choice of nodal geometry also has an impact on the ability to represent transport across the boundary of adjacent computational cells by using averaged variables. As an example, the velocity in a rod bundle could be distributed as follows:

The dominant wall shear in the rod gap depresses the velocity such that the velocity gradient defined by the subchannel average velocity difference

$$\tau_{i+1/2} = \mu \left(\frac{u_{j+1} - u_i}{\Delta y_{j+1/2}} \right)$$

may have little to do with the local shear

$$\tau_{i+1/2} = \mu \left(\frac{\partial u}{\partial y} \right)_{i+1/2}$$

This becomes more serious as the computational cells become larger than subchannels; however, in that case, the shear terms are relatively unimportant should probably be dropped.

Better representation of the actual shear gradient could be obtained by using finer nodes. Reported applications of COMMIX attempt this; however, in that case the selected noding still cannot resolve the true gradients and the selection of nodal geometry also becomes intimately related to the exchange coefficients for heat and momentum as discussed previously.

The distributed resistance term has considerably less dependence on nodal geometry. This is primarily because the hydraulic diameter concept from tubes works reasonably well for rod bundles. The geometry dependencies which do exist usually involve fine tuning of friction factor correlations.

Boundary Conditions

Averaging over the bundle prevents details of the rods to be included as boundary conditions to solve the differential equations. The boundary conditions normally available to the solution of the flow field are those at the boundary of the bundle. This includes a zero flow boundary condition normal to the side walls and free slip condition parallel to the walls. The distributed resistance accounts for shear at the wall. The inlet and outlet of the bundle can use either flow or pressure boundary conditions and they can be distributed provided they do not violate a global mass balance. Internal boundary conditions can only be specified through specifications at computational cell interfaces. Blockages, for example, can be defined by zero flow through one or more cell boundaries. Islands of no-flow are also possible to define irregular geometry. Internal boundary conditions involving the rod surfaces are not normally used because their specification is not normally resolvable in the space scale of the computational grid nor applicable in the space scale of the computational grid. Application of such boundary conditions can provide solutions to problems which can resolve flow fields on a scale which is "micro" with respect to the apparatus, but "macro" with respect to the details of the flow in a computational cell. This "macro-micro" approach has been found to produce reasonable flow solutions for a variety of rod bundle conditions which have been measured experimentally.

The boundary conditions for the energy equation use the flow field together with the specification of temperature or heat flux at the boundary of the apparatus. Heat transfer across internal boundaries are not resolved but are included by using constitutive relations.

Modelling Multi-Dimensional Flow Fields

The previous discussion has shown the similarity between the classical viscous flow equations and those used in the porous media formulation and in subchannel analysis.

There is a basic difference, however, related to the right side of the flow momentum equation. In classical fluid flow equations, the shear force is usually described by a coefficient of fluid viscosity embedded in a second order difference on velocity rather than a distributed resistance which is algebraically related to velocity. Mathematically this difference is important because it reduces the order of the differential equation from two to one. It's reasonable to ask if the distributed resistance approach can properly model the required nature of the flow field. The answer to the question is yes based upon the analysis of Esposito and Spencer^[11]. They point out that by splitting the flow velocity field into its rotational and irrotational components, it is possible to draw some conclusions regarding this question. Vorticity does not diffuse as in classical flow fields; however, there is decay along a stream line which is directionally dependent. They further conclude that, "if the irreversible losses in the energy equation can be neglected, as it is assumed to be in most reactor applications, it can be shown that for compressible flows at small Mach numbers, or for thermally expanding fluids, the irrotational part is unaltered by either momentum diffusion (molecular or turbulent) or flow resistance effects. This means that, in general, only the rotational part of the flow field will be affected by the frictional terms. Therefore, it should be noted that the irrotational flow field is preserved using either the microscopic or M & M formulation, with some difference occurring in the rotational part of the flow field as mentioned above."

The results of Esposito and Spencer can be extended somewhat for nonisotropic flow resistance to provide more insight into the porous media approach. Part of the motivation for this discussion is to help define the level of modelling necessary to consider friction and momentum driven recirculatory flow in bundles. In particular, can the distributed resistance approach model such phenomena or is a contribution required from a second order viscous stress term. This could be important because if the second order term is not included but has an important influence, then the results would be affected by the second order numerical diffusion errors.

