WCAP-9695-A

BART-Al: A Computer Code for the Best Estimate Analysis of Reflood Transients

by

M. Y. Young J. S. Chiou J. Kabadi T. A. Porsching S. R. Rod A. C. Spencer

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Approved:

F. F. Cadek, Manager

Safeguards Engineering and Development

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WESTINGHOUSE ELECTRIC CORPORATION Nuclear Energy Systems P.O. Box 355 Pittsburgh, Pennsylvania 15230

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UNITED STATES NUCLEAR REGULATORY COMMISSION WASHINGTON, D. C. 20555

DEC 2 1 1983

Mr. E. P. Rahe, Jr., Manager Nuclear Technology Division P. O. Box 355 Pittsburgh, Pennsylvania 15230

Dear Mr. Rahe:

Subject: Acceptance for Referencing of Licensing Topical Report WCAP-9561, "BART A-1: A Computer Code for Best Estimate Analysis of Reflood Transients"

We have completed our review of the subject topical report submitted lanuary 15, 1980, by Westinghouse Electric Corporation letter NS-TMA-2169. We find this report is acceptable for referencing in license applications to the extent specified and under the limitations delineated in the report and the associated NRC evaluation which is enclosed. The evaluation defines the basis for acceptance of the report.

We do not intend to repeat our review of the matters described in the report and found acceptable when the report appears as a reference in license applications except to assure that the material presented is applicable to the specific plant involved. Our acceptance applies only to the matters described in the report.

In accordance with procedures established in NUREG-0390, it is requested that Westinghouse publish accepted versions of this report, proprietary and non-proprietary, within three months of receipt of this letter. The accepted versions should incorporate this letter and the enclosed evaluation between the title page and the abstract. The accepted versions shall include an -A (designating accepted) following the report identification symbol.

Should our criteria or regulations change such that our conclusions as to the acceptability of the report are invalidated, Westinghouse and/or the applicants referencing the topical report will be expected to revise and resubmit their respective documentation, or submit justification for the continued effective applicability of the topical report without revision of their respective documentation.

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Sincerely,

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Cecil O. Thomas, Chief Standardization & Special Projects Branch Division of Licensing

Enclosure: As stated

## SAFETY EVALUATION REPORT ON BART CODE

## 1.0 INTRODUCTION

In WCAP-9561 (Ref. 1), Westinghouse presented the BART computer code which will be used to replace the FLECHT correlation used in the reflood stage of the Westinghouse ECCS Evaluation Model reflood calculation. The BART code provides a time and axial location dependent fuel rod clad surface heat transfer coefficient as input to the detailed, previously approved fuel rod heatup code, LOCTA. The BART calculation is based upon the hydraulic information calculated by the approved code, WREFLOOD, and beginningof-reflood fuel rod initial conditions.

The BART computer code is developed based upon mechanistic models to predict the fuel rod quench behavior during reflood. Basically, there are two separate and stand alone rod heat conduction and channel hydraulic models incorporated into BART. The two models are the one-dimensional model (1-D), and the quench front or two-dimensional model (2-D). The 1-D model is fairly conventional. The 2-D model is different from any of the other quench front models currently under development. This model uses the isotherm migration method to track and analyze the heat transfer at the quench front. The 2-D model is tightly coupled to the 1-D calculation. It is a separate, almost parallel, calculation that exchanges both initial conditions, and resulting heat flux information with the 1-D model.

### 2.0 STAFF REVIEW AND EVALUATION

The staff review covered the following three areas:

- Review of the models developed for BART. The models are reviewed to ascertain: (a) soundness of physical assumptions and hypotheses,
   (b) correctness of thermal-hydraulic equations and (c) appropriateness of the application of correlations including the proper range of physical conditions in comparison with the data base used in original references.
- Review of the qualification of the BART code by comparing integral response of the code with the test data. The evaluation addresses:

   (a) adequacy of data base and (b) determination of errors between prediction and experimental data including the percentage of data points captured by the code prediction.
- Review of the applicability of the code replacing the FLECHT correlation for ECCS analysis.

The review will be discussed in the following sections.

## 2.1 The 1-D Hydraulics Model

The hydraulics model used in BART is a pseudo non-equilibrium two-phase model. With the assumption of constant pressure, fluid equations are developed for three flow regimes in the channel. The three flow regimes are (1) single-phase liquid, (2) two-phase fluid and (3) single-phase vapor. These regimes are spatially oriented with the two-phase regime lying between the upper single-phase vapor and the lower single-phase liquid. Three regime interface locations are determined: (a) subcooled liquid - two-phase fluid which separates regime 1 and 2, (b) twophase - single-phase vapor which separates regime 2 and 3, (c) subcooled liquid-steam which specifies the interface at the start of the calculation. In general, BART couples the continuity equations and the energy equation with the assumption of constant pressure to obtain time dependent fluid conditions. Based on our review of the 1-D hydraulics model, the staff has concluded that the physical interpretation of the reflood phenomena as presented in the BART (1-D) model and the analytical model of the reflood hydraulics are consistent with that of other reflood analysis methods (Ref. 2) and are acceptable. The assumption of a constant system pressure made in BART may preclude consideration of the oscilliating gravity reflood phenomena (Ref. 4). However, for the case of forced reflood which is the phenomena simulated by BART, this assumption is consistent with other reflood models and is acceptable. The analytical model used in the two-phase regime in the particular drift flux form is a well developed model (Ref. 3) that is frequently used in the industry and is acceptable.

The two-phase regime is subdivided into film, transition and droplet flow regimes. These three regimes do not encompass all possible expected flow patterns. However, the staff concludes that it is an adequate representation for the reflood heat transfer calculation.

In summary, the 1-D hydraulics model uses a conventional approach in determining the fluid behavior and is, therefore, acceptable.

## 2.2 The Liquid-Vapor and Rod-Fluid Interaction Models

The hydraulic analytical model requires, for its solution scheme, the relative liquid-vapor velocity and the volumetric heat fluxes. All of these are needed in the two-phase regime solution whereas only the heat

sources are needed in the two single-phase regime calculations. The review of the interaction models used in the BART hydraulic model is discussed as follows:

## 2.2.1 The Liquid-Vapor Interaction Modeling

The liquid-vapor modeling provides values for the relative velocity, the volumetric flux and the void fraction which are required to solve

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the two-phase hydraulic equation. In the interaction modeling the two-phase regime is separated into (1) below the quench front, (2) at the quench front and (3) above the quench front.

The "below quench front" regime uses a correlation (Ref. 5) to predict. the local void fraction. The correlation was developed for small break LOCA conditions, which include relatively high pressure (above 400 psia). This correlation may underestimate the void fraction, which, in turn would allow more liquid to be transported to the quench front regime and may account for BART's tendency to slightly overpredict the quench front elevation. However, since BART is used to calculate the reflood heat transfer during LOCA conditions and the data comparisons show that BART overpredicts the peak cladding temperature and underpredicts the heat transfer coefficient for most of the data compared (see Section 2.7 for further discussion), we therefore, conclude that it is adequate to use the correlation in Reference 5 to calculate the local void fraction. In the "above the quench front" model the droplet size is a function of the Weber number (We), i.e., the droplet breakup will occur when the calculated Weber number exceeds a critical Weber number. BART uses a fixed value of 7 for critical Weber number. For nonviscous fluid, Wallis (Ref. 3) suggests a minimum value of 12 for the critical Weber number. Further, Wallis indicates that the viscosity has a stabilizing effect which tends to increase the critical values of the Weber number at which droplets breakup. In response to the staff's questions on the use of a lower critical Weber number for defining the droplet breakup, Westinghouse indicates that the calculated droplet Weber number does not exceed 2.0. Thus, the Weber number of the flow does not approach either of the critical Weber number values that would result in droplet breakup. Since the calculated droplet Weber number does not exceed 2.0, the question as to which critical Weber number, 7.0 or 12.0, is more appropriate becomes a moot question since the critical Weber number does not enter into the calculation.

## 2.2.2 The Rod Surface-Fluid Interaction

The rod heat release models at elevations other than at the quench front are quite conventional. The only modest elaboration is in the introduction of the droplet induced turbulence in the friction factor in an attempt to improve the droplet regime heat transfer correlations by bringing them into closer agreement with the observation that the actual heat transfer in the droplet regime, as seen in the data, was considerably larger than that predicted by the existing models and correlations. The enhanced, more efficient, droplet heat transfer model will probably result in an earlier quench prediction by BART. The BART data analysis (see Section 2.7 for further discussion) indicates that BART does not overestimate the droplet regime heat transfer as evidenced by the fact that BART consistently overpredicts the measured upper elevation clad temperatures in the FLECHT test series (Refs. 7, 8 and 10).

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The correlations used for the various heat transfer coefficients are conventional and accepted within the industry and are used within their region of applicability.

In summary, the BART rod clad surface heat transfer away from the quench front is straightforward and conventional (Ref. 9). The staff has found that there is no significant flaw in the development, and has concluded the model to be acceptable.

The models used for the radiation heat transfer are standard fourth power temperature laws expressing net radiative heat transfer. The staff has concluded that this model also is acceptable.

## 2.2.3 Data Comparison for Parameter Assessment

Only a single test was presented in the BART topical as the basis for the parameter assessment. The single reported test is one of several constant

reflood rate tests. The results indicate that the BART prediction of the quench front is not sensitive to the variations in initial droplet diameter  $(D_d)$  and dispersed regime initial void fraction (a). The precooling effects of the droplet heat transfer model are very minimal. Based on the Westinghouse submitted comparisons, the choice of  $D_d = .0035'$  and a = .90 provides a more accurate, yet still conservative estimate of the rod clad surface temperatures. The staff finds that the values actually chosen,  $D_d = .0035'$  and a = .99 add a conservative margin to the calculation and concludes that they are acceptable.

#### 2.3 The One-Dimensional Rod Heat Transfer Model

In addition, a flow channel housing heat conduction model has also been developed to allow BART to more accurately model some reflood experiments in which the effects of the hot surrounding channel cannot be neglected.

Based on our review, we find that the treatment of the one-dimensional heat conduction equation is consistent with other codes in the industry and we conclude that the model is acceptable.

## 2.4 The Quench Front Model

The degree of success of the simulation of the reflood phenomena has depended upon the accuracy and completeness of the quench front model. In BART the quench front is characterized by a very steep thermal gradient in the fuel rod clad surface temperature. The gradient is defined as a sudden drop from temperatures in excess of 1000°F to fluid saturation

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temperature over a very short axial distance. BART identifies and tracks this gradient and defines the quench front to be the region in close proximity to this gradient. The quench front includes a localized axial as well as radial heat conduction model.

The fundamental assumption of the quench front model in BART is that the surface heat transfer at the quench front is controlled by the liquid flow rate into the region which in turn is controlled by the vapor film flowrate out of the region. The vapor film flowrate is calculated using the Taylor instability model (Ref. 6).

Based on our review, the staff finds that, although the BART quench front model is very different from the more conventional moving mesh or refined mesh models, the methods of the solution of the two-dimensional heat conduction equation in the ouench front model are typical methods used in the industry adequate and that the transformation for the gap and clad are correct and, therefore, the methods of solution of the quench front model are acceptable.

### 2.5 The Quench Front Heat Transfer Model

The rod clad surface heat transfer region near the quench front is illustrated in Figure 3.3-2 of Reference 1. The heat transfer from the rod clad surface to the fluid near the quench front is determined through the use of correlations. The Dittus-Boelter and Rohsenow correlations are used to model the subcooled forced convection and nucleate boiling in Region I. The boundary between Region I and Region II is specified as the elevation at which the vapor generated on the rod begins to take on the characteristics of a vapor film. In this region, the heat transfer is determined from the minimum of the Rohsenow nucleate boiling correlation and a BART transition boiling equation developed for Region III. This transition boiling equation provides the surface heat flux as a function of local vapor generation and wall superheat. The upper boundary of Region III is defined as the elevation at which the transition heat flux drops below the Berenson film boiling heat flux. The Berenson film boiling correlation is used in Region IV.

In the BART model of the transition boiling phenomena at the quench front, the following areas have been addressed: (1) to identify a tractable phenomenon that governs the quench front heat transfer behavior and (2) to develop a model allowing for the eventual transition to the film boiling heat transfer mode. BART proposes the stability of the vapor film as the phenomenon that governs the quench front heat transfer. This model proposes that the maximum heat flux at the quench front is limited by the ability of the vapor to either escape from the film or recondense in the bulk fluid. The right-hand side of equation 3.3-11 in Reference 1 represents that heat flux. To address the second concern itemized above, BART proposes a general transition boiling curve (equation 3.3-46 in Reference 1). This equation is a function of the coefficient B and provides the right-hand side of equation 3.3-11. The approach can be described as follows: The right-hand side of equation 3.3-11 is determined, then B is varied in the heat flux until the left hand side of equation 3.3-11 equals the right-hand side. Thus the value of B is established and the transition boiling curve is established and the heat flux at the quench front is determined. This quench front heat flux is compared with that predicted using the Berenson film boiling heat transfer coefficient at the same wall superheat. When the heat flux of equation 3.3-46 equals the heat flux of the Berenson film boiling calculation, BART ceases using equation 3.3-46 and transfers to the Berenson film boiling transition.

Based on our review, the staff finds that (1) the physical modeling that is used to develop the quench front heat transfer model is selfconsistent, (2) the rod clad surface heat transfer, other than that for the vapor constrained boiling zone, is determined from correlations, that have been widely used in the industry (the Dittus-Boelter, Rohsenow and Berenson correlations have been used by numerous code development efforts) and (3) the determination of the boiling region

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heat flux as a function of fluid properties and vapor generation is consistent with the entire quench front model developed for BART. The staff, therefore, has concluded that the quench front heat transfer model is acceptable.

### 2.6 The Grid Spacer Heat Transfer Model

Westinghouse has submitted a report with their transmittal letter of May 14, 1982 to augment the BART topical report WCAP-9561 with an amendment specifying a grid spacer heat transfer effects model, which has been revised and submitted for review in a letter of August 18, 1982. The grid spacer model modification was introduced to reduce the conservatism inherent in the BART model which tended to overpredict the core upper elevation clad temperatures. It is expected that the grid spacers will contribute to the desuperheating of the upper elevation vapor steam and result in a more significant heat transfer from the upper elevation cladding. During the review of the grid spacer model, the staff concluded that more calculations for data comparison were necessary to justify the accuracy of the grid spacer heat transfer model. In response to the staff's evaluation, Westinghouse submitted an amendment revising the model and providing comparisons to data in a transmittal of August 12, 1983. The staff is reviewing this amendment to the grid spacer model and will provide the safety evaluation result in early 1984. The grid spacer model should not be to used in the BART code prior to the staff's review and approval of the revised model submitted on August 12, 1983.

### 2.7 Data Comparisons for Code Verification

The data base used for the BART code verification included 19 variable and constant reflood rate tests which cover a parameter range typical of

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expected PWR conditions. The ranges of the data base are:

Pressure (psia)	20-60
Initial Temperature (°F)	1100-1600
Initial Power (kw/ft)	0.45 - 1.2
Inlet Subcooling (°F)	20-140
Reflood Rate (in/sec)	0.6 - 1.5

The data comparisons fall into four categories: (1) FLECHT experiments, (2) G-2 17x17 experiments, (3) Semiscale experiments and (4) FLECHT-SEASET experiments. The comparisons of BART calculations with the average clad temperature data are quantified as either an overprediction (+); an underprediction (-), or exact (+-) and are summarized in Table 1. The results of the data comparison are discussed as follows:

## 2.7.1 FLECHT Experiment Data Comparison

The FLECHT comparisons presented are from both the cosine power test series (Ref. 8) and the skewed power test series (Ref. 7), which are conducted for 0.422 inch OD fuel rods (14x14 and 15x15 fuel assemblies). Overall, the BART clad temperature predictions are in good agreement with the test data. For the constant reflood rate analyses, the six and eight foot elevation comparisons show that the clad temperature overpredicts up to 100°F when compared with the temperature data, which are the average clad temperatures of all rods more than two rows away from the test house and more than one row away from the failed heated rods. The clad temperature at the ten foot elevation is consistently overpredicted by 100° to 300°F. For the variable reflood rate analysis, the data comparison also shows that the calculated clad temperature is higher than the measured average temperature at the ten foot elevation at which the maximum clad temperature occurs.

The BART code shows spikes in the calculated results of the heat transfer coefficients. The spikes are indicative of the discontinuous heat

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transfer regime transitions. However, the overall BART predictions are in good agreement with the heat transfer coefficient data.

### 2.7.2 G-2 Experiments Data Comparison

The BART comparison with the G-2 17x17 bundle data show a consistent behavior. BART consistently underpredicts the surface heat transfer coefficient and hence overpredicts the clad surface temperature.

### 2.7.3 Semi-Scale Experiment Data Comparison

The BART Semi-Scale test predictions also show good agreement with the test data. The quenching clad temperature drop predicted by BART for the 14" level for the S-03 comparisons is much better than in either the G-2 or the FLECHT comparisons.

### 2.7.4 FLECHT-SEASET Data Comparison

The FLECHT-SEASET comparisons presented are from the cosine test series (Ref. 10), which are conducted on 0.374 inch OD fuel rods (17x17 fuel assembly). The two tests for data comparison have variable reflood rates. The analyses show that BART consistently overpredicts the average clad surface temperature.

## 2.7.5 Clad Temperature Rise Comparison

Westinghouse has submitted a comparison of BART to data for the clad temperature rise (at the elevations where the clad temperature were measured) to show the conservatism of the BART predictions (Figure 1). The data base was drawn from 17 test series and grouped as data sets from (1) FLECHT-SEASET tests, (2) G-2 tests, (3) FLECHT skewed power tests and (4) FLECHT cosine power tests. Data sets (1), (2) and (3) were obtained from uniform radial power profile tests and the comparisons of the BART calculations to the data are presented as a composite in Figure 1. FLECHT cosine tests, data set (4), are nonuniform radial power tests. To show the hot rod effect, the cosine power data comparisons are identified separately from other test comparisons in Figures 1. The staff has reviewed the results for the clad temperature rise comparisons and has found that BART conservatively predicts 90%, 80% and 88% of FLECHT-SEASET/skew/G-2 composite data, low reflood rate cosine test data and all-test composite data in the range of applicability, respectively.

## 2.8 The Consistency of the Models in BART with the Appendix K Requirements

For the reflood code, the Section I.D.5 of Appendix K requires that:

"New-Correlations or modifications to the FLECHT heat transfer correlations are acceptable only after they are demonstrated to be conservative, by comparison with FLECHT data, for a range of parameters consistent with the transient to which they are applied. During refill and during reflood when reflood rates are less than one inch per second, heat transfer calculations shall be used based on the assumption that cooling is only by steam, and shall take into account any flow blockage calculated to occur as a result of cladding swelling or rupture as such blockage might affect both local steam flow and heat transfer."

The staff review of the testimony and concluding statements presented at the ECCS Rulemaking Hearing concluded that Section I.D.5 of Appendix K permits experimental data to be used as a basis for acceptability of reflood models. We also have found from the low flow FLECHT data (Refs. 7, 8, 10) that there is no difference between heat transfer for reflooding rates above one inch/sec vs. less than one inch/sec. We, therefore, evaluated the BART heat transfer model without flow blockage for flooding rates above and below one inch/sec on the basis of (1) conservative agreement with experimental test data including low flow FLECHT test data (Refs. 7, 8 and 10) and (2) conformance to physical principles with consideration given to the role of steam cooling. Based on our review, we have found that the BART code has met both of the above criteria and conclude that the BART heat transfer model is consistent with the cited requirements in Appendix K. The heat transfer model with flow blockage is addressed in Section 3.0.

## 3.0 ECCS EVALUATION MODEL USING BART

The purpose of using BART in the ECCS evaluation model is to provide a more mechanistic description of reflood heat transfer than is possible with current FLECHT correlations. Since much of the mechanistic description depends on current fuel rod surface conditions, it is not a simple matter to replace the FLECHT correlation in LOCTA with a mechanistic model. Therefore, the fuel rod conduction and surface heat transfer must be run together in BART. However SART does not have a gap heat transfer model or cladding swelling model as required by Appendix K. Therefore BART is run and cladding temperature and heat transfer coefficients are calculated. The heat transfer coefficient history is then applied to LOCTA and "Appendix K" cladding temperatures are generated. The staff questioned this "loose coupling" without feedback in LOCTA, since heat transfer coefficients are dependent on cladding temperature in a mechanistic model. Westinghouse responded to these concerns in References 11 and 12. In Reference 12, Westinghouse was able to assemble a "tightly coupled" BART/LOCTA model and showed that the loosely coupled model proposed at this time is conservative by 134°F. The "tightly coupled" model differs from the "loose coupling" model in that BART and LOCTA are run simultaneously instead of sequentially. We therefore find the loose coupling acceptable for evaluation model calculations.

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The WREFLOOD code was also modified to be more compatible with BART. In particular the empirical "carryover rate fraction" correlation was replaced by a more mechanistic "core mass entrainment" correlation. Both correlations are defined as the core exit mass flow rate divided by the core inlet mass flow rate. However, the "core mass entrainment" correlation is a better match to the BART model.

An explicit metal stored energy wall heat model was also added to the downcomer and lower plenum in WREFLOOD. The mass and energy equation in WREFLOOD were modified to account for flow diversion due to blockage. That flow diversion is calculated in the same manner as that currently approved in LOCTA.

The BART-WREFLOOD combination wa; used to predict two FLECHT-SET experiments. The results showed the code predictions to be conservative with respect to the data (low flooding rates and high cladding temperature compared to the data). We find that these validation results, combined with our knowledge of the inherent conservatism in the previously approved WREFLOOD code, give' the staff reasonable assurance that the calculational results from the BART-WREFLOOD combination will be sufficiently conservative. We point out however, that a substantial amount of separate effects and integral systems data has been generated in the 2D/3D program, the FLECHT program and the NRU program. Many more experiments were used to develop the old empirical carry-over rate correlations. We, therefore, require that Westinghouse provide additional comparisons of the BART/WREFLOOD prediction capability to a wider range of data in order to confirm our conclusion. We request Westinghouse to advise within 45 days of receipt of this SER of their plan and schedule to submit this additional information. We also recommend that conversations with the staff begin immediately to decide upon the specifics of such a program.

In order to account for steam flow diversion due to blockage at flooding rates less than one inch per second, modifications were made to the BART code. The mass continuity equation was properly altered to allow steam

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flow into or out of the channel at each appropriate elevation. The methodology for determining the value of this flow diversion is identical to that approved for use under these conditions in the LOCTA code. No direct heat transfer is allowed to the water droplets from the cladding surface. We conclude this meets the steam cooling requirements of Appendix K. However, we believe comparison to appropriate blockage experiments is required to confirm and help quantify the degree of conservatism in the model. As previously stated, Westinghouse has committed to provide these comparisons within 9 months of issuance of this SER. We find this commitment acceptable.

The BART calculations for data comparison show typical nodal lengths for the fuel rod model to be about 6 inches. In response to the staff's request, Westinghouse did a sensitivity study and indicated that the finer the nodal length is specified, the higher calculated clad temperature BART will predict. The result draws a conclusion that the nodal length specification is a critical input to determine the BART prediction result. While it has been demonstrated that use of a nodal scheme with nodal length of six inches is adequate, it is required that the BART calculation should use a nodal scheme with the nodal length of six inches or less for the licensing applications.

## 4.0 SUMMARY AND CONCLUSIONS

The staff has reviewed the WCAP-9561 (Ref. 1) topical report and the additional supporting information submitted by Westinghouse. Based on this review, we have concluded that BART code is acceptable for use in reactor licensing applications. The conclusions are based on the following:

 The models developed for BART have been reviewed and found acceptable for their (1) adequacy of physical assumptions and hypothesis, (2) correctness of thermal-hydraulic equations and (3) appropriateness of the application of correlations.

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- The data comparisons for BART code verification have been found acceptable for their (1) adequacy of data base, (2) good agreement between the data and the BART calculations.
- The BART predictions are in good agreement with the measured average clad temperature data and the average heat transfer coefficient data.
- The clad temperature rise comparisons demonstrate that BART calculations conservatively predict 88% of the test data presented in the range of applicability.

However, the following restrictions are imposed to the BART applications:

- The BART code is applicable to only the range of operation bounded by the data base range tabulated in Section 2.7 of this report.
- 2. The BART calculation is sensitive to the nodal length. The calculated results show that use of a finer nodal length results in higher clad temperature predictions by BART. In the data comparison, the nodal length of six inches is used to verify the BART code. To address the concern of the sensitivity of the nodal length, the BART calculation should use a nodal scheme with the length of six inches or less for the licensing application.
- The grid spacer model is not allowed for use in the BART code prior to the staff review and approval of the revised grid spacer model submitted on August 12, 1983.

We conclude that the model described in the referenced documents and appropriately restricted in this report may be applied to PWR's using Westinghouse fuel and which use only cold leg ECCS injection. When applying this model and the 1981 model the condition of no single failure as being the worst case should be considered. This model does not necessarily replace the 1981 model (Ref. 13) and either model would be acceptable to demonstrate compliance with 10 CFR 50.46. It should be noted that certain restrictions on the grid spacer model are described in this SER. Also, additional confirmatory validation of the BART/WREFLOOD model for reflood rates less than one inch/sec is required. Westinghouse has committed to provide the necessary additional confirmatory validation information within 9 months of the issuance date of this SER. We find this committment acceptable. We conclude that the model described in this SER conforms to the requirements of Appendix K to 10CFR50 and is therefore acceptable.

## TABLE 1

Test Series	<u>Te</u>	st No.	Reflood Rate (IN/SEC)	Peak Clad Temperatu Prediction		Peak Clad Temperatur Prediction	era ture on
					Elev	ation	(ft)
				4	6	8	10
Cosine	5132		1.0	+-	+	-	+
	6638				+	+	+
	4831		1.5	1.1	+-	+-	+
	. 7934		.62	+-	+	· + .	+
	5342		.8	-	+	-	+
Skewed	13303		1.5	+	+	+	+
	13404		1.0	+	+	+	+
	13609		1.0	+	+	+	+
	11618		1.5	+	+	+	+
	15305		.8	+	+	+	+
	15713		1.0	+-	+-	+	+
	16945	Variable	6.7 (5 sec)				
			.8 (t 5 sec)	-	-	-	+
FLECHT-	32233	Variable	6.36 (5 sec)				
SEASET			0.82 (t 5 sec)		+-	+	+
	32335	Variable	6.53 (5 sec)				
			0.98 (5 sec t 200 se	ec) +	+	+	+
			0.62 (t 200 sec)				

# DATA COMPARISON FOR CODE VERIFICATION

# TABLE 1 (continued)

TEST SERIES	TEST NO.	REFLOOD RATE	PEAK	PEAK CLAD TEMPERATUR PREDICTION		
				Eleva	tion (f	t)
G-2	538	1.3	4	6	8	10
			+	+	+	+
	561	1.0	+	+ - '	+	+
			1.2	2.4	3.3	
Semiscale						
Mod-1	S-03-a	1.0	+-	-	+	
	S-03-h	1.0	+	+-	+	
	S-03-d	1.0	+-	-	+-	

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### 5.0 REFERENCES

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### ABSTRACT

This report describes the BART computer code, its physical models, and its application to PWR reflood analysis. In addition, the code is compared with data from a set of reflood experiments to demonstrate its applicability to this type of transient.

This report also contains the additional analyses and descriptions produced as a result of the NRC review of a preliminary version of this report.

### FOREWORD

To preserve the continuity and readability of this report, an effort has been made to incorporate the results of the NRC review in the body of the text. Some sections have been expanded, and other sections have been added where appropriate to describe analyses performed during the review. Parts of the preliminary report which required additional clarification and detailed description during the review have been expanded in this final report.

Appendix A lists the nomenclature used in this report and appendix B contains answers to a list of questions generated during the NRC review pertaining to sections 2, 3, 5, and 6. Appendixes C and D contain additional information on fluid interfaces and on the quench front model. These two appendixes have been added to clarify these subjects.

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## SECTION 1 INTRODUCTION

In an analysis of the loss of coolant accident in a PWR, it is usual to divide the problem into two parts called the blowdown transient and the reflood transient. In the blowdown transient, the reactor coolant system depressurizes from its initial state to the pressure within the containment structure, expelling most of the water it originally contained during this time. Near the end of this process, accumulators begin to fill the lower plenum of the reactor vessel with cold water. When the lower plenum is full and water begins to enter the core, the reflood transient begins.

The following regions, characteristic of the reflood transient, are listed in the order that a particle of fluid entering the core would encounter them (figure 1-1). The first region is one in which the fuel rod surface temperature is near the fluid satura' in temperature. Liquid is heated or boils in this region due to decay p generated with the rods. Next is a quench region, where the fuel rod surrace temperature changes from near the fluid saturation temperature to a value on the order of a 1000°F, usually over a very short axial distance. In this region, the fluid undergoes complex changes in flow regime because the rod surfaces can no longer be wet by the liquid and because high heat flow rates exist at the quench front. Over a wide range of core inlet mass velocities, the final state of the fluid as it passes up through the bundle is characterized by a highly dispersed liquid phase and a highly superheated vapor phase.

The heat transfer in a nuclear core during reflood is calculated, in most design analyses, using correlations developed from reflood tests. Given several boundary conditions, such as core inlet flow and temperature and core pressure, the fuel rod temperature during reflood can be calculated using these correlations.



Figure 1-1. Typical Conditions in Rod Bundle During Reflood

An important drawback of this method, however, is that its use must be limited to the range of experimental data upon which the correlations were based. Outside this range, several conservative assumptions must be made before the model can be used to calculate peak cladding temperature. In some instances the method becomes unusable because of its lack of detail.

A reflood core heat transfer model fully applicable to a wide range of conditions must be capable of addressing in as much detail as possible the important phenomena occurring in a core during reflood and the way they interact with one another to affect the heat transfer from the fuel rods. Exact physical equations must be used to the greatest extent possible before resorting to correlations to achieve closure. Finally, empirical constants should be clearly identified and applied in a consistent manner.

Several computer codes have been developed recently to calculate conditions during a reflood transient. (1,2,3) These codes can be distinguished one from the other primarily by the manner in which the interactions between liquid and vapor and fluid and solid surfaces are treated. In particular, the methods used to calculate the progression of the quench front vary from one code to the other.

The following sections describe the models contained in the BART computer code, which is designed to calculate fluid and heat transfer conditions in the core during reflood. The first sections describe how the basic two-phase fluid conservation equations are used in BART. These equations require several vapor-liquid and fluid-solid surface interaction terms which, with the exception of the quench front region, are described in following sections.

A number of empirical constants arise from development of these models. These constants are determined by comparing predictions with a selected number of FLECHT tests. In these calculations, the quench front movement is determined from data and the overall heat release supplied to the code.

The behavior of the quench front is crucial in determining the core heat transfer during reflood. A separate section is devoted to the description of the thermo-hydraulic models used to determine quench front progression in

1-3

BART. The empirical constants which arise from the model development are determined by comparing predicted quench front velocity to data from selected FLECHT tests. In this case, rod conditions above the quench front are determined from data and supplied to the quench model.

The primary motive for developing a mechanistic core heat transfer model is to calculate the peak cladding temperature in the core during a postulated loss of coolant accident. To accomplish this, a design procedure is developed with uses BART in conjunction with other emergency core cooling system (ECCS) codes. Some model changes are required in BART and the other ECCS codes to make them compatible and to comply with Appendix K requirements.

Finally, the BARI code and the proposed design procedure are verified by comparison with several reflood tests. Forced flooding tests from several different experiments are used to verify BART, and verification of the design procedure is accomplished by comparison with several FLECHT SET tests.

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## SECTION 2 NONEQUILIBRIUM TWO-PHASE FLUID MODEL

## 2-1. BASIC EQUATIONS

The reflood heat transfer, which determines the peak cladding temperatures during the reflood phase of the loss of coolant accident, is governed by the local fluid conditions in the core. The one-dimensional fluid model used to calculate the fluid properties as a function of time and space is described in paragraphs 2-2 through 2-10.

#### 2-2. Derivation of Equations

Local pressure variations are assumed to be small so that pressure is constant in time. As such, only the continuity and energy equations, along with equations of state and an auxultary equation for relative velocity between liquid and vapor (in a two-phase region), need be considered.

One-dimensional equations are written for the following:

- o Single-phase region (either liquid or vapor)
- o Two-phase region
- o Fluid Interfaces between different regions

2-3. <u>Derivation of Equations for a Single-Phase Region</u> -- For a single-phase region, equations for continuity and energy are written as follows:

Continuity equation:

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho U)}{\partial z} = 0$$
 (2-1)

Energy equation:

$$\frac{\partial(\rho H)}{\partial t} + \frac{\partial(\rho U H)}{\partial z} = q'''$$
 (2-2)

q''' is the rate of heat transferred to the fluid per unit volume, U is the single-phase velocity,  $\rho$  is the single-phase density, and H is the enthalpy.

The above two equations are combined to yield this equation:

$$\frac{\partial U}{\partial z} = -\frac{q^{\prime \prime \prime}}{\rho^2} \frac{\partial H}{\partial \rho}$$
(2-3)

q''' to be used in the flow model is obtained from the heat transfer model.

Solutions of equations (2-3) and (2-1) give U and  $\rho$ , respectively. Although not formally required, equation (2-2) is solved to obtain single-phase enthalpy because property tables used are of the form  $\rho = \rho(H,P)$ .

2-4. <u>Derivation of Equations for a Two-Phase Region</u> -- Equations for the two-phase region are derived in the drift-flux form for the following three flow regimes: inverted annular, transition, droplet. The relative velocity between vapor and liquid, required in the drift-flux formulation, is determined as given in paragraph 2-13.

Liquid in the two-phase region is assumed to be at saturation conditions, so that  $\rho_0$  and  $H_0$  are constants.

Continuity and energy equations are written for the two-phase region as follows.

The continuity equation for vapor is

$$\frac{\partial}{\partial t} (\alpha \rho_{v}) + \frac{\partial}{\partial z} (\alpha \rho_{v} U_{v}) = \Gamma_{v}$$
(2-4)

where  $\Gamma_{\rm V}$  is the amount of steam produced during boiling per unit time and volume.

The continuity equation for liquid is

$$\frac{\partial}{\partial t} (1 - \alpha) \rho_{\varrho} + \frac{\partial}{\partial z} \left[ (1 - \alpha) \rho_{\varrho} U_{\varrho} \right] = -\Gamma_{v}$$
(2-5)

The void fraction,  $\alpha$ , is the volume of vapor per unit volume of the mixture. Subscript v refers to gas, and subscript  $\ell$  refers to liquid.

The energy equation for the mixture is

$$\frac{\partial}{\partial t} \left[ \alpha \rho_{V} H_{V} + (1 - \alpha) \rho_{\varrho} H_{\varrho} \right] + \frac{\partial}{\partial z} \left[ \alpha \rho_{V} U_{V} H_{V} + (1 - \alpha) \rho_{\varrho} U_{\varrho} H_{\varrho} \right] = q^{\prime \prime \prime} (2-6)$$

Fy is related to q''' boil by

$$\Gamma_{v} = \frac{q'''boll}{H_{gv}}$$

where  $H_{QV}$  is the latent heat of vaporization and q'' boil is the rate of heat per unit volume transferred to the liquid from external sources such as rods and superheated vapor.

q''' boil and q''', which are the total heat rate per unit volume supplied to the two-phase mixture from external sources, couple the fluid flow model with the rod heat transfer model and are defined in paragraph 2-16.

By defining the volumetric fluxes

$$j = \alpha U_{v} + (1 - \alpha)U$$
$$j_{v} = \alpha U_{v}$$
$$j_{a} = (1 - \alpha)U_{a}$$

we obtain (from the continuity equations) the following equations:

$$\frac{\partial}{\partial t} \left[ \alpha \rho_{v} + (1 - \alpha) \rho_{\varrho} \right] + \frac{\partial}{\partial z} \left[ \alpha \rho_{v} U_{v} + (1 - \alpha) \rho_{\varrho} U_{\varrho} \right] = 0 \qquad (2-7)$$

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and

$$\frac{\partial \mathbf{j}}{\partial z} + \alpha \frac{\partial}{\partial \mathbf{t}} \ln \rho_{\mathbf{v}} + \mathbf{j}_{\mathbf{v}} \cdot \frac{\partial}{\partial z} \ln \rho_{\mathbf{v}} = \Gamma_{\mathbf{v}} \frac{(1 - \rho_{\mathbf{v}}/\rho_{\mathbf{g}})}{\rho_{\mathbf{v}}}$$

$$= \frac{q^{\prime \prime \prime} boll}{H_{\mathbf{g}} \mathbf{v}^{\rho} \mathbf{v}} \left(1 - \frac{\rho_{\mathbf{v}}}{\rho_{\mathbf{g}}}\right)$$
(2-8)

Let

$$Q_{K} = -\frac{H_{QV}\rho_{V}}{1-\rho_{V}/\rho_{Q}}$$

so that  $Q_{\mathbf{K}}$  is a function of fluid properties only.

Further, we define static mixture density as

$$\rho_{s} = \alpha \rho_{y} + (1 - \alpha) \rho_{0} \tag{2-9}$$

and a flow mixture density and void fraction as

$$\rho_{j} = \alpha_{j} \rho_{v} + (1 - \alpha_{j}) \rho_{\varrho}$$
(2-10)

such that

$$\rho_{1} j = \alpha \rho_{0} U_{0} + (1 - \alpha) \rho_{0} U_{0}$$
(2-11)

 $\alpha_{j}$  is related to  $\alpha$  by

$$\alpha_{j} = \frac{\alpha U_{v}}{j} = \frac{j_{v}}{j}$$

With the above definitions, equations (2-7) and (2-8) become

$$\frac{\partial \rho_{s}}{\partial t} + \frac{\partial}{\partial z} \left( \rho_{j} \right) = 0$$
 (2-12)

and

$$\frac{\partial j}{\partial z} + \alpha \frac{\partial}{\partial t} \ln \rho_{V} + j_{V} \frac{\partial}{\partial z} \ln \rho_{V} = - \frac{q^{++} boil}{Q_{K}}$$
 (2-13)

where

$$Q_{K} = -\frac{\rho_{V}H_{QV}}{1 - \rho_{V}/\rho_{Q}}$$
(2-14)

When relative velocity is defined as

$$U_r = U_v - U_g$$

we have

$$U_{ij} = j + (1 - \alpha)U_{j}$$

and

$$U_q = j - \alpha U_r$$

From the energy equation (2-6), we have

$$\Gamma_{\mathbf{v}} (\mathbf{H}_{\mathbf{v}} - \mathbf{H}_{\mathbf{g}}) + \rho_{\mathbf{v}} \left[ \alpha \frac{\partial \mathbf{H}_{\mathbf{v}}}{\partial t} + \mathbf{j}_{\mathbf{v}} \frac{\partial \mathbf{H}_{\mathbf{v}}}{\partial z} \right] = q^{1/2}$$

Because superheat may occur,

$$\Gamma_{\mathbf{v}} (\mathbf{H}_{\mathbf{v}} - \mathbf{H}_{\mathbf{Q}}) = \Gamma_{\mathbf{v}} \left[ \mathbf{H}_{\mathbf{v}} - \mathbf{H}_{\mathbf{v}sat} + \mathbf{H}_{\mathbf{v}sat} - \mathbf{H}_{\mathbf{Q}} \right]$$
$$= \frac{\mathbf{q}^{\prime \prime \prime} \mathbf{bo11}}{\mathbf{H}_{\mathbf{Q}v}} \left[ \mathbf{H}_{\mathbf{v}} - \mathbf{H}_{\mathbf{v}sat} + \mathbf{H}_{\mathbf{Q}v} \right]$$
$$= \mathbf{q}^{\prime \prime \prime \prime} \mathbf{bo11} \left[ \mathbf{1} + \frac{\mathbf{H}_{\mathbf{v}} - \mathbf{H}_{\mathbf{v}sat}}{\mathbf{H}_{\mathbf{Q}v}} \right]$$

The final system of equations (relative velocity approach) is

$$\frac{\partial j}{\partial z} + \alpha \frac{\partial}{\partial t} \ln \rho_v + j_v \frac{\partial}{\partial z} \ln \rho_v = -\frac{q^{\prime\prime\prime} boil}{Q_K}$$
 (2-i5)

$$\frac{\partial \rho_s}{\partial t} + \frac{\partial}{\partial z} (\rho_j \mathbf{j}) = 0$$
 (2-16)

$$\rho_{\mathbf{v}} \left[ \alpha \frac{\partial H_{\mathbf{v}}}{\partial t} + j_{\mathbf{v}} \frac{\partial H_{\mathbf{v}}}{\partial z} \right] = q^{\prime \prime \prime} - q^{\prime \prime \prime} boll \left[ 1 + \frac{H_{\mathbf{v}} - H_{sat}}{H_{\mathbf{v}}} \right]$$
(2-17)

The equation of state is

$$\rho_{\rm v} = \rho_{\rm v} (\rm H, P) \tag{2-18}$$

Definitions:

$$\rho_{s} = \alpha \rho_{v} + (1 - \alpha) \rho_{\varrho}$$

$$\rho_{j} = \alpha_{j} \rho_{v} + (1 - \alpha_{j}) \rho_{\varrho}$$

$$\alpha_{j} j - \alpha U_{v}$$

$$U_{v} = j + (1 - \alpha) U_{r}$$

$$Q_{K} = - \rho_{v} H_{\varrho} v^{\prime} (1 - \frac{\rho_{v}}{\rho_{\varrho}})$$

$$\mathbf{j}_{\mathbf{v}} = \mathbf{\alpha} \left[ \mathbf{j} + (\mathbf{1} - \mathbf{\alpha}) \mathbf{U}_{\mathbf{r}} \right]$$

Equations (2-15), (2-16), (2-17), and (2-18), with the relative velocity  $U_r$  and heat transfer rates q'' and q'' boil given in paragraphs 2-12 and 2-16, are solved for the four unknowns:  $\alpha$ , j,  $H_v$ , and  $\rho_v$  using the above definitions.

In the droplet acceleration region, equations (2-15) and (2-16) are replaced by equations (2-4) and 2-5.

Using the following equation for vapor generation,

$$\Gamma_{v} = \frac{q^{\prime \prime \prime} boil}{H_{gv}} = \frac{6 q^{\prime \prime} gv^{(1 - \alpha)}}{H_{gv} D_{d}}$$

equations (2-4) and (2-5) are written as

$$\frac{\partial}{\partial t} (\alpha \rho_{v}) + \frac{\partial}{\partial z} (\alpha \rho_{v} U_{v}) = \frac{6 q'' \varrho_{v} (1 - \alpha)}{H_{\varrho v} U_{d}}$$
(2-15a)

$$\frac{\partial}{\partial t} (1 - \alpha) \rho_{\varrho} + \frac{\partial}{\partial z} [(1 - \alpha) \rho_{\varrho} U_{\varrho}] = - \frac{6 q''_{\varrho v} (1 - \alpha)}{H_{\varrho v} U_{d}}$$
(2-16a)

 $D_d$  is the drop diameter and q'' is the droplet heat flux (paragraph 2-11).