Following the approach used by Esposito and Spencer, let the velocity be expressed by

$$\vec{v} = \vec{v}_I + \vec{v}_R \quad (34)$$

where $\vec{v}_I$ is the irrotational velocity field and $\vec{v}_R$ is the rotational field. The irrotational field is defined by the gradient of the velocity potential $\nabla\phi$ and the divergence of the irrotational field is zero; thus,

$$\nabla \cdot \vec{v}_I = \nabla \cdot \nabla\phi = 0 \quad (35)$$

For incompressible and uniform density flow

$$\nabla \cdot \vec{v} = \nabla \cdot (\vec{v}_R + \vec{v}_I) = 0 \quad (36)$$

therefore,

$$\nabla \cdot \vec{v}_R = 0. \quad (37)$$

The rotational and irrotational fields each satisfy the mass balance.

The momentum equation can be written in terms of vorticity ω where

$$\omega = \frac{1}{2} \nabla \times \vec{v} \quad (38)$$

Since $\nabla \times \vec{v}_I = 0$, the vorticity represents only the rotational field. Consider the momentum equation in the form

$$\frac{D\vec{v}}{Dt} + \frac{1}{\rho} \nabla p = -a v \vec{\kappa} v - b v \nabla^2 \vec{v} + \rho \vec{g} \quad (39)$$

where a and b weighting factors.

For incompressible flow and uniform density and viscosity, the curl of the momentum equation gives

$$\frac{D\vec{\omega}}{Dt} = -\frac{1}{2} \alpha \nabla \times \vec{R} \vec{v} - b \nabla^2 \vec{\omega}. \quad (40)$$

In the absence of the distributed resistance term the transport of vorticity is diffusive as it is advected along a stream line of $\vec{v}$. There are no sources (or sinks) of vorticity except at boundaries. Vorticity can occur at boundaries due to no slip (viscous shear) or by velocity distributions imposed at boundaries or created by internal obstacles. In the absence of viscous diffusion terms vorticity can only be advected along stream lines and the vorticity can only be defined at an inflow boundary. The vorticity in that case is advected through the flow field without change.

If the viscous diffusion term is replaced by the distributed resistance term, vorticity is no longer diffusive. If the resistance is isotropic then

$$\frac{1}{2} \nabla \times \vec{R} \vec{v} = R \vec{\omega} \quad (41)$$

where R is now scalar. The vorticity in this case decays along streamlines rather than being diffusive. This is the fundamental difference between the porous media and classical field approach. The only boundary conditions which introduce vorticity are those imposed at the inflow boundary or created by internal obstacles.

In the case of anisotropic resistance, vorticity may not necessarily decay along a stream line. There may be conditions which produce sources of vorticity rather than sinks as for isotropic resistance. To illustrate, consider the two-dimensional case where

$$\hat{\vec{R}} \hat{\vec{v}} = \begin{bmatrix} R_{11} u & R_{12} v \\ R_{21} u & R_{22} v \end{bmatrix}. \quad (42)$$

The curl for uniform R is

$$\begin{aligned} \nabla \times \hat{\vec{R}} \hat{\vec{v}} &= R_{11} \frac{\partial u}{\partial y} + R_{12} \frac{\partial v}{\partial y} \\ &\quad - R_{12} \frac{\partial u}{\partial x} - R_{22} \frac{\partial v}{\partial x} \end{aligned} \quad (43)$$

Using Butterworth's result for flow along principle directions, R_{12} and R_{21} are zero leaving

$$\nabla \times \hat{\vec{R}} \hat{\vec{v}} = R_{11} \frac{\partial u}{\partial y} - R_{22} \frac{\partial v}{\partial x} \quad (44)$$

This can be rewritten as

$$\begin{aligned} \nabla \times \hat{\vec{R}} \hat{\vec{v}} &= \frac{1}{2}(R_{11} + R_{22}) \left(\frac{\partial u}{\partial y} - \frac{\partial v}{\partial x} \right) \\ &\quad + \frac{1}{2}(R_{11} - R_{22}) \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \\ &= (R_{11} + R_{22}) \omega \\ &\quad + (R_{11} - R_{22}) \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \end{aligned} \quad (45)$$

For $R_{11} = R_{22}$, the result is that vorticity only decays along a stream line. For $R_{11} \neq R_{22}$, the last term can produce a source or sink for vorticity. This is not unreasonable because large anisotropy of the resistance tensor can produce the equivalent of an internal blockage which does produce vorticity.

While the previous discussion cannot totally resolve the question of the importance of second order viscous term on flow recirculation it provides some insights. The previous discussion indicates that the porous media approach approximates the rotational flow field by transporting vorticity along stream lines. Vorticity tends to decay along a stream line rather than being diffused, however, it can also be produced under conditions of anisotropic resistance. Vorticity therefore, occurs from boundary velocity distribution or from internal flow disturbances produced by internal boundary conditions or from non-uniform and anisotropic flow resistance.