Equations (2-15a), (2-16a), (2-17), and (2-18), with U<sub>g</sub> as given in paragraph 2-12, give the four unknowns  $\alpha$ , U<sub>y</sub>, H<sub>y</sub>, and  $\rho_y$  as discussed in paragraph 2-6.

2-5. <u>Derivation of Equations for Fluid Interfaces</u> -- Three types of fluid interfaces can occur during a bottom flooding transient:

- Subcooled liquid/steam interface
- Subcooled liquid/two-phase interface
- o Two-phase/steam interface

Each of these interfaces is given special treatment to predict its movement and changing conditions across the interface.

The subcooled liquid/steam interface occurs early in the transient, prior to boiling. If  $z_F$  is the location of this interface, density is discontinuous across  $z_F$ , but fluid velocity is continuous.

Hence,

$$U_{2,z_{F}} = U_{v,z_{F}} = U_{z_{F}}$$
 (2-19)

where U is the velocity of this interface (front) given by  $\mathbf{z}_{\mathbf{F}}$ 

$$\frac{dz_F}{dt} = U_{Z_F}$$
(2-20)

On the liquid side

$$\frac{\partial U_{\varrho}}{\partial z} \bigg|_{z_{F}^{-}} = -\frac{q'''_{\varrho}}{\rho_{\varrho}^{2}} \frac{\partial H_{\varrho}}{\partial H_{\varrho}}$$
(2-21)

$$P_{\mathbf{g}} \frac{dH_{\mathbf{g}}, z_{\bar{r}}}{dt} = q'''_{\mathbf{g}}$$
(2-22)

Similarly, on the steam side

$$\frac{\partial U_{v}}{\partial z}\Big|_{z_{F}^{+}} = -\frac{q^{\prime\prime\prime}}{\rho_{v}^{2}}\frac{\partial H_{v}}{\partial \rho_{v}}$$
(2-23)

and

$$p_{v} \frac{dH_{v,z_{F}}}{dt} = q'''_{v}$$
 (2-24)

 $z_F$  refers to the liquid side and  $z_F$  refers to the steam side of the interface, and d/dt denotes differentiation along the particle path. The above equations, along with the equations of state, can be solved to obtain conditions on both the sides of the interface.

The subcooled liquid/two-phase interface appears when boiling first occurs.

The initial location of the saturation line is usually, but not necessarily, at the location of the subcooled liquid/steam interface.

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(2-26)

The fluid velocity at the saturation line is determined by integrating equation (2-3) from core inlet to the liquid/two-phase interface. Equation (2-27) then gives the movement and the new location of the saturation line. Along this interface  $\alpha = 0$ ,  $p = p_{g}$ . Thermal equilibrium is assumed in the region close to the saturation line on the two-phase side.

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The two-phase/steam interface appears along with the subcooled liquid/ two-phase interface when boiling occurs. Volumetric flux j is continuous across this interface, but the void fraction  $\alpha$  is discontinuous. The interface propagates at the liquid velocity. If  $z_F$  is the location of this interface (front), then

$$\frac{dz_F}{dt} = U_{R, z_F} = U_F$$
(2-28)

The requirement of continuous volumetric flux gives

$$j_{z_{F}} = j_{z_{F}} = 0_{v, z_{F}}$$
 (2-29)

To simplify front calculations, it has been assumed that no additional superheating of the vapor occurs on the two-phase side of the front in the region where front calculations are made.

#### 2-6. Numerical Solution Method

The fluid flow equations in the single-phase and two-phase regions are solved numerically to obtain the transfert fluid behavior in the core. When the fluid flow and heat transfer conditions at time t are known, the fluid flow equations are integrated to obtain conditions at time  $t + \Delta t$  using the heat transfer values at t. The quasi-steady-state heat transfer assumption allows the fluid flow equations to be solved independent of the heat transfer model

(2 - 27)

during the time increment  $\Delta t$ . The calculated fluid conditions are then fed to the heat transfer model to obtain the heat transfer values at time t +  $\Delta t$ .

The numerical solution is carried out by using a fully implicit differencing scheme for most of the fluid flow equations in BART. In general, for linear equations with constant coefficients, such methods are unconditionally stable. The stability of BART difference equations is discussed in paragraph 2-10.

Numerical solution procedure is given for single-phase and two-phase regions, and for fluid interfaces.

2-7. Numerical Solution Procedures for a Single-Phase Region -- Equations in the single phase region are

$$\frac{\partial U}{\partial z} = -\frac{q^{\prime \prime \prime \prime}}{\rho^2 \frac{\partial H}{\partial \rho}}$$
(2-30)

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial z} \left( \rho U \right) = 0 \tag{2-31}$$

$$\frac{\partial}{\partial t} (\rho H) + \frac{\partial}{\partial z} (\rho U H) = q'''$$
 (2.32)

Defining  $\gamma = \rho H$ , the numerical solution is obtained from the following difference equations.

$$\frac{u_{1}^{n+1} - u_{1-1}^{n+1}}{\Delta z_{1-1}} = -\left[\frac{q_{1}}{Q_{K}}\right]^{n}$$
(2-33)

$$\frac{\rho_1^{n+1} - \rho_1^n}{\Delta t} + \frac{\rho_1^{n+1} U_1^{n+1} - \rho_{1-1}^{n+1} U_{1-1}^{n+1}}{\Delta z_{1-1}} = 0$$
 (2-34)

$$\frac{\gamma_{1}^{n+1} - \gamma_{1}^{n}}{\Delta t} + \frac{\gamma_{1}^{n+1} U_{1}^{n+1} - \gamma_{1-1}^{n+1} U_{1-1}^{n+1}}{\Delta z_{1-1}} = q^{\prime \prime \prime , n}$$
(2-35)

Except for the terms in q''', the above equations are all implicit.

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Superscript n refers to the old time level and n+l refers to the new time level. Similarly, subscripts i-l and i denote axial locations separated by  $\Delta z_{1-1}$ . The conditions at all spacial locations for time level n and up to location i-l for time level n+l are known.  $Q_{\rm K}$  in equation (2-33) (a property relation) is defined as

$$d^{K} = b_{5} \frac{9b}{9H}$$

Equation (2-33) can thus be solved for  $U_1^{n+1}$ , followed by equations (2-34) and (2-35) to obtain  $\rho_1^{n+1}$  and  $\gamma_1^{n+1}$ . Enthalpy H is then given by H =  $\gamma/\rho$ . Once H is known, all the other properties are determined from the property tables as functions of H and system pressure.

The value of  $q'''/Q_{K}$  used in equation (2-33) is the value at node 1-1, except in the presence of single-phase vapor flow, in which case the average between 1 and 1-1 is taken.

2-8. <u>Numerical Solution Procedures for a Two-Phase Region</u> -- Equations to be solved in the two-phase region are given in paragraph 2-2. The vapor enthalpy equation, equation (2-17), is integrated first to determine the vapor temperature at all locations.

It is shown in paragraph 2-11 that the total volumetric heat flow from solid surfaces to the fluid, q''', and the heat flow directly responsible for vapor generation,  $q'''_{boil}$ , can both be split into components which are and are not functions of the vapor temperature,  $T_{o}$ , as shown below:

$$q''' = (hA)_{wQ} (T_w - T_{sat}) + (hA)_{wv} (T_w - T_v)$$
  
 $q'''_{boil} = (hA)_{wQ} (T_w - T_{sat}) + (hA)_{Qv} (T_v - T_{sat})$ 

where  $(hA)_{WQ}$ ,  $(hA)_{WV}$ ,  $(hA)_{QV}$  represent the overall heat transfer coefficient from the wall to the liquid, from the wall to the vapor, and from the vapor to the liquid, respectively, multiplied by appropriate heat transfer areas per unit volume.

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The relation  $dH = C_p dI$ , used with equation (2-17) gives

$$\alpha C_{p} \rho_{v} \left[ \frac{\partial T_{v}}{\partial t} + U_{v} \frac{\partial T_{v}}{\partial z} \right] = (hA)_{wQ} (T_{w} - T_{sat}) + (hA)_{wv} (T_{w} - T_{v}) - [(hA)_{wQ} (T_{w} - T_{sat}) + (hA)_{Qv} (T_{v} - T_{sat})] \left[ 1 + \frac{1}{H_{Qv}} \int_{T_{sat}}^{T_{v}} C_{p} dT \right]$$

$$(2-36)$$

The following approximations are made:

$$\int_{T_{sat}}^{T_{v}} c_{p} dT = \overline{c}_{p} (T_{v} - T_{sat})$$

$$P_{v} = \frac{p}{RT_{v}}$$

where

$$\overline{C}_{p} = \frac{C_{p} (T_{sat}) + C_{p} (T_{v})}{2}$$

$$p = pressure$$

$$R = gas constant$$

With these approximations, equation (2-36) can be put in finite difference form as follows:

(2-37)

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Equation (2-37) is a polynominal which can be solved for  $T_v$ , by iteration.

After a new value for  $T_v$  is obtained, other properties such as enthalpy and density are updated using property routines.

Four flow regimes are considered: bubbly (or churn turbulent), film, transition, and droplet. The bubbly, film, and transition regimes are treated using the drift flux model; in the droplet regime the relative velocity is obtained by calculating the acceleration of the droplets in the vapor. These two approaches lead to different methods for solving the continuity equations. The drift flux model is examined first. The finite difference form of equation (2-15) is:

$$\frac{j_{1}^{n+1} - j_{1-1}^{n+1}}{\Delta z_{1-1}} = -\left[\frac{q'''_{boll}^{n+1}}{Q_{K}}\right] - \alpha_{1-1}^{n+1}\left[\frac{n \rho_{V_{1}}^{n+1} - n \rho_{V_{1}}^{n}}{\Delta t}\right] - (2-38)$$

$$- j_{V_{1-1}}^{n+1}\left[\frac{n \rho_{V_{1}}^{n+1} - n \rho_{V_{1-1}}^{n+1}}{\Delta t}\right] - (2-38)$$

The finite difference form of equation (2-16) can be put in terms of  $\alpha$  by using the definitions for  $\rho_s$  and  $\rho_1$ :

(2-39)

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2-14

In deriving equation 2-39, the assumption has been made that the void fraction in the control volume propagates (by convection) in the same direction as j. However, the void propagation direction is determined by the local value of the rate of change of j with respect to  $\alpha$ . Under some circumstances the void fraction that determines mass velocity at the downstream end of control volume 1 is more appropriately defined by the void fraction in the downstream control volume, i+1. Special calculations must be performed to determine when this condition occurs. If this condition is detected, the method of solving the continuity equation at this location must be altered.

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In finite difference form, the void progagation velocity at the intersection between control volumes i and i+l is

$$prop = \frac{J_{v_1}^{n} - J_{v_{1+1}}^{n}}{\alpha_1^{n} - \alpha_{1+1}^{n}}$$
(2-40)

where

V

$$J_{v_{1}}^{n} = \alpha_{1}^{n} J_{1}^{n+1} + \alpha_{1}^{n} (1 - \alpha_{1}^{n}) U_{r_{1}}^{n}$$
$$J_{v_{1+1}}^{n} = \alpha_{1+1}^{n} J_{1}^{n+1} + \alpha_{1+1}^{n} (1 - \alpha_{1+1}^{n}) U_{r_{1+1}}^{n}$$

If V is negative, the vapor volumetric flux is decreasing in the direction of increasing vold fraction. Because this implies that vapor is accumulating at the intersection between volumes i and i+1, the higher void fraction  $(\alpha_{i+1})$  should be chosen at the downstream end of control volume i. Thus, equation (2-39) takes a new form.

(2-41)

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Because the right-hand side of equation (2-41) consists entirely of old time values of  $\alpha$ , care must be taken in choosing time step size under conditions where equation (2-41) is used. These limitations are discussed in paragraph 2-10.

In the droplet flow regime, the velocity of the droplet under the competing effects of vapor-droplet drag and gravity is first calculated. Void fraction, vapor velocity, and volumetric flux are then determined from the continuity equations.

It is shown in paragraph 2-13 that the droplet acceleration is given by

$$\frac{dU_{Q}}{dt} = \frac{3}{4} \frac{C_{dd} \rho_{v} (U_{v} - U_{Q})^{2}}{\rho_{Q} D_{d}} - g \qquad (2-42)$$

where  $C_{dd}$  = droplet drag coefficient and  $D_d$  = droplet diameter related to the total vapor droplet heat flux  $q_{gv}$  by

$$\frac{\partial D_{d}}{\partial t} + U_{g} \frac{\partial D_{d}}{\partial z} = -\frac{2 q_{g,v}^{*}}{\rho_{g} H_{g,v}}$$
(2.43)

Equation (2-42) is integrated first to obtain  $U_{g}$ , and  $D_{d}$  is calculated from equation (2-43). The liquid phase continuity equation, equation (2-16a), is then used to determine the void fraction.

Equation (2-16a) is written in finite difference form and rearranged to give the liquid fraction at node i at the new time step.

Next, to obtain the vapor velocity the overall continuity equation, equation (2-16), is rearranged to give

(2 45)

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(2 44)

Finally, the volumetric flux  $j_1^{n+1}$  is calculated.

$$j_{1}^{n+1} = U_{v_{1}}^{n+1} \alpha_{1}^{n+1} + U_{\varrho_{1}}^{n+1} (1 - \alpha_{1}^{n+1})$$
(2.46)

## 2-9. Numerical Solution Procedure for Fluid Interfaces

Equations for the movement of the different fluid interfaces are given in paragraph 2-2. To find the fluid flow conditions in the vicinity of any of these interfaces, its location is obtained first explicitly. The solution

procedure for obtaining the fluid conditions is similar to the one described earlier, except that a moving boundary control volume is used for the mass conservation equation.

The subcooled liquid/steam interface is approached in the same manner as the single-phase region.  $U_{\rm F}^{n+1}$  is obtained first from equation (2-21) after locating the position of the interface. To calculate the density and enthalpy at the interface, a moving boundary control volume is considered as shown in figure 2-1 (control volume A).

Conservation of mass gives

$$\frac{d}{dt} \left[ \rho_{F} - (z_{F} - z_{1-1}) \right] = \rho_{1-1} U_{1-1}$$

01

$$z_F - z_{1-1}$$
  $\frac{d\rho_F}{dt} + \rho_F - \frac{dz_F}{dt} = \rho_{1-1} U_{1-1}$ 

Using implicit form,

$$\frac{\rho_{\rm F}^{n+1} - \rho_{\rm F}^{n}}{\Delta t} (z_{\rm F}^{n+1} - z_{1-1}) + \rho_{\rm F}^{n+1} U_{\rm F}^{n+1} = \rho_{1-1}^{n+1} U_{1-1}^{n+1}$$
(2-47)

Similarly for enthalpy.

$$\frac{(\rho_{\rm F} - H_{\rm F} -)^{n+1} - (\rho_{\rm F} - H_{\rm F} -)^{n}}{\Delta t} = (z_{\rm F}^{n+1} - z_{1-1}) + (\rho_{\rm F} - H_{\rm F} -)^{n+1} U_{\rm F}^{n+1} = (\rho_{1-1} - U_{1-1} + H_{1-1})^{n+1} + q^{n+1} + q^{n+1} - z_{1-1}) + (\rho_{\rm F} - H_{\rm F} -)^{n+1} U_{\rm F}^{n+1} = (2.48)$$

Equations (2-47) and (2-48) give  $\rho_{\rm F}^{n+1}$  and  ${\rm H}_{\rm F}^{n+1}$ , respectively. Subscript F refers to the interface; F<sup>-</sup> is the upstream side of the interface. Similarly, mass and energy balances are taken on control volume B in figure 2-1 to obtain fluid conditions at node 1.

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Figure 2-1. Subcooled Liquid/Steam Interface





Figure 2-3. Two-Phase/Steam Interface

Along the subcooled liquid/two-phase interface (saturation line).



а



The volumetric flux across the two-phase/steam interface is continuous.

$$J_{z_{F}} = J_{z_{F}} = U_{v,zF}$$

 $j_{z_F}^{n+1}$  is obtained from the known value of  $j_{1-1}^{n+1}$ , assuming no additional superheating in the region between 1-1 and  $z_F$  (figure 2-3).

Conservation of mass in a control volume with  $z_F$  as the upper boundary (moving) and either node 1-1 or saturation line as the lower boundary gives

(2-51)

a

 $U_{\rm F}$  is the velocity of the interface, and subscript  ${\rm F}^-$  refers to the upstream side of the interface.

$$\rho_{F} = \alpha_{F} \rho_{V, z_{F}^{-}} + (1 - \alpha_{F}) \rho_{Q}$$

$$(1 - \alpha_{F})U_{F} = (1 - \alpha_{F})U_{Q, z_{F}^{-}} = J_{z_{F}^{-}} - \alpha_{F}U_{V, z_{F}^{-}}$$

and

$$U_{v,z_{F}} = J_{z_{F}} + (1 - \alpha_{F})U_{r,z_{F}}$$

As in other calculations, equation (2-51) is solved to obtain  $\alpha_{g}^{n+1}$  by

] a,c

Calculation of other fluid conditions at this interface follows the same procedure as in the two-phase region. The simplified equation for  $\alpha_F^{n+1}$  is given below.  $\neg a.c$ 

(2-52)

## 2-10. Stability and Time Step Selection

The implicit finite differencing scheme used in most of the BART equations frees them from time step limitations imposed by stability requirements. In two areas, however, the explicit nature of the equations being used means that time step size must be carefully controlled.

If the void propagation velocity  $V_{prop}$  is negative at any location, equation (2-14) must be used. When this occurs the time step size is limited as shown below.

$$\Delta t \leq \frac{\Delta z_{1-1}}{J_1}$$

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2-25

(2 - 53)

The smallest time step at all locations where V prop is negative is used.

In single-phase flow, the q''' term in equation (2-35) contains old time values of the fluid temperature. A linear analysis of this equation shows that a time step limitation arises of the form

$$\Delta t \leq \frac{\rho C \rho}{4h} \frac{0}{(2-54)}$$

where h = the local heat transfer coefficient.

If neither of the limitations listed above applies at a particular time, the time step size is set to an input value (normally 0.1 second).

## 2-11. VAPOR-LIQUID AND FLUID-ROD SURFACE INTERACTIONS

To implement the equations derived in paragraph 2-1, auxiliary models must be developed that describe the way the liquid and vapor phases interact physically and thermally with one another, and the way the two-phase mixture interacts physically and thermally with the rod surfaces.

The vapor-liquid interaction is sufficiently described, within the scope of the equations of paragraph 2-1, if the average relative velocity and the heat transfer rate between the phases is known. In addition, the following fluid-to-rod surface heat transfer components must be specified:

o Heat transfer from rod to liquid through the following:

- -- rorced convection and nucleate boiling
- -- Vapor film conduction or convection (film boiling)
- -- Radiation
- -- Direct contact

o Heat transfer from rod to vapor through forced convection and radiation

## 2-12. Vapor-Liquid Interactions

2-13. <u>Flow Regime Models</u> -- Three relatively well-defined flow regimes exist in a nuclear core or heater rod bundle during reflood. These flow regimes are encountered below, near, and above the guench front.

Vapor is formed below the quench front primarily as a result of power generated in the rod. Bubbles generated at the rod surface coalesce to form larger vapor regions, which possibly span several subchannels as the void fraction increases (figure 2-4). A void fraction correlation intended specifically for this flow regime was developed by Yeh.<sup>(4)</sup> Its general form is

$$\alpha = 0.925 \left(\frac{\rho_{\rm v}}{\rho_{\rm g}}\right)^{0.239} \left(\frac{j_{\rm v}}{u_{\rm rb}}\right)^{\rm m} \left(\frac{j_{\rm v}}{j}\right)^{0.6}$$
(2-55)

where

j<sub>v</sub> = vapor volumetric flux j = total volumetric flux

$$U_{rb} = 1.53 \left[ \frac{\sigma \Delta \rho \cdot q}{\rho_{q}^{2}} \right]^{-0.25}$$
(2-56)
$$m = 0.67 \text{ if } J_{v} / U_{rb} < 1$$
(2-57)
$$m = 0.47 \text{ if } J_{v} / U_{rb} \ge 1$$

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Figure 2-4. Flow Regimes Below Quench Front in Rod Bundle

At all void fractions except near zero, the ratio  $j_v/j$  is nearly equal to 1 if the mass velocity is low. Equation (2-55) can be simplified and rearranged to give

$$J_v = F_1 \alpha^n$$
 (2-58)

$$F_{1} = U_{rb} \left[ \frac{1}{0.925} \left( \frac{\rho_{g}}{\rho_{v}} \right)^{0.239} \right]^{n}$$
(2-59)

and

n = 1.49 if 
$$j_v/U_{rb} < 1$$
 (2-60)  
n = 2.13 if  $j_v/U_{rb} > 1$ 

The relative velocity,  $U_r$ , is defined by

$$U_{r} = U_{v} - U_{0}$$
 (2.61)

The relationship expressed in equation can be stated approximately in terms of  $U_{\rm p}$  (again assuming that  $j_{\rm p}/j$  - 1) by

$$U_{r} = \frac{J_{v}}{\alpha}$$
$$= F_{1} \alpha^{n-1}$$
(2-62)

Therefore in the drift flux formulation

$$J_{v} = \alpha J + \alpha (1-\alpha) U_{r}$$

$$= \alpha J + F_{1} \alpha^{n} (1-\alpha)$$
(2-63)

for the flow regime below the quench front.