As for the velocity field, only the global vorticity field can be calculated. Local details of vorticity transport from one rod surface to another cannot be considered. Vorticity generated at no-slip walls of the rod bundle would be a very small fraction of that which is internally diffused and eliminated by averaging. As for the velocity field, the global diffusion of vorticity would be more than an order of magnitude smaller than the decay or production of vorticity by the distributed resistance in a rod bundle.

In conclusion, the magnitude of the diffusive component of vorticity is small in the porous media concept of rod bundle modelling. This is not, however, proof that it does not affect momentum driven flow recirculation. Measurements may be necessary in rod bundle to resolve the question. Additional theoretical work beyond this brief attempt may also be useful.

Summary Remarks on Equations

The porous media approach to rod bundle analysis has significant value for thermal-hydraulic analysis. It provides an alternate approach which is analogous to the previously developed subchannel methods. The basic idea is to use averaged fluid variables and distributed resistances considering the "micro" details of the flow and internal boundary conditions defined by geometry of the rod surfaces. The advantage of drawing the analogy between the subchannel and porous media formulation is that they can both be related to computational methods used to solve the classical equations of fluid flow. There is a danger in this analogy because the porous media formulation can be "oversold" as having more rigor because it has the appearance of the classical field equations. This is especially true when second order terms involving viscous diffusion are included in the momentum equations. As discussed previously, those terms are usually small in rod bundles because of the dominance of the distributed resistances. There is also danger of concluding that the porous media approach is superior to the subchannel approach when in fact they are equivalent. The differences are primarily items of choice, convenience, and assumption in particular computer codes. The use of surface porosity does not offer a fundamental advantage over the subchannel approach because they are factors which, together with cell dimensions, define the equivalent of channel areas, gap spacings and the cell aspect ratios. The advantage of the porous media approach is that it can be more easily set up to consider full 3-D problems and is more easily adapted to the computational methods used for classical field equations.

COMPUTATIONAL METHODS

One of the advantages of the porous media formulation is that it provides an equation form which is very similar to classical fluid mechanics. This similarity allows a clearer communication link between numerical analysts; thus, it allows direct applications of the computational methods of fluid mechanics. [7, 12]

The method of solution which has become very popular over the past several years is based on the Implicit Continuous Eulerian (ICE) algorithm. [7] It has been effectively applied in various forms to subchannel, [7,11,13,15] porous media [1,4] and classical flow fields. [10] It is very powerful method which can consider a wide range of problems. It can also be applied to two-phase flows. The method solves for the pressure and velocity directly rather than using defined variables such as stream function and vorticity. [12]

To illustrate the method, consider the following balance equations of mass, momentum and energy in difference form

$$\frac{\rho^{n+1} - \rho^n}{\Delta t} + \nabla_d \cdot \rho \vec{v}^{n+1} = 0 \quad (46)$$

$$\frac{\rho \vec{v}^{n+1} - \rho \vec{v}^n}{\Delta t} + \nabla_d \cdot \rho \vec{v} \vec{v}^n + \nabla_d p^{n+1} = \vec{F}^n + \vec{g} \rho^n \quad (47)$$

$$\frac{h^{n+1} - h^n}{\Delta t} + \vec{v}^n \cdot \nabla_d h^n = \frac{q^n}{\rho^n} \quad (48)$$

where ∇_d is a linear spatial difference operator. As an example in two-dimensions, the divergence of the mass flux is

$$\nabla_d \cdot \rho \vec{v} = \frac{\rho u_{i+1/2,j} - \rho u_{i-1/2,j}}{\Delta x_i} + \frac{\rho v_{i,j+1/2} - \rho v_{i,j-1/2}}{\Delta y_j} \quad (49)$$

and the x-direction pressure gradient is

$$\nabla_d P = \frac{P_{i+1,j} - P_{i,j}}{\Delta x_{i,j}} \quad (50)$$

and the y-direction pressure gradient is

$$\nabla_d P = \frac{P_{i,j+1} - P_{i,j}}{\Delta y_{i,j}} \quad (51)$$

Operations such as $\nabla_d \cdot \nabla_d$ can be deduced from the above definition by index shifting. The momentum and energy flux terms are handled similarly, but incorporate various forms of averaging and spatial differencing (up-wind, centered, etc.). Roache has an excellent discussion of this in his book^[12]. The solution of the above equations has received much attention over the past decade. The method of solution to be discussed has its origin at LASL with development of the ICE Method^[10].

The basic approach used in ICE requires a solution for the pressure field from which the velocity can be determined. Iterative methods have been found to provide efficient solutions through an iterative update of a tentative solution. The basic idea is similar to the predictor-corrector methods. Let the correct solution at $n + 1$ be expressed as

$$p^{n+1} = \tilde{p}^{n+1} + \delta p \quad (52)$$

$$\rho^{n+1} = \tilde{\rho}^{n+1} + \delta \rho \quad (53)$$

$$\rho v^{n+1} = \tilde{\rho v}^{n+1} + \delta \rho v \quad (54)$$

where the " $\sim$ " represents tentative new-time solution and " δ " represents the increment necessary to achieve the correct solution. The tentative values at the start of a time step are taken to be $\tilde{p}^{n+1} = p^n$,

$$\tilde{\rho}^{n+1} = \rho(p^n, h^{n+1})$$

$$\tilde{\rho v}^{n+1} = \rho v^n - \Delta t (\tilde{F}^n + \tilde{g}^n - \nabla_d p^n - \nabla_d \cdot \rho \tilde{v}^n \tilde{v}^n) \quad (55)$$

Subsequent "tilde" values are the current values during the iteration. By noting the linear relationship between pressure and velocity and by using the above definitions, the difference equations can be replaced by

$$\frac{\partial \rho}{\partial p} \frac{\delta p}{\Delta t} + \nabla_d \cdot \delta \rho \tilde{v} = -D \quad (56)$$

and

$$\frac{\delta \rho \tilde{v}}{\Delta t} + \nabla_d \delta p = 0 \quad (57)$$

where

$$D = \frac{\tilde{\rho}^{n+1} - \rho^n}{\Delta t} + \nabla_d \cdot \tilde{\rho} \tilde{v}^{n+1} \quad (58)$$

represents the mass balance error, or residual, for a cell. Equations (56) and (57) can be combined to give

$$\frac{\partial \rho}{\partial p} \frac{\delta p}{\Delta t} - \Delta t \nabla_d \cdot \nabla_d \delta p = -D \quad (59)$$

This equation is a Poisson equation for the pressure correction required to achieve a correct solution.

A variety of solution strategies have been used to solve the above equations. Except for the simplest of problems, it is customary to use an iterative solution. Most involve iteration by successive point over relaxation (SOR) or by successive line over relaxation (SLOR). The SOR method is simple and is suitable for many situations. The approach is to solve for a single value of δp at a point in the flow field where the difference equation is centered at that point and the surrounding values of δp are zero. That value of δp is then used to calculate ρ and all the affected values of $\rho \tilde{v}$ (there are 6 for 3=D). The update of $\rho \tilde{v}$ and ρ are done as the updates become available. This is done for each node of the flow field and the process is repeated iteratively until the mass balance is satisfied at each node such that $D \rightarrow 0$. The

SLOR procedure is done in the same way except that an entire line vector of δp is solved at one time. The line solution is fast because the coefficient matrix is tridiagonal and can be solved by well developed methods^[32].

While the solution for the pressure field correction is straight forward, it is a time consuming computation. It is essential that it be done fast and efficiently. There are convergence "traps" for problems which involve distorted meshes, high aspect-ratio meshes or highly anisotropic properties. The solutions to achieve rapid convergence are often problem dependent.

Boundary conditions are included by setting p or $\rho \hat{v}$ at appropriate boundaries and inhibiting their update as necessary. Boundary conditions for slip or no slip can be included by setting appropriate velocities in boundary cells.

The above method of solution is stable with respect to sonic velocity propagations. It is time step limited; however, by transit time through a node. Specific time step requirements depend upon the spatial differencing of the convective energy flux and momentum flux.

The time step limitation can be removed by making the energy and momentum flux terms implicit. This has been done for solutions where it is desired to reach a steady state without performing the full transient solution. The most well known technique is SIMPLE.^[16] It uses a predict and correct strategy for successive iterations rather than stepping through time. Each iteration involves a tentative solution of the momentum equation followed by a pressure correction to conserve mass. Solution of the energy or other scalar transport equations are also included in the iteration. The SIMPLE algorithm

has been applied to the subchannel code SABRE^[6]. The ICE and SIMPLE type of algorithms have also been applied to heat exchanger applications by using porous media concepts^[17,18].