Near (just above) the quench front, a vapor film will form on the rods, producing the flow regime shown in figure 2-5. The vapor-liquid interface is unstable, and a characteristic relative velocity can be defined based on stability considerations (paragraph 3-12). It is assumed that the film is

At void fractions above  $\alpha_f$ , vapor will break away from the film and enter the bulk fluid. At this point a transition flow regime is formed (paragraph 2-14).

In the vapor film flow regime, the relative velocity,  $U_{ra}$ , given by equation (3-56) in paragraph 3-9 is assumed to represent the average relative velocity between the phases,  $U_{rf}$ . This regime is assumed to exist in the quench front region and anywhere above the quench front where  $\alpha < \alpha_{f}$ . An exception to this rule occurs if the fluid entering the quench front contains vapor. In this case, the flow regime becomes a transitional flow regime (paragraph 2-14).

Above the quench front, where the liquid fraction is low, the flow regime consists of superheated vapor and dispersed liquid drops (figure 2-6). If the droplets are assumed to be spherical and do not significantly interact with one another or with solid surfaces, a force balance yields droplet acceleration = (drag - droplet weight)/droplet mass or

$$\frac{du_{\varrho}}{dt} = \frac{3}{4} C_{dd} \frac{\rho_{v} (U_{v} - U_{\varrho})^{2}}{\rho_{\varrho} D_{d}} - g$$

(2-64)


14.380-6





where

U<sub>2</sub> = droplet velocity C<sub>dd</sub> = droplet drag coefficient D<sub>d</sub> = droplet diameter g = gravitational acceleration

The terminal velocity of the oroplets relative to the vapor velocity is

$$U_{rd} = \left(\frac{4 \Delta p g D_d}{3 \rho_v C_{dd}}\right)^{0.5}$$
(2-65)

The drag coefficient,  $C_{dd}$ , is calculated using a correlation for solid spheres. (5)

$$C_{dd} = \frac{24}{Re_d} \left[ 1 + 0.15 Re_d^{0.687} \right] + \frac{0.42}{1. + 42500 Re_d^{-1.16}}$$
 (2-66)

where

$$Re_{d} = \frac{\rho_{v}(U_{v} - U_{g}) D_{d}}{\mu_{v}}$$
(2-67)

Equations (2-64) and (2-65) also require that a droplet diameter be specified. A droplet of initial size  $D_{d_0}$  will experience the following forces leading to reduced droplet size as it travels up the bundle:

- o Evaporation
- Contact with rod surfaces
- o Hydrodynamic forces

Because the heat transfer from the droplet is known, the rate of change of droplet mass due to this heat transfer can be determined, leading to the following expression for the droplet diameter  $D_{d}$ .

$$\frac{\partial D_{d}}{\partial t} + U_{g} \frac{\partial D_{d}}{\partial z} = -\frac{2 q'' gv}{\rho_{g} H_{gv}}$$
(2-68)

where

q''<sub>&v</sub> = vapor-droplet heat flux U<sub>0</sub> = droplet velocity

The effect of droplet impingement on hot surfaces on droplet breakup is not well understood. However, the importance of this mechanism in determining the droplet physical behavior is closely related to the relative magnitude of the fluid-rod heat transfer component due to drop contract. [

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The effect on the droplet of momentum and shearing forces due to relative motion can be characterized by the Weber number.

$$We = \frac{\rho_v \left(U_v - U_{\varrho}\right)^2 D_d}{\sigma}$$
(2-69)

The maximum stable drop size at any point in the bundle can be found from equation (2-69) if We is specified. If the actual droplet diameter,  $D_d$ , exceeds the value calculated from equation (2-69), the droplet is assumed to break into two smaller droplets of equal volume. In terms of the droplet diameter,

$$D_{d}$$
 (NEW) =  $\left(\frac{1}{2}\right)^{0.333} D_{d}$  (OLD) (2-70)

The tendency for droplets to break and form smaller droplets appears to be well characterized by the Weber number criterion [equation (2-69)]. However the mechanisms which form droplets from large volumes of liquid near the quench front are less well understood. Liquid impingement on solid surfaces and shearing and momentum forces are probably all important breakup mechanisms. The choice of initial droplet size will therefore have to rely primarily on data. The basis for choosing the initial size is described in paragraph 2-19.

2-14. <u>Transition flow Regime</u> -- An additional flow regime, less well defined than those discussed in preceding paragraphs, is encountered during reflood. It forms a transition from the bubbly and vapor film regimes near the quench to the droplet flow regime. In this region, the flow is highly chaotic and probably contains elements of the three flow regimes described previously as well as other, more complex characteristics.

The transition flow regime is assumed to progress through several stages as void fraction increases, as shown in figure 2-7. In the first stage, the transition flow regime is assumed to be a vapor film regime. If boiling is occurring below the quench front, the voids thus formed are assumed to exist in the bulk of the fluid [figure 2-7(a)]. If there are no voids in the channel and the void fraction is less than a specified value  $\alpha_{f}$ , the relative velocity is given by equation (3-56) in paragraph 3-9.

$$U_{r} = \left(\frac{\Delta \rho \ g \ D_{h}}{\rho_{v}}\right)^{1/2}$$
(2-71)

If voids already exist in the channel, the additional vapor produced at the rod surface is assumed to collect as a vapor film, and the void fraction in the channel is assumed to remain at its value below the quench front until the film void fraction exceeds  $\alpha_e$ . [

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2-33

a

where

 $\alpha_a$  = area of vapor film/channel area  $\alpha_c$  = void fraction in channel

a

If  $\alpha_r$  is the void fraction below the quench front, then

 $\alpha_a$  grows with  $\alpha$  until  $\alpha_a > \alpha_f$ , at which point the flow regime enters a new stage.

In the first stage described above, in which the vapor generated remains in the film and the bulk liquid may contain voids generated below the quench front, the mixture may be idealized as a continuous liquid phase moving at an average velocity,  $U_{g}$ , and a discontinuous vapor phase moving at two velocities, the film velocity,  $U_{vf}$ , and the channel velocity,  $U_{vc}$  [figure 2-7(a)]. Relative to the local relative velocity, the local liquid velocity is

$$U_{2}^{1} = j^{1} - \alpha^{1} U_{r}^{1}$$
 (2-74)

where the 1s denote local values. The average across the rod bundle yields

$$U_{n} = j - \alpha U_{r}$$
(2-75)

where

$$\alpha U_{r} = \frac{1}{A_{F}} \int_{A_{F}} \alpha U_{r} \, dA \qquad (2-76)$$

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2-34

(2-72)

(2 - 73)

Because two relative velocities have been assumed, aU\_ becomes

$$\overline{\alpha U}_{r} = \alpha_{a} U_{rf} + \alpha_{c} (1 - \alpha_{a}) U_{rc}$$
(2-77)

where  $U_{rf}$  is given by equation (3-56) in paragraph 3-9, and  $U_{rc}$  is given by equation (2-62).

The overall volumetric flux of liquid and vapor can thus be described by the following equations:

$$j_{a} = (1 - \alpha)U_{a} = (1 - \alpha)j_{a} - (1 - \alpha)\alpha V_{cm}$$
(2-78)

$$\mathbf{j}_{\mu} = \alpha \mathbf{U}_{\mu} = \alpha \mathbf{j} + (1 - \alpha) \alpha \mathbf{V}_{rm}$$
(2-79)

where

$$V_{rm} = \frac{\overline{\alpha U}_r}{\alpha}$$
 for  $\alpha < \alpha_f + \alpha_c (1 - \alpha_f)$ 

As the film void fraction begins to exceed  $\alpha_{\rm f}$ , the liquid in the bundle is assumed to break into large slugs [figure 2-7(b)]. Because a major portion of the interface between these slugs and the vapor is still contained in the film surrounding the rods, it is reasonable to assume that the velocity of the slugs relative to the vapor is characterized by film flow and can, therefore, be described by equation (3-56). In the vapor region between the liquid slugs, a droplet regime is assumed to form with void fraction  $\alpha_{\rm d}$ , droplet diameter  $D_{\rm d}$ , and relative velocity  $U_{\rm rd}$  [equation (2-65)]. Let  $\alpha_{\rm b}$  = the

fraction of total vapor volume which resides in the regions between the liquid slugs. Then, because

(2-80)

a

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it follows that

 $\alpha_b$  increases proportionally with  $\alpha$ . Thus, the amount of vapor in which droplets are dispersed is assumed to increase linearly with  $\alpha$ .

The overall fluxes in this stage may be calculated in a manner analogous to that used for the first stage, if it is assumed that the vapor phase is now continuous and is traveling at a uniform velocity,  $U_v$ . Then a bundle average gives

$$U_{v} = j + (1 - \alpha) U_{r}$$
 (2-82)

where

$$(1 \ \alpha) U_{r} = \alpha_{b} (1 \ \alpha_{d}) U_{rd} + (1 \ \alpha_{b}) (1 \ \alpha_{f}) U_{rf}$$
(2.83)

for  $\alpha > \alpha_f + a_c (1 - \alpha_f)$ 

The overall fluxes can then be calculated from equations (2-78) and (2-79) using the new definition of  $V_{rm}$ :

$$V_{\rm rm} = \frac{\overline{(1 - \alpha) U_{\rm r}}}{(1 - \alpha)}$$

Two additional parameters have been introduced to define the transition flow regime,  $\alpha_{\rm f}$  and  $\alpha_{\rm d}$ . A value for  $\alpha_{\rm f}$  is obtained in paragraph 3-9 from quench data [equation (3-58)]. The value of  $\alpha_{\rm d}$  must be obtained by comparison with data. The methods used to obtain  $\alpha_{\rm d}$  and other empirical parameters are described in paragraph 2-19.

2-15. Summary of Flow Regime Models — The models described in the preceding paragraphs define a flow regime map for the relative velocity  $U_r$  as a function of void fraction and rod temperature. Figure 2-8 defines the flow regime boundaries. These regimes are felt to represent the key features of the reflood transient adequately.

The local void fraction, the location of the quench front, and the void fraction at the quench front determine the flow regime. For example, figure 2-9 illustrates the changes in flow regime which occur near the quench front. Below the quench front,  $U_r$  is determined from equation (2-62). Just above the quench front the flow regime is the combined churn turbulent/film regime described by equation (2-77). Above this regime the flow becomes a combined slug-droplet regime described by equation (2-83). Finally, when the void fraction exceeds  $\alpha_d$ , the flow becomes fully dispersed. At this point droplet acceleration begins and the equations developed in paragraph 2-13 are used.

#### 2-16. Vapor-Liquid Heat Transfer

The models developed in paragraph 2-1 allow a temperature difference to exist between liquid and vapor, where the liquid is assumed to be saturated. Heat will therefore flow from vapor to liquid as the vapor superheats in the bundle. The models which will be used to describe these heat transfer mechanisms are developed below.

Four distinct vapor-liquid heat transfer mechanisms can be identified: these mechanisms correspond to the flow regimes described in the preceding paragraphs.

All heat flowing from the rod below the quench front is assumed to heat the liquid or to generate vapor. The mixture is therefore in thermal equilibrium, and no temperature difference exists between the phases.



Figure 2-8. Map of BART Flow Regimes



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Figure 2-9. Typical Flow Regimes in BART Channel

In the vapor film regime (at the quench front), heat transfer occurs by conduction or convection through the vapor film from the rod surface to the liquid. Because the vapor volume is small, no serious error results from the assumption that all heat flowing from the rod is used to heat the liquid or generate additional saturated vapor. Thus in this regime the phases are also assumed to be in thermal equilibrium.

Liquid droplets in the dispersed regime  $(\alpha \ge \alpha_d)$  will evaporate as they travel up the bundle. Convective heat transfer between a droplet and superheated vapor is<sup>(6)</sup>

$$NU_d = 2.0 + 0.55 Re_d^{0.5} Pr_v^{0.33}$$

where

$$NU_{d} = \frac{h_{\ell} v_{dc} D_{d}}{k_{v}}$$
$$Re_{d} = \frac{\rho_{v} (U_{v} - U_{\ell}) D_{d}}{\mu_{v}}$$

 $h_{lvdc}$  is the heat transfer coefficient between liquid and vapor (droplet regime) due to convection.

Because of the high vapor temperatures expected during reflood, the heat transfer to the droplets will be limited by the high mass transfer from the droplet surface. To account for this, the droplet Nusselt number is modified by the mass transfer parameter as follows:

$$\overline{Nu}_{d} = \frac{Nu_{d}}{1 + 0.5 \text{ Cp} (T_{v} - T_{sat})}$$

$$\frac{H_{2v}}{H_{2v}}$$
(2-84)

In addition to heat transfer due to conduction and convection, radiation occurs between the superheated vapor and the liquid. Because vapor-droplet radiation is coupled to rod-vapor and rod-liquid radiation, radiation heat transfer is treated generally in paragraph 2-18.

If  $h_{2vdr}$  = the vapor-liquid radiation heat transfer coefficient, then the overall vapor-liquid heat transfer coefficient in the droplet regime is

$$h_{\ell vd} = h_{\ell vdc} + h_{\ell vdr}$$
(2-85)

The surface area of droplets per unit channel volume is

$$A_{HTd} = \frac{6(1-\alpha)}{D_d}$$
(2-86)

The heat flow rate per unit channel volume from vapor to liquid is

$$q'''_{QV} = h_{QVd} (T_v - T_{sat}) \frac{6(1 - \alpha)}{D_d}$$
 (2-87)

In the transition flow regime, heat transfer is assumed to occur from the vapor to the droplets contained in the large vapor regions shown in figures 2-7(b) and 2-7(c). The additional heat transfer area available from the liquid slugs is assumed to be negligible. The heat transfer area between liquid and vapor in the transition regime is then

$$A_{\rm HT} = \frac{6 (1 - \alpha_{\rm d}) \alpha_{\rm b}}{D_{\rm d}}$$
(2-88)

Equation (2-84) is used to calculate the heat transfer coefficient.

# 2-17. Fluid-Rod Surface Interactions

Heat transfer from rod surfaces to the fluid can be best described relative to the location of the quench front.

Heat transfer below the quench front is due to forced convection if the wall temperature is below the saturation temperature and is due to a combination of nucleate boiling and forced convection if the wall temperature is above the saturation temperature.

In the forced convection case, if the Reynolds number of the liquid is less than 2000, the flow is assumed to be laminar. Kays<sup>(7)</sup> presents Nusselt numbers for laminar flow past tube banks which are a function of the tube diameter and pitch. For fuel rod arrays, the following Nusselt number is used:

(2-89)

$$Nu_0 = 10$$
 when  $Re_0 < 2000$ 

At low Reynolds numbers, hydrodynamic and thermal entry lengths may be significant. This effect is estimated in rod bundles by assuming that circular tube solutions apply.<sup>(7)</sup> The Nusselt number as a function of nondimensional distance from the bottom of the bundle is shown in figure 2-10.

At Reynolds numbers greater than 2500, turbulent flow is assumed to exist. Heat transfer is calculated using the Dittus-Boelter equation:

$$Nu_{g} = 0.023 \text{ Re}_{g}^{0.8} \text{ Pr}_{g}^{0.4}$$
 (2-90)

Entry lengths are small, in general, in turbulent flow and are therefore ignored.

Between Reynolds numbers of 2000 and 2500, a linear interpolation is used between equations (2-89) and (2-90).



If the wall temperature is above saturation temperature, heat transfer is assumed to be a combination of forced convection and nucleate boiling.

$$q^{*} = h_{wlc} (T_w - T_l) + h_{wlnb} (T_w - T_{sat})$$

where  $h_{wlc}$  is calculated using equations (2-89) and (2-90), and  $h_{wlnb}$ , the nucleate boiling heat transfer coefficient, is given by a correlation in Rohsenow. (8)

$$h_{wlnb} = \left[\frac{C_{pl}(T_w - T_{sal})}{0.013 H_{lv}}\right]^3 \frac{\mu_l H_{lv}}{P_r^{5.1}}$$
(2-91)

where

$$L = \left[\frac{\sigma}{\Delta \rho g}\right]^{0.5}$$

Heat transfer at the quench front is closely coupled with axial conduction in the rod. These phenomena are discussed in detail in section 3.

Heat transfer above the quench front is examined according to vapor film flow regime, a droplet flow regime, and a transition flow regime.

Near the quench front, a heat transfer regime similar to pool film boiling exists. Heat transfer due to conduction and radiation across the vapor film is considered. Heat transfer due to conduction is given by Berenson's model for film boiling.<sup>(9)</sup>

$$h_{wlfc} = 0.62 \left[ \frac{k_v^3 H_{lv} \rho_v \Delta \rho_g}{\mu_v (T_w - T_{sal}) L} \right]^{0.25}$$

where

$$L = \left[\frac{\sigma}{\Delta \rho g}\right]^{0.5}$$

2-44

(2-92)

Bereason assumed that conduction heat transfer occurred at points on the vapor film where the unstable interface came close to the rod surface. The ratio of the heat transfer area to the total area of the vapor film was determined to be 0.7. Radiation can be accounted for by assuming that the rod surface and liquid interface form two parallel planes. Then the radiation heat flux from the rod surfaces is

$$q'' w_{fr} = \frac{S (T_w^4 - T_{sat}^4)}{\frac{1}{e_w} + \frac{1}{e_0} - 1}$$
(2-93)

where

S = Stefan-Boltzmann constant
e = rod surface emissivity
e = liquid emissivity

It is assumed that radiation heat transfer takes place in the same area fraction as conduction; therefore, the heat flux  $q''_{wlfr}$  averaged over the rod surface is given by equation (2-93) multiplied by the ratio 0.7. The total heat transfer from the rod surface to the liquid in the vapor film regime is

 $q''_{wf} = h_{wlfc} (T_w - T_{sat}) + q''_{wlfr}$ 

There are four possible mechanisms for heat transfer from the rod surface to the fluid in dispersed flow (droplet flow regime), as follows:

- o Droplet contact
- o Forced convection to vapor
- o Radiation to fluid
- o Radiation to other rods

A careful analysis was made of the forced convection and radiation components of the heat transfer in the FLECHT Cosine and Skewed tests.<sup>(10)</sup> From that analysis, the following conclusions were drawn.

Radiation accounted for roughly 20 to 30 percent of the total heat flux from the heat rods. The three components of radiation were found to be the same order of magnitude.

The remaining heat flux, attributed to forced convection from the rod surface to the vapor, was significantly higher than could be accounted for by standard correlations such as Dittus-Boelter.

Drop contact heat transfer, ignored in the FLECHT analysis, could account for the excess heat transfer. However, an additional effect of drop contact heat transfer is to generate saturated vapor, reducing or slowing the rate of temperature increase of the vapor. The fact that steam probes have measured extremely high vapor temperatures within the FLECHT bundle casts doubt on drop contact heat transfer as an explanation for the excess heat flux found in the FLECHT tests.

It was concluded in the FLECHT analysis that the physical mechanism providing the most consistent explanation for the excess heat transfer was the level of turbulence within the vapor. Given the same local Reynolds number, turbulent diffusion of heat was more effective during two-phase flow in the FLECHT bundle than would be the case in single-phase flow in a tube.

The intensity of the turbulence in a single-phase fluid flowing in a channel can usually be scaled by the local shear stresses at the wall. <sup>(11)</sup> Carrying this relationship a step further, one can assume that any mechanism which creates shearing stresses in the fluid wall will enhance the turbulent diffusion processes in the fluid. One such mechanism in two-phase dispersed flow is the drag imposed by the droplets on the vapor. An analysis of typical

conditions within a FLECHT bundle shows that the local pressure gradient produced by droplet drag

$$\frac{dp}{dx}\Big|_{1} = \frac{3 C_{dd} P_{v} U_{r}^{2} (1 - \alpha)}{4 D_{d}}$$
(2-94)

is of the same order of magnitude as the wall shear stress induced pressure gradient

$$\frac{dp}{dx}\Big|_{2} = \frac{f \rho_{v} U_{v}^{2}}{2 D_{h}}$$
(2-95)

where f = local single-phase wall friction factor.

Several semianalytical models have been developed which predict the local heat transfer coefficient in turbulent flow. One of these is written below.<sup>(7)</sup>

$$Nu_{wvdc} = \frac{Re_{v} Pr_{v}\sqrt{f/8}}{0.833 [5 Pr_{v} + 5 \ln (5 Pr + 1) + 2.5 \ln (\frac{Re_{v\sqrt{f/8}}}{60})]}$$
(2-96)

where

$$Re_{v} = \frac{\rho_{v} U_{v} D_{h}}{\mu_{v}}$$
$$Nu_{wvdc} = \frac{h_{wvdc} D_{h}}{k_{v}}$$

Equation (2-96) was derived by constructing a turbulent velocity profile using Prandtl's relation for the eddy diffusivity. The energy equation was then integrated across the channel to obtain the heat transfer rate. The single-phase friction factor f was used to relate the shear stress (which through the eddy diffusivity is proportional to the velocity gradient) to the average fluid velocity. The effect of droplet drag on the local heat transfer can then be accounted for by defining a modified friction factor.

a,c

(2.97)

The stresses caused by the wall are different from those caused by the droplets. Wall-produced shear stress is largest near the wall, whereas the stresses caused by droplet drag occur in the bulk of the fluid. In addition, droplet drag consists of a form loss component as well as a shear component. However, models such as equation (2-96), which use simple relationships for the turbulent diffusivity, agree with data, indicating that the equally simple treatment of the wall and droplet stresses may be adequate to allow calculation of the heat transfer coefficient.