The basic idea of the above numerical method is used in both SCORE-EVET and in COMMIX by using point iteration. The numerical approach in COMMIX, however, introduces a mass residual in the energy and momentum equations with a claim of faster convergence. It is not known what effect this has on the accuracy of the numerical solution. The basic idea of the residual can be illustrated for the momentum equation with an analogous situation for the energy equation. The momentum equation can be written in the following form with the introduction of the mass balance:

$$\rho \frac{\partial \vec{v}}{\partial t} + \rho \vec{v} \cdot \nabla \vec{v} + \vec{v} \left(\frac{\partial \rho}{\partial t} + \nabla \cdot \rho \vec{v} \right) = -\nabla p + \dots \quad (60)$$

The third term of the equation is zero; however, it is not set to zero for computation in COMMIX. During iteration, the mass balance has a value given by D which tends to zero for a converged solution. COMMIX admits the possibility of a nonconverged solution such that D need not be zero; thus, a source term $\vec{v}D$ appears in the momentum equation in regions of local nonconvergence. Furthermore, the source term is included in calculating the tentative solution for the next time step.

For the momentum equation to be correctly solved it must be both transport and conserve momentum properly. The momentum equation in the form of Equation (16) does conserve momentum because it can produce correct global momentum balance. This can also be preserved by proper finite differencing of the momentum flux term^[12]. The momentum equation with continuity introduced ($D = 0$) as in Equation (17) transports momentum correctly along stream lines. Depending upon the finite differencing and the flow direction (reversals) it may not necessarily conserve momentum^[12]. Both conservative and transportive schemes can be derived for Equation (16)^[12].

The introduction of the mass residual seems to confuse the matter and it is not clear that the solution method used in COMMIX properly transports and conserves momentum. If the computation logic is accepted, then a correct mass balance is not required to obtain a correct solution to the momentum equation.

ADDITIONAL COMMENTS

The previous discussion has primarily been concerned with thermal-hydraulic modelling in bare-rod bundles. Additional considerations exist for grid spacers, wire wrap spacers and distorted bundles (blockage). Comments are also included regarding code verifications.

Grid Spacers

Grid spacers are included in subchannel codes by modifying the frictional resistance to flow and modifying the lateral mixing factor (eddy diffusion). This same approach is applicable to the porous media concept. The added flow resistance of the grid spacer is treated as an added contribution to the distributed resistance. The use of the porous media concept offers the same degree of resolution as the subchannel approach. Both methods provide global predictions (averages) of flow, pressure, and temperature within a subchannel.

Wire Wrap Spacers

Wire wrap spacers present a special problem for rod bundle analysis since they tend to force flow from one subchannel to another. This is not a turbulent mixing process, but one which involves macroscopic motion resulting in mixing. The edge of the bundle always has a preferential lateral velocity in the direction of wire wrap rotation. Eddy diffusion, through turbulence modelling alone, cannot model wire wrap bundles. The convective nature of the wraps must be modelled. Furthermore, a single channel with a wire wrap cannot be modelled unless all other channels are considered or proper boundary conditions are included. The role of porous media modelling in this context is the same as subchannel modelling. The porous media approach does not offer a more fundamental approach to the modelling of wire wraps.

It is conceivable that the use of a specialized permeability together with momentum source terms could be formulated for a porous media code. While this may sound like a different approach, the idea is very similar to that used in COBRA-IV-I. The formulation of such models would also depend on the choice of computational cell geometry and the location of solids within them.

Distorted Bundles

Variation in porosity (channel area) along the length together with related changes in the distributed resistance can accommodate distorted bundles. This is equivalent to the approach used in subchannel codes. Both approaches fix the spatial location of the computation grid and vary the parameter of interest within the fixed grid. Geometry dependence related to the nature of the distortion must be factored into the exchange coefficients for heat transfer between computation cells. This approach for the subchannel formulation has been checked successfully against velocity measurements in a bundle containing sleeve blockages.

Unfortunately, "real world" distortions or blockages are seldom well defined. It may be that the porous media concept may provide an easier analysis format for cases where the normal rod geometry can no longer be described.

Code Verification

Porous media and subchannel codes are required to perform two specific calculations for single-phase flow. One is the calculation of the flow and pressure field. The other is the calculation of the temperature field. Quite often codes are compared directly with temperature data and mixing coefficients are adjusted to get agreement with the data. This obscures deficiencies in calculated flow field. If the flow field can be properly calculated then comparison and adjustment of mixing coefficients could be done with greater confidence. It is essential that a code be checked and verified with flow data for a variety of

conditions including; normal flow distribution, effects of spacers, and effects of flow upsets (blockage, inlet maldistributions, etc.).

Noding and the effect of numerical diffusion should also be checked and defined because it "artificially" adds to the true diffusion coefficients for thermal mixing.

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