The radiation heat transfer from the wall to the liquid,  $q_{wkdr}^{\prime\prime}$ , and the radiation heat transfer from wall to vapor,  $q_{wvdr}^{\prime}$ , are described in paragraph 2-18. The overall heat transfer from the wall to the fluid in the droplet regime is given by

$$q''_{wd} = h_{wvdc} (T_w - T_v) + q''_{wvdr} + q''_{wvdr}$$
 (2-98)

Heat transfer from wall to the fluid in the transition flow regime is assumed to consist of the vapor film component and the dispersed flow component, weighted according to void fraction shown in figure 2-7. The total heat flux from the wall in the transition regime is given by

$$q''_{wt} = (1 - \alpha_b) q''_{wf} - \alpha_b q''_{wd}$$
 (2-99)

where  $q''_{wf}$  is defined by equation (2-93) and  $q''_{wd}$  is defined by equation (2-98).

# 2-18. RADIATION HEAT TRANSFER IN DISPERSED FLOW

It has been shown that radiation represents a significant fraction of the total heat flux in the FLECHT experiments.<sup>(10)</sup> Four types of radiation are possible:

- o Rod to vapor
- o Rod to liquid
- o Vapor to liquid
- o Rod to rod

The first three are calculated in BART; rod-to-rod radiation is ignored.

The method of Sun and Gonzalez(12) is used to calculate radiation heat transfer. The equations used are summarized below.

$$q''_{wedr} = F_{we} S (T_w^4 - T_{sat}^4)$$
 (2-100)

$$q''_{wvdr} = F_{wv} S (T_w^4 - T_v^4)$$
 (2-101)

$$q''_{evdr} = F_{ev} S (T_v^4 - T_{sat})$$
 (2-102)

where

q''www.dr' q''www.dr' q''www.dr' = radiation heat flux from wall to droplets, wall to vapor, and vapor to droplet, respectively.

Fwe, Fwv, Fev = gray body factors

 $T_w$ ,  $T_{sat}$ ,  $T_v$  = wall, droplet, vapor temperature, respectively (°R)

S = Stefan-Boltzmann constant

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.

The gray-body factors  $F_{wQ}$ ,  $F_{wv}$ ,  $F_{vQ}$  are defined as follows:

$$F_{wQ} = \frac{1}{R_2 (1 + R_3/R_1 + R_3/R_2)}$$
(2-103)

$$F_{WV} = \frac{1}{R_1 (1 + R_3/R_1 + R_3/R_2)}$$
(2-104)

$$F_{v2} = \frac{1}{R_2 (1 + R_1/R_2 + R_1/R_3)}$$
(2-105)

$$R_{1} = \frac{1 - e_{v}}{e_{v} (1 - e_{v} e_{\ell})}$$
(2-106)

$$R_2 = \frac{1 - e_{\varrho}}{e (1 - e_{v} e_{\varrho})}$$
(2-107)

$$R_{3} = \frac{1}{1 - e_{v} e_{u}} + \frac{1 - e_{w}}{e_{w}}$$
(2-108)

where

# ev, ev, e = vapor, droplet, wall emissivity, respectively

The wall emissivity is taken to be 0.9. Assuming an optically thin vapor and droplet medium, the vapor and droplet emissivity can be expressed as shown below.

$$e_v = 1 - e^{-a_v L_m}$$
 (2-110)

$$e_{g} = 1 - e^{-a_{g}L_{m}}$$
 (2-111)

where

 $a_v$ ,  $a_q$  = absorption coefficient for vapor and droplets, respectively

L = mean beam length

Based on a survey of a variety of geometries, the mean beam length can be taken to be 0.9 of the hydraulic diameter  $(D_h)$  for a rod array.

$$L_{p} = 0.9 D_{p}$$
 (2-112)

The vapor absorption coefficient is computed from Kirchner.<sup>(1)</sup>

$$a_{v} = p \left(\frac{1000}{T_{v}}\right)^{2} - 0.03 \left(\frac{1000}{T_{v}}\right)^{4}$$
(2-113)  
$$a_{e} = X_{a} \frac{\pi D_{d}^{2}}{4} N_{e}$$
(2-114)

where:

P = system pressure in atmospheres
X<sub>a</sub> = absorption efficiency
N<sub>2</sub> = droplet number density (no. of droplets/ft<sup>3</sup>)

Assuming that the droplets are in the geometric scattering regime, it can then be shown that  $X_a$  is equal to 0.74. The geometric scattering regime is defined as one in which  $D_d >> \lambda_r$  where  $D_d$  = droplet diameter and  $\lambda_r$  = characteristic length of radiation. For a wall temperature of 1800°F, Wien's<sup>(13)</sup> displacement law gives  $\lambda_r = 2.3\mu$  (7.6 x 10<sup>-6 ft</sup>). During reflooding, the typical drop diameter ranges from 0.002 to 0.008 foot. Hence, the assumption is justified that the droplets are in a geometric scattering regime.

# 2-19. COMPARISON OF BART FLUID MODELS WITH DATA

The initial droplet diameter,  $D_{d_0}$ , the Weber number, and the initial void fraction of the dispersed flow regime,  $\alpha_d$ , were identified in the preceding paragraphs as quantities which must be determined by interpretation of available data.

Many correlations exist for the determination of droplet size in various kinds of equipment.<sup>(14)</sup> In general, these correlations show that

$$D_{d_0} = \frac{C}{U_r^n}$$
(2-115)

where the proportionality constant C is a function of system geometry and fluid properties, and  $U_r$  is the initial relative velocity between the phases.

A possible mechanism for droplet formation during reflood is the shearing action between liquid and vapor which occurs above the quench front, where large liquid volumes still exist [figure 2-7(b)]. The initial relative velocity  $U_r$  in equation (2-115) would then be  $U_{rf}$  defined by equation (2-71).

The success of a particular choice for  $D_{d_0}$  can be determined indirectly by comparing BART calculations with data because the heat transfer in dispersed flow depends strongly on the initial droplet size used.

The void fraction  $\alpha_d$  which determines the upper limit of the transition region can also be determined from data because the magnitude of heat transfer from liquid to vapor near the quench front depends on this parameter.

Figures 2-11 through 2-16 illustrate the effect of varying the parameters  $D_{d_0}$  and  $\alpha_d$  on the BART calculation for Flecht test 13404. (Table 4-1 gives test conditions.) The larger droplet diameters reduce heat transfer. ( $D_{d_0}$  is given in feet in the figures.) The effect of increasing the liquid fraction at which pure droplet flow begins is relatively small.

A Weber number of 7 was used in the BART calculations. It was found that the small initial droplet sizes used precluded any subsequent droplet breakup. Consequently, variations in the Weber number within reasonable limits will have no effect on the calculations.

Comparisons with several additional FLECHT tests led to the following parameter values for best overall agreement with data:



A complete set of test predictions is presented in section 4. In these predictions the above parameters were used.



Rod Temperature (Test 13404)









(Test 13404)



Coefficient (Test 13404)



# SECTION 3

#### ROD HEAT RELEASE MODEL

#### 3-1. ROD MODEL FOR NORMAL RADIAL CONDUCTION CALCULATIONS

A fuel rod radial conduction model determines the transient rod temperatures. Of particular interest is the temperature of the outside wall of the cladding, T, which is required to obtain the heat flux from the rod to the coolant. It is given by

 $q^* = h(T_u - T_a)$ 

where T<sub>a</sub> is the coolant temperature. The axial conduction along the rod is neglected in comparison to the normal radial conduction in the rod.

The rod conduction model, along with the fluid flow model, will predict the peak cladding temperatures occurring during the reflood phase of the loss of coolant accident. The two-dimensional axial conduction model near the quench front is treated separately in paragraph 3-5.

#### 3-2. Formulation and Equations

Consider a fuel rod which consists of a fuel pellet, cladding surrounding the pellet, and a gap between the pellet and cladding (shown in figure 3-1).<sup>(15)</sup>

The heat balance equation gives

$$\frac{\partial}{\partial r} (rq^{*}) + rq^{**} = r\rho C_{p} \frac{\partial T}{\partial t}$$
(3-1)



Figure 3-1. Diagram of Fuel Rod Components

where

r = radius (ft) q" = heat flux ( $Btu/ft^2$ -sec) q''' = internal heat generation rate ( $Btu/ft^3$ -sec)  $\rho$  = density ( $lb_m/ft^3$ ) C<sub>p</sub> = heat capacity ( $Btu/lb_m^{-\circ}F$ ) T = temperature (°F) t = time (sec)

Substituting q" from Fourier's law

$$q^{\prime\prime} = -K \frac{\partial T}{\partial r}$$

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3-2

(3-2)

-

into equation (3-1), and rearranging, yields the familiar heat conduction equation applied to the fuel pellet region.

$$\frac{1}{r} \frac{\partial}{\partial r} \left( r \kappa \frac{\partial T}{\partial r} \right) = \left( \rho C_p \frac{\partial T}{\partial t} \right) - q^{\prime \prime \prime} \qquad (3-3)$$

where K = thermal conductivity in Btu/sec-ft-°F.

The rod is considered to be symmetric; thus q" = 0 at the center of the rod, or

$$\frac{\partial T}{\partial r} = 0 \tag{3-4}$$

The heat flux across the gap is

$$q'' = h_{gap} (T_F - T_C)$$
 (3-5)

where

T<sub>F</sub> = temperature at fuel boundary (°F)

$$T_{C}$$
 = temperature at the inner cladding boundary (°F)

The heat balance equation, (3-3), with  $q^{*'} = 0$ , also applies to the cladding region:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(rK\frac{\partial T}{\partial r}\right) = \frac{\partial}{\partial t}\left(\rho C_{p}T\right)$$
(3-6)

At the outer cladding boundary

$$\kappa \frac{\partial T}{\partial r} = h \left( T_{w} - T_{a} \right)$$
(3-7)

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where

- h = heat transfer coefficient between the cladding wall and the ambient coolant (Btu/sec-ft<sup>2</sup>-°F)
- $T_{u}$  = cladding wall temperature (°F)
- $T_a = amblent temperature (°F)$

Equations (3-3) through (3-7) govern the transient heat transfer behavior of the fuel rod. This system of differential equations is solved numerically.

### 3-3. Method of Solution

Figure 3-2 shows the cross section of a fuel rod of radius r, which has been divided into increments of width  $\Delta r$ . Consider a point i within the fuel pellet at time t = n $\Delta t$ . If the temperature at this point is known, then temperature at a new time, t = (n+1) $\Delta t$ , can be determined from the heat balance equation. Writing equation (3-3) in finite difference form yields this equation:

$$\frac{1}{r_{1}} \frac{1}{\Delta r} \left[ \left( \kappa_{1+\frac{1}{2}}^{n} r_{1+\frac{1}{2}} \frac{T_{1+1}^{n+1} - T_{1}^{n+1}}{\Delta r} \right) - \left( \kappa_{1-\frac{1}{2}}^{n} r_{1-\frac{1}{2}} \frac{T_{1-\frac{1}{2}}^{n+1} - T_{1-\frac{1}{2}}^{n+1}}{\Delta r} \right) \right]$$
(3-8)  
=  $\left( \rho C_{p} \right)_{1}^{n} \frac{T_{1-\frac{1}{2}}^{n+1} - T_{1}^{n}}{\Delta t} - \left( q^{(++)} \right)_{1}^{n+1}$ 



Figure 3-2. Cross Section of a Fuel Rod

Equation (3-8) contains three unknown temperodures at points i-1, i, and i+1. When this equation is written for each point within the rod, plus an equation at each boundary, a set of N equations results with N unknowns which can be solved simultaneously. This method of solution is known as the implicit finite difference method.

By rearranging the terms in equation (3-8), one obtains a finite difference form of the heat conduction equation:

$$\begin{bmatrix} 1 - \frac{\Delta r}{2r_{1}} - \frac{\kappa_{1+1} - \kappa_{1-1}}{4\kappa_{1}} \end{bmatrix} T_{1-1} + \begin{bmatrix} -2 - \frac{(\Delta r)^{2}}{\Delta t} \left(\frac{\rho C_{p}}{\kappa}\right)_{1} \end{bmatrix} T_{1} + \\ + \begin{bmatrix} 1 + \frac{\Delta r}{2r_{1}} + \frac{\kappa_{1+1} - \kappa_{1-1}}{4\kappa_{1}} \end{bmatrix} T_{1+1} = -\left(\frac{\rho C_{p}}{\kappa}\right)_{1} \frac{\Delta r^{2}}{\Delta t} T_{1} - \frac{q_{1}^{*} \Delta r^{2}}{\kappa_{1}}$$
(3-9)

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where the T, denotes the known temperature at time  $t = n\Delta t$ .

In short, at any interior point, the heat conduction equation can be approximated by a set of linear equations of the form

$$B_{1}T_{1-1} + A_{1}T_{1} + C_{1}T_{1+1} = D_{1}$$
(3-10)

where  $A_1$ ,  $B_1$ ,  $C_1$  and  $D_1$  are the coefficients defined as follows:

$$A_{1} = \left[ -2 - \frac{\Delta r^{2}}{\Delta t} \left( \frac{\rho C_{p}}{K} \right)_{1} \right]$$

$$B_{1} = \left[ 1 - \frac{\Delta r}{2r_{1}} - \frac{K_{1+1} - K_{1-1}}{4K_{1}} \right]$$

$$C_{1} = \left[ 1 + \frac{\Delta r}{2r_{1}} + \frac{K_{1+1} - K_{1-1}}{4K_{1}} \right]$$

$$D_{1} = \left[ - \left( \frac{\rho C_{p}}{K} \right)_{1} \frac{\Delta r^{2}}{\Delta t} T_{1} - \frac{\Delta r^{2}}{K_{1}} q_{1}^{*} \right]$$

4

At the center point r = 0 (1=1), equation (3-3) has a limiting value. By integrating equation (3-3) over the interval r = 0 to  $r = \Delta r/2$ , or by doing an energy balance for cylindrical element of radius  $\Delta r/2$ , one obtains the following finite difference equation for 1=1:

$$\left[4 + \frac{\Delta r^2}{\Delta t} \left(\frac{\rho C_p}{\kappa}\right)_1\right] T_1 + 4.0 T_2 = -\frac{\Delta r^2}{\Delta t} \left(\frac{\rho C_p}{\kappa}\right)_1 T_1 - \frac{\Delta r^2}{\kappa} q_1^{\prime \prime \prime} \quad (3-11)$$

To get the finite difference equation for 1 = m (the fuel outer boundary), a fictitious temperature  $T_{fl}$  at a distance  $\Delta r$  from the fuel boundary is defined such that

$$T_{f1} = T_{m-1} - \frac{2\Delta r}{K_m} h_{gap} (T_m - T_{m+1})$$
 (3-12)

satisfies the boundary condition of equation (3-5).  $T_{m+1}$  is the cladding inner boundary temperature.

Thus, for 1 = m

$$2T_{m-1} + \left[A_m - \frac{2\Delta r}{K_m}h_{gap} C_m\right]T_m + \left[\frac{2\Delta r}{k_m}h_{gap} C_m\right]T_{m+1} = 0_m \qquad (3-13)$$

where  $A_m$  and  $D_m$  are as defined for equation (3-10) and

$$C_{m} = 1 + \frac{\Delta r}{2r_{m}} + \frac{K_{f1} - K_{m-1}}{4K_{m}}$$
(3-14)

 $K_{f1}$  is thermal conductivity based on temperature  $T_{f1}$ .

Because the heat conduction equation (3-9) applies to the cladding with q''' = 0, the D<sub>1</sub>'s in equation (3-10) become

$$D_{1} = \left[ -\frac{\Delta rc^{2}}{\Delta t} - \left( \frac{\rho C_{p}}{\kappa} \right)_{1} T_{1}^{*} \right]$$
(3-15)

and  $A_1$ ,  $B_1$ , and  $C_1$  are as defined for equation (3-10) with  $\Delta r$  replaced by  $\Delta r_c$ .  $\Delta r_c$  is the width of the radial increment in the cladding.

A fictitious temperature  $T_{f2}$  is defined at a distance  $(-\Delta r_c)$  from the inner cladding surface, yielding for i = m+1 (cladding inner boundary), in a manner similar to that of determining the fuel outer boundary.

$$T_{f2} = T_{m+2} + \frac{2\Delta r_c}{K_{m+1}} h_{gap} (T_m - T_{m+1})$$
(3-16)

and

1

$$\begin{bmatrix} \frac{2\Delta r_{c}}{K_{m+1}} h_{gap} B_{m+1} \end{bmatrix} T_{m} + \begin{bmatrix} A_{m+1} - \frac{2\Delta r_{c}}{K_{m+1}} h_{gap} B_{m+1} \end{bmatrix} T_{m+1} + 2T_{m+2} = D_{m+1}$$
(3-17)

where  $A_{m+1}$  and  $D_{m+1}$  are as defined for equation (3-10) and equation (3-15), respectively, and

$$B_{m+1} = 1 - \frac{\Delta r_c}{2r_{m+1}} - \frac{K_{m+2} - K_{f2}}{4 K_{m+1}}$$
(3-18)

 $K_{f2}$  is the thermal conductivity at temperature  $T_{f2}$ .

Finally, at the cladding outer boundary (1 = n)

$$2I_{n-1} + \left[A_n - \frac{2\Delta r_c}{K_n} + C_n\right] T_n = D_n - \frac{2\Delta r_c}{K_n} + C_n T_A \qquad (3-19)$$

where

$$C_n = 1 + \frac{\Delta r_c}{2r_n} + \frac{K_{f3} - K_{n-1}}{4 k_n}$$
 3-20)

K<sub>F3</sub> is evaluated at T<sub>F3</sub>.

$$T_{f3} = T_{n-1} - \frac{2\Delta r_c}{K_n} h (T_n - T_A)$$
 3-21)

In figure 3-2,  $N_p + 1$  and  $N_c + 1$  are unknown temperatures in region 1 and region 11. For a total of  $(N_p - 1) + (N_c - 1)$  interior points, there are an equal number of equations of form of equation (3-10). At four boundary points 1 = 1, 1 = m, 1 = m + 1, and 1 = n, the unknown temperatures are related to

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(3-13), (3-17), and (3-19). Therefore, there is a linear system of  $(N_p + N_c + 2)$  equations for as many unknown temperatures. Furthermore, if these equations are written in the order of increasing i, the coefficient matrix is tri-diagonal with diagonal dominance. Therefore, a unique solution vector is guaranteed and can be obtained easily.

Once the temperature of the outside cladding wall  $(T_{y})$  is known, the heat flux can be calculated from the following equation:

$$q'' = h(T_1 - T_2)$$
 (3-22)

## 3.4. Housing Heat Conduction Model

To verify the BART model, a housing heat conduction model has been included in the BART code to resemble the FLECHT test facility. The housing heat release is assumed to be distributed evenly across each flow channel. The heat conduction model is a simplified one-dimensional transient heat conduction model with axial conduction assumed to be negligible. A slab geometry has been used, as the radius of the housing is large compared to its thickness. A convective boundary condition is used on the inside surface, and the outside of the housing is considered to be insulated.

### 3-5. Quench Front Model

In most reflood experiments, a well-defined interface can be identified on the heater rod across which surface temperatures change suddenly from values on the order of 1000°F to values near the fluid saturation temperature (figure 3-3). This interface is usually termed a "quench front." Such a front exists because the heat transfer coefficient increases by orders of magnitude as the surface temperature drops below the wetting temperature. The axial distance across which most of the temperature change occurs, and the speed at which the front moves along the rod, depends on the surface heat transfer and on the composition of the rod. In most situations this distance is small, because the heat flowing axially through the rod due to conduction is of the same order of magnitude as the heat flowing from the rod to the liquid due to boiling.

14.380-9



Figure 3-3. Typical Transient Midplane Cladding Temperature Behavior for Constant Flooding Rate Tests

the heat flowing axially through the rod due to conduction is of the same order of magnitude as the heat flowing from the rod to the liquid due to boiling.

Because the quench front which propagates from the bottom of the heated rod bundle or nuclear core is most important in determining the overall heat transfer in those situations where only bottom flooding is occurring, this case has been analyzed in the most detail.

# 3-6. Isotherm Migration Method

BART considers the following version of the quench front problem. An infinite strip is given, such that relative to the polar coordinates r and z, it is assumed to correspond to a section of a long, axially symmetric fuel rod consisting of fuel, gap, and cladding regions. T (r, z, t) denotes the temperature, completing the initial boundary value problem.

$$\rho c \frac{\partial T}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} (kr \frac{\partial T}{\partial r}) + \frac{\partial}{\partial z} (k \frac{\partial T}{\partial z}) + 0 \qquad (3-23)$$

$$0 < r < r_{f}$$

$$r_{c} < r < r_{w}$$

$$r_{o} < z < \infty$$

$$T (r, -\infty, t) = T_{s}, T(r, \infty, t) = T_{w} \qquad (3-24)$$

$$0 < r < r_{f}$$

$$r_{c} < r < r_{w}$$

$$t > 0$$

$$\frac{\partial T}{\partial r} (0, z, t) = 0 \qquad (3-25)$$

$$-\infty < z < \infty$$

$$t > 0$$

$$75728: 1b/032284 \qquad 3-11$$

$$- k \frac{\partial I}{\partial r} (r_{f}, z, t) = - k \frac{\partial I}{\partial r} (r_{c}, z, t)$$
$$= h_{gap} \left[ T(r_{f}, z, t) - T(r_{c}, z, t) \right]$$
(3-26)

(3-27)

(3-28)

.

-co < z < ro t > 0

$$-k \frac{\partial T}{\partial r} (r_w, z, t) = q^{*} (z, \bar{1}, t)$$

where

 $q^{*} = \begin{cases} q_{NB}^{*} \ (z, T, t) \text{ if } T \leq T^{*} \\ q_{FB}^{*} \ (z, T, t) \text{ if } T > T^{*} \end{cases}$ 

 $T(r, z, 0) = T^{0}(r, z)$ 

0 < r < r r c r < r -∞ < z < ∞

These definitions apply to equations (3-23) through (3-28):

p(r, z, t, T)		density				
C(r, z, t, T)		specific heat				
k(r, z, t, T)		thermal conductivity				
Q(r, z, t)		external heat source				
T <sub>s</sub> T <sub>u</sub>	=	sink and wall temperatures				
hgap	=	heat transfer coefficient for the gap				
9"NB . 9"FB	=	heat transfer functions at the wall				
T*	-	wetting temperature				
$T^{O}(r, z)$		Initial condition				

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Typically,  $0 = q_{FB}^{*} < q_{NB}^{*}$ , and it is this property of equation (3-27) which generates the thermal wave. Simpler versions of this problem involving only one region (fuel or cladding only) have been treated by Duffy and Porthouse, <sup>(16)</sup> Yeh, <sup>(17)</sup> Durack and Wendroff, <sup>(18)</sup> and Dendy, Swartz, and Wendroff<sup>(19)</sup> among others. The inclusion of the gap and its associated boundary condition appears to be new (although Avila and Sun have considered the gap condition in some unpublished work).

The technique of isotherm migration to study heat conduction phenomena has very desirable characteristics when applied to the quench front problem. The basic idea is to interchange the dependent sole of the space coordinate z. Thus, instead of writing T = T(r, z, t), z = (r, T, t) can be written. For this inversion to be meaningful, however, it must be assumed that for fixed r and t, T is a strictly monotone (increasing) function of z. Assuming that this is the case, equation (3-23) can then be transformed to obtain an equation for the new dependent variable z. The interchange of T and z as independent and dependent variables thus establishes a new (r, T, t) coordinate system.

As a numerical method, isotherm migration is particularly suited to the calculation of steep traveling wave fronts because the steep moving front in the (r, z) system becomes a translating flat front in the (r, T) system. Correspondingly, the large number of mesh points required to resolve the front in the (r, z) system reduces to a moderate number in the (r, T) system.

In the past, numerical implementations of the isotherm migration method have used explicit finite difference equations to track the positions of the isotherms. In this case a common requirement for stable calculations is that

$$\Delta t < \frac{\left(\frac{\partial z}{\partial T}\right)^2 \rho c}{2k} (\Delta T)^2$$
(3-29)

where  $\Delta t$  and  $\Delta T$  denote the t and T mesh spacings. Furthermore, because equation (3-29) is obtained by neglecting the diffusion effects in the radial

direction, the maximum stable time step At may be considerably less than the right side of equation (3-29). This in turn leads to a prohibitive computational effort in the case of typical quench front tracking problems.

In view of this potential difficulty, the finite difference equations that are developed in paragraph 3-7 are implicit. It turns out that for any  $\Delta t > 0$ , the finite difference equations are "of positive type," according to Forsythe and Wasow, <sup>(20)</sup> and therefore have favorable stability properties. While it is true that it requires the solution of a nontrivial linear system to advance the solution from one time level to the next, this system is highly structured and has a bandwidth on the order of the number of temperature mesh lines. Because the isotherm migration method promotes the use of a coarse temperature mesh, it is to be expected that the solution of the linear system will proceed rapidly.

## 3.7. Transformed System and Finite Difference Equations

The fundamental identity defining the inversion of T and z in the isotherm migration method is  $z \equiv z[r, T(r, z, t), t]$ . Differentiation obtains

$$\frac{\partial z}{\partial T} \frac{\partial T}{\partial z} = 1$$

$$\frac{\partial z}{\partial t} + \frac{\partial z}{\partial T} \frac{\partial T}{\partial t} = 0$$

$$\frac{\partial z}{\partial r} + \frac{\partial z}{\partial T} \frac{\partial T}{\partial r} = 0$$

$$\frac{\partial^2 z}{\partial r^2} + 2 \frac{\partial^2 z}{\partial r \partial T} \frac{\partial T}{\partial r} + \frac{\partial^2 z}{\partial T^2} \left(\frac{\partial T}{\partial r}\right)^2 + \frac{\partial z}{\partial T} \frac{\partial^2 T}{\partial r^2} = 0$$

The following expression is deduced from these expressions:

$$\frac{1}{r} \quad \frac{\partial}{\partial r} \left(\frac{T}{\partial r}\right) = \frac{1}{\frac{\partial z}{\partial T}} \left\{ \frac{\partial}{\partial T} \left[ \frac{\left(\frac{\partial z}{\partial r}\right)^2}{\frac{\partial z}{\partial T}} \right] - \frac{1}{r} \quad \frac{\partial}{\partial r} \left(r \quad \frac{\partial z}{\partial r}\right) \right\}$$

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$$\frac{\partial^2 T}{\partial z^2} = \frac{1}{\frac{\partial z}{\partial T}} \quad \frac{\partial}{\partial T} \left( \frac{1}{\frac{\partial z}{\partial T}} \right)$$

and it then follows that equation (3-23) transforms into

$$\rho_{c} \frac{\partial z}{\partial t} + \frac{\partial}{\partial T} \left\{ \frac{k[(\frac{\partial z}{\partial r})^{2} + 1]}{\frac{\partial z}{\partial T}} \right\} - \frac{1}{r} \frac{\partial}{\partial r} (rk \frac{\partial z}{\partial r}) + Q \frac{\partial z}{\partial T} = \frac{-\partial k}{\partial z} = 0 \quad (3-30)$$

 $0 < r < r_{f}$   $r_{c} < r < r_{w}$   $T_{s} < T < T_{w}$ t > 0

If temperature T(r, z, t) is a smooth function of z, then equation (3-24) implies that

$$\frac{\partial z}{\partial T} (r, T_{s}, t) = \frac{\partial z}{\partial T} (r, T_{w}, t) = \infty$$

$$0 < r < r_{f}$$

$$r_{c} < r < r_{w}$$

$$t > 0$$

$$(3-31)$$

Moreover, the initial condition equation (3-28) becomes

$$z(r, T, 0) = z^{0}(r, T)$$
 (3-32)

0 < r < r r<sub>c</sub> < r < r T<sub>s</sub> < T < T

where  $z^{0}(r, T)$  is the function inverse to  $T^{0}(r, z)$ .

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For transformed boundary conditions, equations (3-25) and (3-27) give

$$\frac{\partial z}{\partial r}(0, T, t) = 0$$
 (3-33)

 $T_s < T < T_w$ t > 0

and

1

$$k \frac{\frac{\partial z}{\partial r}}{\frac{\partial z}{\partial T}} = q^{n} (z, T, t)$$
(3-34)
$$(r_{w}, T, t)$$

t > 0

Hence the gap condition on the fuel side becomes

a,c

(3-35)

a,c

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.

In the same way, the transformed cladding side gap condition is

(3 - 36)

a.c

a,c

where now  $z_{g}$  is the position of a reference temperature  $T_{g}$  on  $r = r_{f}$ .

The motion of the isotherms is governed by the system of equations (3-30) through (3-36). Note the following about this system:

o In equations (3-35) and (3-36), the reference temperatures must coincide with the values of T on certain T coordinate lines. This is essential for the formulation of the implicit equations.

We now proceed to the difference equations. As shown in figure 3-4, the open region  $0 < r < r_f$ ,  $r_c < r < r_w$ ,  $T_s < T < T_w$  is overlaid with a rectangular mesh:

 $r = r_{1}, 1 = 1, ..., 1; r_{1} < r_{1} + 1$ 

$$T = T_1, J = 1, ..., J; T_1 = T_5 + J\Delta T$$

with the uniform temperature increment  $\Delta T \equiv (T_{w} - T_{s})/(J + 1)$ . A uniformly spaced temperature mesh has been chosen only to simplify the presentation.



Figure 3-4. Diagram of the Isotherm Temperature Mesh

 $z(r_1, T_j, t_{n+1})$  is denoted by  $z_{1,j}^{n+1}$  where  $t_{n+1} = t_n + \Delta t$  and  $\Delta t$  is the (nonuniform) time step. Furthermore,  $c(r_1, z_{1,j}^n, t_n, T_j) = C_{1,j}^n$ , and so forth. At the mesh points (intersections of the previously defined mesh lines), equation (3-30) is replaced by an appropriate difference equation. Consider first the term  $\{(a/at)[d/(az/aT)]\}$ , where  $d \equiv k [(az/ar)^2 + 1]$ . If subscript  $j \pm \frac{1}{2}$  denotes evaluation at  $T_j \pm \frac{\Delta T}{2}$ , then by Taylor's theorem, as any time in the interval  $[t_n, t_{n+1}]$ 

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$$\begin{split} & \frac{\partial}{\partial T} \left( \frac{d}{\partial z} \right)_{1,j} = \frac{\left( \frac{d}{\partial z} \right)_{1,j+\frac{1}{2}} - \left( \frac{d}{\partial z} \right)_{1,j-\frac{1}{2}}}{\Delta T} + 0 \left( \Delta T \right)^{2} \\ & = \left[ \left( \frac{d_{1,j+\frac{1}{2}}}{\left( \frac{z_{1,j+\frac{1}{2}}}{\Delta T} - \frac{z_{1,j-1}}{\Delta T} \right)} - \left( \frac{d_{1,j-\frac{1}{2}}}{\Delta T} \right) \right] + 0 \left( \Delta T \right)^{2} \\ & = \frac{d_{1,j+\frac{1}{2}} \left( \frac{z_{1,j-\frac{z_{1,j-1}}{\Delta T}}}{\left( \frac{z_{1,j-\frac{z_{1,j-1}}{\Delta T}} \right)} - \frac{d_{1,j-\frac{1}{2}}}{\left( \frac{z_{1,j-\frac{z_{1,j-1}}{\Delta T}} \right)} \right] + 0 \left( \Delta T \right)^{2} \\ & = \frac{d_{1,j+\frac{1}{2}} \left( \frac{z_{1,j+\frac{1}{2}}}{\left( z_{1,j+1} - z_{1,j} \right) \left( z_{1,j-\frac{z_{1,j-1}}{\Delta T} \right)} + \frac{d}{1,j-\frac{1}{2}} \left( \frac{z_{1,j-\frac{z_{1,j+1}}{\Delta T}} - z_{1,j+1} \right)}{\left( z_{1,j+1} - z_{1,j} \right) \left( z_{1,j-\frac{z_{1,j-1}}{\Delta T} - z_{1,j-1} \right)} + 0 \left( \Delta T \right)^{2} \\ & = \frac{d_{1,j+\frac{1}{2}} \left( \frac{z_{1,j+\frac{1}{2}} - z_{1,j-1} \right) + d_{1,j-\frac{1}{2}} \left( \frac{z_{1,j} - z_{1,j+1} \right)}{\left( z_{1,j+1} - z_{1,j} \right) \left( z_{1,j-\frac{z_{1,j-1}}{\Delta T} - z_{1,j-1} \right)} + 0 \left( \Delta T \right)^{2} + \Delta t \right] \end{split}$$

From this it follows naturally that

$$\frac{\partial}{\partial T} \left( \frac{d}{\partial z} \right)_{1,j} \leftarrow \frac{D_{1,j+\frac{1}{2}}^{n} (Z_{1,j}^{n+1} - Z_{1,j-1}^{n+1}) + D_{1,j-\frac{1}{2}}^{n} (Z_{1,j}^{n+1} - Z_{1,j+1}^{n+1})}{(Z_{1,j+1}^{n} - Z_{1,j}^{n})(Z_{1,j}^{n} - Z_{1,j-1}^{n})}$$

where  $a \leftarrow b$  means "b replaces a",  $Z_{1,j}^n$  is the finite difference approximation of  $z_{1,j}^n$ 

$${}^{D}_{1,j+\frac{1}{2}} \equiv {}^{K}_{1,j+\frac{1}{2}} \left[ \left[ \frac{Z_{1+1,j}^{n} - Z_{1-1,j}^{n} + Z_{1+1,j+1}^{n} - Z_{1-1,j+1}^{n}}{2(r_{1+1} - r_{1-1})} \right]^{2} + 1 \right]$$
(3-37)

and

$$K_{1,j+\frac{1}{2}} \equiv k \left( r_{1}, z_{1,j}^{n}, t_{n}, \frac{T_{j} + T_{j+1}}{2} \right)$$

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A similar but less involved line of reasoning leads to

$$-\frac{1}{r}\frac{\partial}{\partial r}\left(r + \frac{\partial z}{\partial r}\right) + (3-28)$$

$$\leftarrow \frac{(r_{1} + r_{1+1}) \kappa_{1+\frac{1}{2},j}^{n} \left( \frac{z_{1,j}^{n+1} - z_{1+1,j}^{n+1}}{r_{1+1} - r_{1}} \right) + (r_{1} + r_{1-1}) \kappa_{1-\frac{1}{2},j}^{n} \left( \frac{z_{1,j} - z_{1-1,j}^{n+1}}{r_{1} - r_{1-1}} \right)}{r_{1} (r_{1+1} - r_{1-1})}$$

where  $k_{1+1/2,j}^n$  is an obvious approximation of  $k_{1+1/2,j}^n$ .

The remaining replacements in equation (3-30) are straightforward:

$$(\rho c \frac{\partial z}{\partial t})_{1,j} \leftarrow (RC)_{1,j}^{n} \left( \frac{Z_{1,j}^{n+1} - Z_{1,j}^{n}}{\Delta t} \right)$$
 (3-39)

and

$$(Q \frac{\partial z}{\partial T})_{1,j} \leftarrow Q_{1,j}^{n} \frac{(z_{1,j}^{n+1} - z_{1,j-1}^{n+1})}{\Delta T} (Q_{1,j}^{n} \ge 0)$$
 (3-40)

RC denotes the analog of K for the product  $\rho c$ .

If equations (3-37) through (3-40) are used, the difference equation resulting from equation (3-30) may be written as follows:

$$A_{1,j}^{n} Z_{1,j}^{n+1} - A_{1,j-1}^{n} Z_{1,j-1}^{n+1} - A_{1,j+1}^{n} Z_{1,j+1}^{n+1} - A_{1-1,j}^{n} Z_{1-1,j}^{n}$$

$$- A_{1+1,j}^{n} Z_{1+1,j}^{n+1} = b_{1,j}^{n}$$
(3-41)

where  $A_{1,j}^n$ ,  $A_{1\pm1,j}^n$ ,  $A_{1,j\pm1}^n$  and  $b_{1,j}^n$  are readily constructed from equations (3-37) through (3-40) and, assuming that  $Z_{1,j}^n$  is a strict monotone function of J, are seen to satisfy the inequalities,  $A_{1\pm1,j}^n \ge 0$ ,  $A_{1,j\pm1}^n \ge 0$ , and  $A_{1,j}^n \ge A_{1,j-1}^n$ +  $A_{1,j+1}^n$  +  $A_{1-1,j}^n$ ,  $A_{1+1,j}^n$ .

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These conditions define an equation of positive type. In the linear case, a system of such equations is known to be unconditionally stable. (20)

Equation (3-41) applies for 1 < j < T and either  $r_0 < r_1 < r_f$  or  $r_c < r_1 < r_w$ . The remaining cases require either a modification of terms in equation (3-41) or a new difference equation. For example, if j = J and either  $r_0 < r_1 < r_f$  or  $r_c < r_1 < r_w$ , the effect of the boundary condition of equation (3-31) is incorporated as follows. With  $d = [k (\partial z/\partial r)^2 + 1]$  as before,

$$\frac{\partial}{\partial T} \left( \frac{d}{\partial z} \right)_{1,j} = \frac{\left( \frac{d}{\partial z} \right)_{1,w} - \left( \frac{d}{\partial z} \right)_{1,j-1/2}}{\frac{3\Delta T}{2}} + 0 \quad (\Delta T)$$

$$= -\frac{2}{3} \frac{d_{1,J-1/2}}{z_{1,J} - z_{1,J-1}} + 0 (\Delta T) = \frac{2}{3} \frac{d_{1,J-1/2}^{n}}{z_{1,J-1}^{n+1} - z_{1,J}^{n+1}} + 0 (\Delta T + \Delta t)$$

$$= \frac{2}{3} d_{1,J-1/2}^{n} \left[ \frac{2}{z_{1,J-1}^{n} - z_{1,J}^{n}} + \frac{z_{1,J}^{n+1} - z_{1,J-1}^{n+1}}{z_{1,J-1}^{n} - z_{1,J}^{n}} \right] + 0 (\Delta T + \Delta t) \quad (3-42)$$

Therefore, the term  $\frac{\partial}{\partial T} \left( \frac{d}{\frac{\partial z}{\partial T}} \right)$  in equation (3-30) is replaced by the expression

$$\frac{2}{3} D_{1,J-1/2}^{n} \left[ \frac{2}{z_{1,J-1}^{n} - z_{1,J}^{n}} + \frac{z_{1,J}^{n+1} - z_{1,J-1}^{n+1}}{z_{1,J-1}^{n} - z_{1,J}^{n}} \right]$$
(3-43)

where  $D_{1,J-1/2}^n$  is defined as it is for equation (3-37). In the same way, the analogous replacement for j = 1,  $r_0 < r_1 < r_f$  or  $r_c < r_1 < r_w$  is

$$\frac{2}{3} p_{1,3/2}^{n} \left[ \frac{2}{z_{1,2}^{n} - z_{1,1}^{n}} + \frac{z_{1,1}^{n+1} - z_{1,2}^{n+2}}{z_{1,2}^{n} - z_{1,1}^{n}} \right]$$
(3-44)

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Note that in both cases the result is a linear equation of positive type having the form of equation (3-41).

To close the set of implicit difference equations, it remains to consider the lines  $r = r_w$ ,  $r_c$ ,  $r_f$ ,  $r_o$ . On  $r = r_w$  if an "upwind" difference method is applied to equation (3-34), then

$$\kappa_{w,j}^{n} \frac{Z_{w,j}^{n+1} - Z_{w-1,j}^{n+1}}{r_{w} - r_{w-1}} = \frac{\Phi_{w,j}^{n}}{\Delta T} \begin{cases} Z_{w,j}^{n+1} - Z_{w,j-1}^{n+1} \\ \\ \\ Z_{w,j+1}^{n+1} - Z_{w,j}^{n+1} \end{cases} , j = 2, \dots, j-1 \quad (3-45)$$

where

$$\Phi_{w,j}^{n} = \begin{cases} \left(q_{NB}^{n}\right)^{n} & \text{if } T_{s} < T_{j} \leq T^{*} \\ \left(q_{FB}^{n}\right)^{n} & \text{if } T^{*} < T_{j} < T_{w} \end{cases}$$

and the upper or lower line of the bracketed term is to be used as is whether or not it is negative. Equation (3-45) introduces "w" as an index for mesh points on  $r_w$ . For j = 1 and j = J in equation (3-45), the corresponding boundary indexes j = 0, J + 1 are eliminated by the difference method.

The difference equation on  $r = r_f$  proceeds from a heat balance on a region adjacent to  $r = r_f$ . Let  $r = r_p$  denote the mesh line immediately in from  $r = r_f$ . If a heat balance is performed over the shaded region shown in figure 3-5, a rearrangement yields

Apc 
$$\frac{\partial T}{\partial t} = r k \frac{\partial T}{\partial r} \begin{vmatrix} r & f \\ r & \frac{\Delta \kappa}{\Delta z} \frac{\partial T}{\partial z} \end{vmatrix} z + \Delta z + AQ$$
 (3-46)

where  $\bar{r} = (r_p + r_f)/2$  and  $A = (r_f^2 - \bar{r}^2)/2$ . Letting  $\Delta z \rightarrow 0$ .

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Figure 3-5. Diagram of the Fuel Boundary Treatment

Apc 
$$\frac{\partial T}{\partial t} = A \frac{\partial}{\partial z} \left(k \frac{\partial T}{\partial z}\right) + r k \frac{\partial T}{\partial r} + AQ$$
 (3-47)

Transforming the terms in equation (3-47) as before yields

$$A_{pc} \frac{\partial z}{\partial T} = A \frac{\partial}{\partial T} \left(\frac{k}{\frac{\partial z}{\partial T}}\right) - \left(r k \frac{\frac{\partial z}{\partial r}}{\frac{\partial z}{\partial T}}\right|_{\bar{r}}^{r} \int \frac{\partial z}{\partial T} + AQ \frac{\partial z}{\partial T} = 0$$
(3-48)

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### which, by equation (3-35), may be written

a,c

(3-49)

a,c

Now equation (3-49) can be used to obtain a difference equation on  $r = r_{f}$ . The first, second, and third terms of equation (3-49) are differenced according to equations (3-39), (3-37), and (3-40). The third term is replaced by

$$\left[\vec{r} \; k \; (\vec{r}, \; t_{n}, \; T_{j}) \left( \frac{Z_{f,j+1}^{n} - Z_{f,j-1}^{n}}{Z_{j-1}^{n} - Z_{j-1}^{n}} \right) \right] \; \left( \frac{Z_{f,j}^{n+1} - Z_{p,j}^{n+1}}{r_{f} - r_{p}} \right)$$

where  $Z_j^n = (Z_{f,j}^n + Z_{p,j}^n)/2$ , while on the fourth term the quantity  $O[(z - z_m)^2]$ is neglected and the remainder is approximated by a,c

These replacements apply for 1 < j < J. For j = 1, J, the centered differences are replaced by appropriate one-sided differences.

The difference equation on  $r = r_c$  is developed by similar considerations.

a,c

For the r = 0 case, the same heat balance is performed as was done for the  $r_f$  and  $r_c$  cases, with the volume lying between r = 0 and  $r = r_{1/2}$ . The equation analogous to equation (3-47).

$$A\rho c \quad \frac{\partial z}{\partial t} = A \quad \frac{\partial}{\partial T} \quad \left(\frac{k}{\partial z}\right) \cdot \left(r \quad k \quad \frac{\partial z}{\partial T} \\ \frac{\partial z}{\partial T} \\ r = 0 \\ r$$

where  $A = \frac{r_1^2}{8}$ . Except for noting that by equation (3-33),  $\frac{\partial z}{\partial r} \bigg|_{r=0} = 0$ , the solution follows the standard form.

The linear system defined by the aggregate of these difference equations has a structure similar to that resulting from the usual "5-point" discretization of Laplace's equation. In factor, if m = 1 = j, the structures coincide.

# 3-8. HEAT TRANSFER NEAR AND AT THE QUENCH FRONT

### 3-9. Heat Transfer at the Quench Front

The physical picture of a quench front propagating upward with bottom flooding is shown in figure 3-6. Fluid approaching the quench front increases in steam quality or looses subcooling because of power generated in the rod. At distances of several rod diameters from the quench front, the heat flux may increase because residual stored energy in the interior of the rod is released. In the region where surface temperatures increase rapidly, violent boiling takes place, followed by a transition to film boiling when surface temperatures have exceeded the Liedenfrost, or wetting, temperature.

Because the boiling zone is extremely narrow, it is not possible to accurately determine the heat transfer coefficient within the zone. But most, if not all, of the vapor generated in this region will end up in the vapor film immediately above the quench front. A possible limit to the amount of vapor generated in the narrow boiling zone is suggested by considering the stability of the liquid vapor interface in the film boiling zone, because the rate at which vapor leaves the boiling zone should control the rate at which liquid can enter the boiling zone. If the stability of the interface is destroyed by excessive steam velocities in the film, the resulting increase in interfacial resistance should reduce the rate of liquid supply to the boiling zone, because vapor will accummulate there (figure 3-6). If film velocities are below the critical value, however, the interface will become relatively smooth. The resulting decrease in interfacial resistance should lead to higher steam velocities and higher liquid flow rates to the boiling zone. Accordingly, the heat transfer in the boiling zone is controlled by the liquid flow rate to the zone, which is in turn controlled by the vapor flow in the film.

The description presented above of heat transfer in the narrow boiling zone of the quench front follows closely that of Zuber in his analysis of the critical heat flux on a heated surface.<sup>(21)</sup> In both cases, the heat flux is related to the vapor flow rate from the surface by

(3-50)

where

4

 $A_{HT}$  = boiling area surface area  $A_F$  = flow area

and where  $j_{vc}$ , the vapor flux averaged over the flow area, is the maximum allowed by two-phase interface stability.



# Figure 3-6. Hydrodynamic and Heat Transfer Regions Near the Quench Front

In the case of a horizontal plate or rod, the boiling zone surface area is equivalent to the flow area. However, in a quench front traveling up a rod bundle, the boiling zone is defined by the rod perimeter and the boiling zone width (figure 3-6), whereas the flow area is the channel cross-sectional area.

Analysis of the equations governing the stability of an interface in the presence of a disturbance (21, 22, 9) leads to two equations describing the relative velocity at which instability occurs:

$$U_{ra} = \left[\frac{\sigma \ m^2 \ D_h}{\rho_v}\right]^{0.5}$$

$$U_{rb} = \left[\frac{\sigma \ m}{\rho_v}\right]^{0.5}$$
(3-51)
(3-52)

where

 $\sigma$  = surface tension

 $D_{h}$  = channel width or diameter

m = wave number of the disturbance on the interface =  $2\pi/\lambda$ , where  $\lambda$  is the wavelength of the disturbance

In general, equation (3-51) is applicable when  $D_h$  is of the same order as the disturbance wave length  $\lambda$ , and equation (3-52) is applicable when channel dimensions are larger than  $\lambda$ .

It has been observed (9,23) that the average wavelength of the disturbances on a vapor liquid film in film boiling can be well represented by the Taylor instability wavelength:

$$\lambda = 2\pi \left[ \frac{c_1 \sigma}{\Delta \rho g} \right]^{0.5}$$
(3-53)

where

 $1 \leq C_1 \leq 3$ 

Inserting equation (3-53) into equations (3-51) and (3-52) leads to

$$U_{ra} = \left[\frac{\Delta \rho g D_{h}}{C_{1} \rho_{v}}\right]^{0.5}$$
(3-54)

and

$$U_{rb} = \begin{bmatrix} \frac{\sigma \ \Delta \rho q}{C_1^2 \ \rho_v^2} \end{bmatrix}^{0.25}$$
(3-55)

It is expected that  $j_{vc}$  will be a function of the void fraction of the bulk fluid (that is, of the voids produced by heat release below the quench front). One way to include this effect is through equation (3-53), because, as a first approximation the term  $\Delta \rho g$ , the driving force due to density differences between the bulk and the film should be scaled by  $(1-\alpha_c)$ , the liquid fraction in the bulk fluid.

Equations (3-54) and (3-55) become

$$U_{ra} = \left[\frac{\Delta \rho g (1 - \alpha_c) D_h}{C_1 P_v}\right]^{0.5}$$
(3-56)

and

$$U_{rb} = \left[\frac{\sigma \Delta \rho g (1 - \alpha_c)}{c_1^2 \rho_v^2}\right]^{0.25}$$
(3-57)

Equations (3-56) and (3-57) exhibit the same sensitivity to pressure, because  $\sigma$  and  $\Delta\rho$  remain approximately constant between 15 and 60 psia and both equations are proportional to  $\rho_v = \frac{-1/2}{2}$ . The sensitivity of these two equations to  $\alpha_c$  is different, as are the magnitudes of U<sub>ra</sub> and U<sub>rb</sub>.

Because

$$l_v = \alpha_f U_r \tag{3-58}$$

where  $\alpha_{f}$  = the film void fraction in the channel (area of vapor film/area of channel),  $\alpha_{f}$  must be found before q" in equation (3-50) can be calculated. The value of  $\alpha_{f}$ , as well as the most correct form for U<sub>r</sub>, can be determined best by comparisons with data.

If the description of the heat transfer process in the boiling zone as one controlled by hydrodynamic stability is correct, two conditions must be true:

- o The heat transfer in the zone is virtually independent of the surface condition of the rod.
- o If the quench front is moving sufficiently slowly so that all the rod stored energy is released in the boiling zone, the heat transfer in the zone, and thus the quench front velocity, will depend on only the fluid conditions and the rod heat capacity, because a heat balance yields heat released in the boiling zone equal to maximum heat flow allowed by equation (3-50).

$$(\rho cA)_{rod} V_{q} (T_{q} - T_{sat}) = C_{2} \rho_{v} H_{qv} U_{r} A_{F}$$
(3-59)

where

 $(\rho cA)_{rod}$  = heat capacity of rod/unit length  $T_q$  = rod temperature immediately above quench front  $V_q$  = quench front velocity

Here the quench front has been treated as an interface across which the temperature of the rod changes from  $T_{g}$  to  $T_{sat}$ .

The first condition appears to be verified by quenching data obtained by Piggott and Duffey.<sup>(24)</sup> They showed that there was virtually no effect of surface finish on the quench velocity in bottom flooding.

According to equation (3-59),  $C_2$  must be nearly constant for the second condition to be verified. In the FLECHT tests, (25,10,26,27) the quench front velocity is slowest when the subcooling at the quench front is zero. One would therefore expect nearly all the rod-stored energy to be released in the boiling zone in these tests. The value of  $C_2$  was determined from equation (3-59) for two low subcooling FLECHT tests at 20 and 40 psia (runs 15713 and 13914) as well as a test at 60 psia (run 13711) at times and locations when conditions at the quench front were saturated. Table 3-1 lists test conditions for these and other FLECHT tests.

Table 3-1 lists the important test parameters involved and calculated value of  $C_2$ , using equations (3-56) and (3-57) and assuming  $C_1 = 1$ . The void fraction was determined from  $\Delta p$  cell data, and the velocity of the quence front was deduced from thermocouple data. (28) Figure 3-7 shows more clearly the dependence of  $C_2$  on void fraction. The value of  $C_2$  is practically constant if equation (3-56) is used, indicating that the mechanism which limits heat transfer in the boiling zone is hydrodynamic in nature, and that equation (3-56) is the correct choice for  $U_p$ .

If one accepts the view that the quench front velocity is controlled by vapor film instability at the quench front, then the effect of subcooling can be included (again, following Zuber's approach) by adding a term to equation (3-50):

$$q^* A_{HT} = \rho_v H_{gv} A_F (j_v + j_{vcond})$$
(3-60)

where  $j_{vcond} = vapor$  condensed by subcooled water. That is, more vapor can be generated at the quench front before instability sets in because some of the vapor is condensed.

# TABLE 3-1

EXPERIMENTAL FLUID AND ROD CONDITIONS AT QUENCH-CALCULATED VALUE OF C2 USING EQUATIONS (3-56) AND (3-57)

Test	Elevation	Void Fraction	۵ïq	٧q	Eq. (3-56)	Eq. (3-57)
	1	0.15	737	0.025	0.16	0.57
	2	0.40	711	0.022	0.16	0.53
15713	3	0.57	711	0.019	0.16	0.50
	4	0.68	728	0.020	0.20	0.58
	5	0.75	690	0.015	0.16	0.44
	1	0.16	658	0.018	0.14	0.48
	2	0.50	648	0.014	0.14	0.42
13914	3	0.72	631	0.012	0.15	0.40
	4	0.75	650	0.011	0.15	0.39
	5	0.80	654	0.009	0.14	0.34
13711	5	0.33	736	0.027	0.15	0.56

One form of equation (3-60) which accounts for subcooling is:

$$(\rho cA)_{rod} V_q (T_q - T_{sat}) = C_2 A_F \rho_v H_{Qv} U_r +$$
  
+  $C_3 (\rho c)_Q U_Q (T_{sat} - T_Q) A_F$  (3-61)

where

 $U_{\varrho}$  = liquid velocity  $T_{\varrho}$  = liquid temperature  $(\rho c)_{\varrho}$  = liquid heat capacity

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Figure 3-7. Effect of Pressure and Void Fraction on C<sub>2</sub> [Equation (3-59)]

C<sub>3</sub> is plotted in figure 3-8 versus liquid subcooling just upstream of the quench front for various FLECHT tests. It is apparent that the quench velocity increases considerably at high subcooling and that a transition in the quench front velocity coincides with the transition from laminar to turbulent flow conditions in the liquid.

Figure 3-8 also shows that a simple term like the second one on the right-hand side of equation (3-61) cannot adequately account for the effect of subcooling on the quench front velocity because at high quench front velocities the conduction within the rod becomes significantly two dimensional. A large portion of the stored energy within the rod is released several rod diameters away from the boiling zone. Rather than contributing to vapor flow in the film, this stored energy heats the liquid approaching the quench front. Thus, if equation (3-60) is to be used to determine the maximum heat transfer within the boiling zone, it must be used in conjunction with an appropriate twodimensional conduction model.

It is evident from figure 3-8 that different condensation models need to be developed, depending on the Reynolds number of the fluid. Two simple models are described in paragraphs 3-10 and 3-11 which account for vapor condensation at the boiling zone in laminar and turbulent flow.

3-10. Laminar Flow -- It is assumed that heat transfer from the vapor to the liquid occurs mainly by transient conduction as liquid moves by the boiling zone as shown in figure 3-9. The heat flow into the liquid is given by

$$q_{cond}^{*} = 2 (T_{sat} - T_{\ell}) \left| \frac{(\rho c k) \ell}{\pi \tau} \right|^{0.5}$$
 (3-62)

which represents the heat flux, averaged over time  $\tau$ , into a semi-infinite solid initially at T<sub>g</sub>, and the surface of which has been raised to T<sub>sat</sub>. Let

$$\tau = \frac{L}{U_r \varrho}$$
(3-63)



Figure 3-8. Effect of Subcooling FLECHT Tests on C<sub>3</sub> [Equation (3-61)]

14.380-13



Figure 3-9. Condensation Heat Transfer at Quench Front During Laminar Flow where

- L = characteristic length over which condensation heat transfer is effective in suppressing film instability
- $U_{r\ell}$  = velocity of liquid relative to boiling zone =  $U_{\ell} - V_{q}$

The total amount of vapor condensed at the boiling zone is

$$y_{\text{cond}} = q_{\text{cond}}^{\text{"}} \frac{P L}{A_{\text{F}} H_{\text{Qv}} P_{\text{v}}}$$
(3-64)

where P = rod perimeter.

0r

$$J_{vcond} = \frac{4 q_{cond}^{*} L}{D_{h} H_{gv} \rho_{v}}$$
(3-65)

because

$$q_{cond}^{"} = 2 \Delta T_{s} \left[ \frac{(\rho c)_{\varrho} k_{\varrho} U_{\varrho}}{\pi L} \right]^{0.5}$$
(3-66)

Where  $\Delta T_s = T_{sat} - T_{\varrho}$ 

$$J_{vcond} = \frac{8\Delta T_s}{D_h H_{Qv} \rho_v} \left[ \frac{(\rho c)_{Q} k_{Q} U_{Q} L}{\pi} \right]^{-0.5}$$
(3-67)

It is assumed that L can be scaled by the characteristic length of the unstable interface,  $\lambda$  [equation (3-53)]; then

$$J_{vcond} = \frac{4 C_4 \Delta T_s}{D_h H_{gv} \rho_v} \left[ (\rho c)_g k_g U_g \lambda \right]^{0.5}$$
(3-68)

where several of the constants have been lumped into  $C_A$ .

The important assumption contained in equation (3-68) is that  $j_{vcond}$  is proportional to  $\Delta T_s$  and to the square root of the liquid velocity during laminar flow.

3-11. <u>Turbulent flow</u> -- It is assumed that the condensation heat transfer in turbulent flow is given by

$$q_{cond}^{*} = h_{cond} \left( T_{sat} - T_{\varrho} \right)$$
(3-69)

where

$$h_{cond} = C_5 (\rho c U)_{2} Re_{2}^{-0.2}$$
 (3-70)

Here it has been assumed that  $j_{vcond}$  is proportional to  $\Delta T_s$  and to the liquid velocity to the 0.8 power.

Equations (3-68) and (3-70) will be judged adequate for determining the effect of subcooling at the quench front if, when the model is compared to data,  $C_4$  and  $C_5$  remain constant over a reasonably wide range of test conditions.

Equation (3-60) must now be combined with standard heat transfer correlations in such a way as to complete the heat transfer model for use with a two-dimensional axial conduction solution.

It must be recognized that the length over which the stored energy of the rod is released after quench may be significant. Thus, provision must be made for calculating changing fluid conditions as the fluid approaches the quench front.

# 3-12. Hydraulic Model Near the Quench Front

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The axial conduction model is presented in detail in paragraph 3-5. As described there, the axial position of several isotherms within the rod is calculated. Fluid and heat transfer conditions must be determined at the locations specified for the isotherms on the surface of rod. Because these locations are moving with time, the mass and energy balance equations must be developed from this point of view.

A control volume between two isotherm locations is shown in figure 3-10. The two axial surfaces of the volume are moving at two different velocities,  $V_1$  and  $V_{1-1}$ . (for all practical purposes, these velocities are small compared to the fluid velocity.) It is assumed that the mass velocity through the quench region is uniform axially ( $\partial G/\partial z = 0$ ), although the absolute value of G may change with time. It is also assumed that any vapor present is saturated. The change in volumetric flux, j, from  $z_{1-1}$  to  $z_1$  depends only on the net amount of vapor generated between  $z_{1-1}$  and  $z_1$ , if the component densities are constant [(paragraph 2-4, equation (2-15)].

$$\dot{p}_{1} = \rho_{V} J_{V1} + \rho_{Q} J_{Q1}$$
 (3-71)

and

1

1 = 1 + 1 + 1 21

Combining

 $G = \rho_0 J_1 - \Delta \rho J_{v1}$  (3-73)

(3-72)

Let

$$J_{1} = J_{1,1} + (a_{1} - b_{1} a_{1}) \Delta z_{1}$$
 (3-14)

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14.380-14



Figure 3-10. Control Volume Used to Calculate Fluid Conditions Near the Quench Front

where

a<sub>1</sub> = vapor generation rate due to boiling from the rod b<sub>1</sub>α<sub>1</sub> = vapor reduction rate due to concensation of bubbles in the bulk fluid

$$12_1 = 2_1 - 2_{1-1}$$

The terms a and b are defined later.

The vapor volumetric flux,  $j_{v1}$ , is given as a function of  $\alpha_1$  by [see paragraph 2-13, equation (2-58)]

$$J_{v1} = F_1 \alpha_1^n$$
 (3-75)

where  $F_1$  and n are given by equations (2-59) and (2-60) in paragraph 2-13.

Using equations (3-74) and (3-75) in equation (3-73) yields

$$G = \rho_0 J_{1-1} + \rho_0 a_1 \Delta z_1 - \rho_0 b_1 \alpha_1 \Delta z_1 - \Delta \rho F_1 \alpha_1$$
(3-76)

Equation (3-76) can be solved for  $\alpha_1$ .

Referring again to the control volume in figure 3-10, and taking an energy balance across the volume.

$$\frac{d}{dt} (M_{\varrho}, H_{\varrho}) = W_{\varrho 1-1} H_{\varrho 1-1} - W_{\varrho 1} H_{\varrho 1} + q_{\varrho}$$
(3-77)

where

 $M_{g}$  = mass of liquid in the volume  $\overline{H}_{g}$  = average liquid enthalpy in volume  $W_{g,1-1}, W_{g,1}$  = liquid mass flows across surfaces  $z_{1-1}, z_{1}$
$H_{21-1}, H_{21} =$  liquid enthalpy crossing surfaces  $z_{1-1}, z_1$ 

 $\bar{q}_{g}$  consists of two components: forced convection from wall to liquid and condensation of bubbles in liquid. The second component arises from the term  $b_{1}\alpha_{1}$  in equation (3-74). If the liquid becomes saturated,  $q_{g} = 0$  and all heat released from the rod is used to generate vapor.

Assume that  $\overline{H} = H_1$ ,  $\overline{\alpha} = a_1$ , and  $\overline{q}_2 = \overline{q}_1$  because

$$M_{g} = (1 - \alpha_{1})(z_{1} - z_{1-1}) A_{F} P_{g}$$
$$W_{g1} = P_{g} (U_{g1} - V_{1})(1 - \alpha_{1}) A_{F}$$
$$q_{g1} = q_{g1}^{'''} (z_{1} - z_{1-1}) A_{F}$$

Then

$$\frac{a}{dt} \left[ \rho_{\varrho} (1 - \alpha_{1})(z_{1} - z_{1-1}) H_{\varrho 1} \right] = \rho_{\varrho} (U_{\varrho 1-1} - V_{1-1})(1 - \alpha_{1-1}) H_{\varrho 1-1} - (3-78)$$

$$- \rho_{\varrho} (U_{\varrho 1} - V_{1})(1 - \alpha_{1}) H_{\varrho 1} + q_{\varrho 1} (z_{1} - z_{1-1})$$

Expanding the derivative and rearranging gives

$$\Delta z_{1}(1-\alpha_{1}) \quad \frac{dH_{Q1}}{dt} = (U_{Q1-1} - V_{1-1})(1-\alpha_{1-1})(H_{Q1-1} - H_{Q1}) + q_{Q1} \quad \frac{\Delta z_{1}}{\rho_{Q}} \quad (3-79)$$

Equation (3-79) can be integrated to obtain liquid enthalpy at each isotherm location.

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#### 3-13. Heat Transfer to the Liquid Near the Quench Front

Heat transfer from the rod at z, is given by

$$q_{1}^{*} = h_{1} \left( T_{u1} - T_{f1} \right)$$
(3-80)

where

 $h_1 = wall heat transfer coefficient at z_1 (paragraph 3-14)$  $<math>T_{w1} = wall temperature at z_1$  $T_{F1} = fluid temperature at z_1$ 

If  $T_{w1}$  is below the saturation temperature,  $q_1^*$  consists of a single-phase forced convection component:

 $q_{1}^{*} = q_{21}^{*} = h_{w2c} (T_{w1} - T_{21})$ 

where q"01 = heat flux to liquid and

$$q_{21} = \frac{4 q_{21}}{D_h}$$
 (3-81)

If  $T_{w1}$  is above the saturation temperature and the liquid is subcooled,  $h_1$  consists of a single-phase component and a nucleate boiling component.

$$q_{1}^{*} = h_{wlc} (T_{w1} - T_{l1}) + h_{wlnb} (T_{w1} - T_{sat})$$
 (3-82)

The nucleate boiling component of the heat transfer creates vapor at the wall, some of which enters the bulk fluid (paragraph 3-14), where it condenses and heats the liquid. The vapor is assumed to enter the bulk fluid in the form of bubbles of diameter  $D_b$ .<sup>(9)</sup>

$$0_{b} = 4.7 \left[ \frac{\sigma}{\Delta \rho g} \right]^{0.5}$$
(3-83)

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Relative velocity Urbb 1s<sup>(29)</sup>

$$U_{rbb} = 1.53 \left[ \frac{\sigma \Delta \rho q}{\rho_{g}^{2}} \right]^{0.25}$$
(3-84)

The heat transfer coefficient from the bubbles to the liquid is given by (30)

$$h_{bb} = \left[\frac{(\rho c \ k)_{\ell} \ U_{rbb}}{\pi \ 0_{b} \ P_{r}^{0.33}}\right]^{0.5}$$
(3-85)

The terms a and b in equation (3-74) can now be defined.  $a_1$  is the vapor generation rate/unit volume:

$$a_{1} = \frac{4 q'' nbb}{D_{h} H_{gv} \rho_{v}}$$
(3-86)

where  $q_{nbb}^{*}$  = the portion of nucleate boiling heat transfer supplying vapor to the bulk fluid (pargraph 3-14).  $b_1 \alpha_1$  is the vapor condensation rate/unit volume:

$$b_{1}\alpha_{1} = \frac{h_{bb} (T_{sat} - T_{l}) 6\alpha_{1}}{D_{v} \rho_{v} H_{lv}}$$
(3-87)

where

$$\frac{6 \alpha_1}{D_b}$$
 = bubble surface area/unit volume

Finally, in terms of the quantities above, the total heat flow to the liquid is

$$q_{l1} = h_{wlc} (T_{w1} - T_{l1}) 4/D_h + h_{bb} (T_{sat} - T_{l1}) 6\alpha/D_b$$
 (3-88)

### 3-14. Heat Transfer From Rod Surface to Fluid Near the Quench Front

Figure 3-6 shows the two heat transfer regions that are assumed to exist along the rod near the quench front. Region I is assumed to exist when heat fluxes are low (comparable to those caused by internal heat generation alone). If the wall temperature is below the saturation temperature, the Dittus-Boelter equation is used to calculate heat transfer (table 3-2). If the wall temperature is above the saturation temperature, a nucleate boiling component is added using a correlation developed by Rohsenow<sup>(8)</sup> and table 3-2.

In region II, it is assumed that the vapor generated on the rod begins to take on the character of a vapor film at a heat flux intermediate between the low range of region I and that normally encountered at dryout. An approximate boundary between region I and region II is obtained as described below.

In region I, vapor is released to the bulk fluid in the form of bubbles. These bubbles have a relative velocity with respect to the liquid,  $U_{rbb}$ . As the heat flux increases and a vapor film is formed, bubbles are released from the vapor film at a rate determined by the geometry of the film and the velocity of the bubbles (figure 3-2). That is, the vapor flux, averaged over the rod surface, to the bulk fluid is approximately:

$$\mathbf{j}_{\mathbf{vb}} = \mathbf{U}_{\mathbf{rbb}} \, \mathbf{\alpha}_{\mathbf{bb}} \tag{3-89}$$

where  $\alpha_{bb}$  is determined by assuming that bubbles of diameter D are released in an area enclosed by  $\lambda^2/2$ .<sup>(9)</sup>

$$\alpha_{bb} = \frac{\pi 0_b^2}{2\lambda^2}$$
(3-90)

Equation (3-89) is similar to that derived by Zuber in his analysis of the minimum film boiling heat flux, except that the multiplicative constants are different. Equation (3-87) determines only how much vapor will reach the bulk fluid, however, The total vapor generated at the wall depends on the nucleate boiling or transition boiling coefficients used.

	TABLE 3-2							
HEAT	TRANSFER	CORREI	ATTONS	USED	NEAR	OHENCH	FRONT	

Region	Heat Transfer Mechanism	Correlation	Reference No.
I	Single-phase	$Nu = 0.023 \text{ Re}^{0.8} \text{ Pr}^{0.4} \text{ Re} > 2000$	
	convection	Nu = 10 Re < 2000	7
п	Nucleate boiling	$h_{NB} = \left[\frac{C_{PQ} \Delta T}{0.013 H_{QV}}\right]^3 \frac{\mu_{Q} H_{QV}}{Pr^{5.1}}$	
		where L = $\left  \frac{\sigma}{\Delta \rho g} \right ^{0.5}$	8
	Nuclear or transition boiling	Minimum of h <sub>NB</sub> above and:	
		$h = q_{max}^{"} c^{(1 - B\Delta T)} B$	
		where:	
		$q_{max}'' = 0.15 \rho_v H_{\xi v} \left[ \frac{\sigma \Delta \rho q}{\rho g^2} \right]^{0.25}$	21
IV	Film boiling	$h_{fb} = 0.62 \left[ \frac{K_v^3 H_{\xi v} \rho_v \Delta \rho q}{\mu_v \Delta T L} \right] 0.25$	9

The remaining vapor generated at the wall is assumed to remain close to the rod surface and to contribute to the film instability further downstream.

Equations (3-83) and (3-84) are used to calculate D, and U, in region II.

The heat transfer coefficient in region II is taken to be the minimum between that given by the Rohsenow correlation and by a modified transition boiling equation which is developed in the next topic.

In region III, heat fluxes are approaching those normally encountered during dryout. The total amount of vapor generated at the quench front is controlled by the stability of the vapor film formed just above the quench front. One would expect that the heat transfer in region III would follow, more or less, a boiling curve to the minimum film boiling temperature, as shown in figure 3-11. In addition, this region should be the most sensitive to changes in the stability of the vapor film. If the stability is destroyed because of an overabundance of vapor generation, region III will be most affected by the resulting accumulations of vapor.

Although the details of the heat transfer are not known in this region, only the total vapor flow in the film is required. Thus, the heat transfer in this region can be specified as some arbitrary function of the wall temperature, adjusted to provide the total vapor flow determined by equation (3-60).

A reasonable form of the heat flux in region III is

$$q^{*} = Ae^{-B\Delta T} \Delta T \qquad (3-91)$$

which is the form used in several transition boiling correlations. (31,32) The maximum heat flux occurs where

$$\frac{dq^{"}}{d\Delta T} = 0 = -ABe^{-B\Delta T} \Delta T + Ae^{-B\Delta T} = Ae^{-B\Delta T} (1 - B\Delta T)$$
(3-92)

If the maximum heat flux is denoted by  $q^{*}_{max}$ , then because the maximum occurs at  $\Delta T = 1/B$ ,

Therefore

$$A = q^{u}_{max} Be$$
 (3-94

Equation (3-91) becomes

$$q'' = q''_{max} e^{(1 - B\Delta T)} B\Delta T$$
 (3-95)

The Zuber equation for critical heat  $flux^{(21)}$  is used to calculate  $q^{max}$  (table 3-2).

B is a parameter which is allowed to vary to control the total amount of vapor generated at the quench front: total vapor generated at quench front = Q.

$$Q = \frac{P}{\rho_v H_{Qv}} \int_{z_1}^{z_{\text{III}}} q^{\text{"d}z}$$
(3-96)

where

P = rod perimeter

z = dividing line between region I and region II (vapor film begins here)

 $z_{111}$  = point where heat flux approaches minimum film boiling heat flux

If Q calculated from equation (3-96) is less than that allowed by equation (3-60), B is adjusted upward. If Q exceeds the maximum allowable, B is adjusted downward. Effectively this is equivalent to varying the wall superheat at which the maximum heat flux (and consequently the minimum film boiling heat flux) occurs, as shown in figure 3-11.

90h 350 300 250 B = 0.01 ΔT (°F) 200 150 B = 0.028 20 0 0 0.2 1.0 0.6 0.4 0.8 <sup>w</sup>b/ "b

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Figure 3-11. Transition Boiling Heat Flux [Equation (3-95)]

The boundary between region IV and region III is the point where the heat flux calculated from equation (3-95) falls below a value representative of normal film boiling. The film boiling correlation used is that of Berenson<sup>(9)</sup> and table 3-2.

### 3-15. Summary of the Model

The preceding paragraphs have detailed the underlying assumptions and equations which make up the quench front heat transfer model. The important features are now summarized to focus the model more sharply. The quench front has been divided into four regions, each defined by a distinct set of fluid and thermal conditions (figure 3-6).

In region I, the wall temperature is near the fluid temperature. Heat flow from the rod is dominated by power generation and by one-dimensional radial conduction of energy stored in the interior of the rod. Because the axial dimension of this region extends to several rod diameters, changes in fluid conditions due to heating are explicitly calculated. Single-phase forced convection and nucleate boiling are possible heat transfer regimes in this region, and standard heat transfer correlations are used (table 3-2).

The wall temperature in regions II and III (boiling zone) changes from a value near the saturation temperature to one above the Liedenfrost temperature. The boundary between region I and region II is defined as the point where vapor generated at the wall begins to form a vapor film. This is assumed to occur at a heat flux controlled by bubble formation at the vapor film [equation (3-89)]. Heat flow from the rod is dominated by radial and axial conduction of stored energy.

Because the axial length of the boiling zone is extremely narrow, the local heat transfer coefficient within the zone is unknown. However, the maximum amount of vapor generated within this region, and thus the overall heat transfer, is a function only of the fluid conditions. All that must be known about the local heat transfer within this region is its order of magnitude, and this is known to be comparable to nucleate boiling heat transfer near the

critical heat flux. The heat transfer is assumed to behave exponentially with wall temperature, in a manner analogous to the transition boiling regime on a surface with uniform temperature. Coefficients in the exponential function are adjusted such that the total vapor generated within the zone is consistent with the hydrodynamic limit.

A basic assumption underlying the heat transfer model in the boiling zone is that the details of the heat flow occurring within the zone are unimportant in determining the overall rod behavior, as long as the total heat transfer is well defined. Intuitively this is to be expected because even large oscillations in wall temperature will be lost in the overall change from Liedenfrost temperature to saturation temperature over a very short distance. Thermal diffusion within the rod is also expected to reduce the importance of the local behavior of the surface temperature.

Region IV represents the rod prior to rewetting, where the heat transfer is dominated by film boiling. The boundary between region III and region IV is the minimum film boiling heat flux, which occurs at a temperature which depends on the overall heat transfer in region III.

The models described above deal with conditions on the surface of the rod and in the flow channel. Coupled to these models is a conduction model which describes the heat transfer occurring within the rod as a function of surface heat flux, power generation, and rod material properties. This model is described in paragraph 3-5.

# 3-16. Comparison of Quench Model With FLECHT Experiments

A computer code was developed which combined the quench heat transfer model and the conduction model so that necessary constants could be determined and predictions made. Table 3-3 lists the constants that were required and the FLECHT tests which were used to determine them.

## TABLE 3-3 EMPIRICAL CONSTANTS REQUIRED FOR QUENCH HEAT RELEASE MODEL

Constant	Source	Tests Used to Determine	Value
ar	Equation (3-58)	15713, 13914	[] <sup>a,c</sup>
C,	Equation (3-68)	13404	
C <sub>5</sub>	Equation (3-70)	12102	Ĺ

The isotherm configuration used for the quench velocity predictions is shown in figure 3-12. The position of the first isotherm was the test section inlet, and fluid conditions at this position were the inlet conditions for the specific test being investigated. The last isotherm was set at a value corresponding to the temperature of the rod just before quench, at a particular test section elevation.

The remaining isotherms were arranged initially as shown in figure 3-12. The calculation was allowed to proceed until a steady state was reached, when typical isotherm positions were as shown in figure 3-13. The three constants listed in table 3-3 were determined by performing several calculations until agreement with data was obtained. Comparisons were then made with several additional FLECHT skewed tests covering a wide range of inlet conditions. Figure 3-14 shows that agreement between data and model predictions is good over the entire range of inlet conditions.

#### 3-17. QUENCH HEAT RELEASE MODEL

#### 3-18. Initial Conditions

The initial position of the isotherms in the axial conduction model, relative to the positions of the fluid nodes in BART, is shown in figure 3-15.



TEST SECTION INLET ---- FIRST ISOTHERM

Figure 3-12. Axial Conduction Isotherm Initial Locations to Calculate Quench Velocity at 3 Feet





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Figure 3-14. Predicted Versus Measured (FLECNT Skewed Tests) Quench Velocity



Figure 3-15. Axial Conduction Isotherm Locations at Time = 0

Isotherms with values below the saturation temperature are spaced linearly below the heated rod bundle (a short, adiabatic entrance region used in BART as shown in figure 3-15). Isotherms with values between the saturation temperature and the minimum film boiling temperature (typically around 500°F) are spaced along a narrow region (0.25 inch) and remaining isotherms are located in the appropriate positions along the rod bundle. The last isotherm is placed at the first available BART fluid node, as shown in figure 3-15.

#### 3-19. Transtent Heat Release

At any specific time, the location of the axial conduction isotherms (relative to the BART fluid nodes) is as shown in figure 3-16. The quench front location is defined as the location of the isotherm with a value near the minimum film boiling temperature. This isotherm is specified by input and is always located within the steep temperature gradient shown in figure 3-16.

The first isotherm, which has the same value as the inlet fluid temperature, remains at the location of the lowest BART fluid node. The last isotherm is moved to the next BART fluid node and is set to the rod surface temperature at that node, when the N-1 isotherm overtakes the current BART fluid node (figure 3-16).

Figure 3-17 illustrates how the 1-D BART mesh and the 2-D isotherm mesh are coupled. The BART quench region is defined as that region having a length equal to one BART node. Within this region, only fluid conditions are calculated. The total heat flow from the two nodes just below the quench front (the leading edge of the quench front) is equal to the total heat flow calculated by integrating the heat fluxes determined from the isotherm mesh. The heat flow is apportioned between the two BART nodes to effect a smooth transition from node to node as the quench front advances (the triangular profile shown in the figure allows this transition to occur).

Outside the quench region, fluid conditions and heat transfer coefficients at each isotherm location are interpolated BART values. Inside the quench region, local fluid conditions and surface heat fluxes are calculated for each isotherm. The heat flux is then integrated to give the total heat release to be used in the BART calculation.

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Release at Time Greater Than Zero



Figure 3-17. Coupling Between BAR1 and Quench Front Models

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## SECTION 4 CODE VERIFICATION

In section 2, the development of BART fluid and heat transfer models is discussed. Some empirical parameters were determined by comparing BART predictions with selected data. In section 3, the development and verification of the two-dimensional axial conduction model was detailed. In this section, the verification of the complete BART code using data from FLECHT, (25,26) G-2, (33) Semiscale, (34, 35) and FLECHT SEASET(36) reflood tests is explained.

#### 4-1. BOUNDARY AND INITIAL CONDITIONS

The following boundary and initial conditions are required for BART:

- o Flooding rate
- o Inlet enthalpy
- o Power decay rate
- o Pressure
- o Housing quench heat release
- o Initial housing and rod temperature distribution
- o Axial distribution of gap heat transfer coefficient
- o Rod power distribution

Most of these conditions were determined directly from the data of the various tests. Some analysis was required in two areas: inlet fluid temperature and housing heat release.

In all the experiments being considered here, the test section lower plenum was filled with water sometime prior to the beginning of the test. The lower grid thermocouples in the Semiscale tests, which measured the inlet fluid temperature directly, showed a delay before the inlet fluid reached the prescribed temperature. It has been assumed in the following FLECHT and G-2 comparisons that a similar delay, equivalent to the time it would take to sweep out the water volume just below the heated length, also occurred.

Housing quench heat release is important in the Semiscale and FLECHT cosine tests, which used thick housing walls. In the FLECHT test it was found that the housing quench front coincided with the rod quench front. In the Semiscale tests, a quench front curve was constructed using the housing thermocouple data, and was input to BART. Similar quench front curves were constructed for the FLECHT Skewed and G-2 tests.

Figures 4-1 through 4-5 show the power shape in each experiment and the BART noding scheme used in the predictions. For the FLECHI SLASET tests, the initial fluid conditions used are also shown.

#### 4-2. VERIFICATION TEST MATRIX

Table 4-1 lists the conditions in the various experiments used to verify BART. The basis for the test matrix was to choose tests whose conditions bracketed several key quantities in the PWR calculations. Additional tests from the FLECHI SEASET program were added for more coverage.

Table 4-2 summarizes several key parameters and their ranges in current PWR analysis. The test matrix adequately covers this range.

In addition, the following items were compared:

- o Power shape (figures 4-6 to 4-8)
- o Peak power/inlet velocity (figures 4-9 and 4-10)

In every case the tests used in section 4 bracket the range of reflood conditions in PWR calculations.

### 4-3. DATA COMPARISONS

Figures 4-11 through 4-15 show comparisons for FLECHT Cosine test 5132. Figure 4-11 shows the cladding temperature at 4, 6, 8, and 10 feet, and figure 4-12 shows the heat transfer coefficient at the same locations. The data depicted here are averages consisting of all the 1.0 power rods which are at least one row away from the housing and from dead rods.

# TABLE 4-1

## TESIS USED FOR BART VERIFICATION

Test Sertes	Test No.	Flood Rate (1n./s)	Power (kw/ft)	Inlet Subcool (°F)	Pressure (psia)	Initial Temp ( <u>°F)</u>
	5132	1.0	0.95	140	40	1600
	6638	0.80	0.95	140	20	1600
FLECHT	4831	1.5	0.95	140	40	1600
Cosine <sup>(25)</sup>	7934	0.6	0.95	140	40	1600
	5342	8.0	0.95	20	40	1600
	13303	. 1.5	0.7	140	40	1600
	13404	1.0	0.7	140	40	1600
FLECHT	13609	1.0	0.7	140	20	1600
Skewed <sup>(26)</sup>	11618	1.5	0.45	140	40	1600
	15305	0.8	0.7	140	40	1600
	15713	1.0	0.7	2	40	1600
	16945	6.7 (5 sec)	0.7	121-265	40	1600
Westinghouse	538	1.3	0.73	110	40	1400
G-2 <sup>(33)</sup>	561	1.0	0.55	120	20	1400
	\$-03-3	0.6	0.7	140	60	1400
Semiscale	S 03-a	1.0	0.1	130	21	1400
Mod 1 (34,35)	S 03-b	1.0	1.2	140	60	1400
	S 03-c	1.0	0.1	140	50	1100
	5 03-d	1.0	0.7	40	60	1400
FLECHT	32235	h.4 (<5s)	0.7	125	40	1600
SEASEI		0.8 (>5s)				
161-Rod	32333	6.5 (<5s)				
Cosine <sup>(36)</sup>		0.98 (<200s	) 0.7	88	20	1600
		0.62 (>2005	1			

TABLE 4-2

COMPARISON OF REFLOOD TEST RANGE AND CALCULATED PWR RANGE

Parameter	PWR Range	Test Range
Initial temperature	13001600°F	11001600
Pressure	2060 psia	2060
Initial power	0.7-0.8 kw/ft	0.45 -1.2
Inlet subcooling	12010°F	14020°F

The usual method for comparing heat transfer calculations to FLECHT data is to base the heat transfer coefficient on the saturation temperature; that is,

 $h = \frac{q^{*}}{T_{u} - T_{cat}}$ 

where q" = local heat flux.

During some portions of the reflood transient, steam flowing past the maximum temperature location (usually 6 feet) will increase significantly in temperature. This superheated steam then flows past cooler locations at upper elevations. The heat flux, and consequently the h as defined above, are negative during these times. Figure 4-13 presents a snapshot of fluid and rod conditions at 100 seconds in the test. Figure 4-14 compares predicted and measured bundle mass, and figure 4-15 compares quench front elevations. The same group of figures is presented for each cosine test listed in table 4-1 (figure 4-16 through 4-35).

Figures 4-36 through 4-70 show prediction and data comparisons for the FLECHT Skewed test series. These illustrations are arranged in the same format as are the cosine series of illustrations.

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G-2 tests are shown in figures 4-71 through 4-76. Figure 4-71 compares predicted and measured cladding temperature at the 3-, 4-, 6-, and 8-foot elevations for test 538. The heat transfer coefficient is examined in figure 4-72. Quench front elevation is compared in figure 4-73. Similar figures are presented for test 561 (figures 4-74 through 4-76). Note that these comparisions are presented in slightly different format.

Semiscale tests are shown in figures 4-77 through 4-80. The 14-, 29-, and 39- inch locations are used for rod temperature comparisons.

FLECHT SEASET data is shown in figures 4-81 through 4-84. Two stepped flooding rate tests from the FLECHT SEASET program were compared with BART. BART predictions were compared with data from the following rods at elevations where the data was available (refer to figure 4-104 for locations of these rods): 6L, 6K, 7J, 8K, 9K, 11E. These comparisons are presented in figure 4-81 for test 32333.

A more representative comparison is shown in figure 4-82 where the BART prediction is compared to the mean temperature of all rods more than two rows away from the test housing and dead rods. Similar comparisons are shown for test 32235 (figures 4-83 and 4-84).

## 4-4. HEAT TRANSFER COEFFICIENT COMPARISONS -- FLECHT COSINE SERIES

Additional comparisons were made between BART predicted heat transfer coefficients and data. The data presented (in figures 4-85 to 4-88) consists of the average of all rods at least one row away from the housing or dead rods and includes  $\pm 1\sigma$  uncertainty bands. The heat transfer coefficient is defined in paragraph 4-3.

#### 4-5. ADDITIONAL COMPARISONS WITH FLECHT SKEWED DATA

To assure that BART properly accounts for variations in power shape, a more detailed comparison was made with several FLECHT Skewed tests. <sup>(26)</sup> The following tests were analyzed:

13303	Figures	4-89	and	4-90
13404	Figures	4-91	and	4-92
11618	Figures	4-93	and	4-94
15305	Figures	4-95	and	4-96
15713	Figures	4-97	and	4-98
16945	Figures	4-99	and	4-100

The comparisons, including  $\pm$  lo uncertainty bands, present time histories of the local heat transfer coefficient and the cladding temperature at the 6-foot, 8-foot and 10-foot elevations for each test. In all cases the BART prediction is depicted by the dashed line. In general, agreement is good, which indicates that BART can account for variations in power shape.

#### 4-6. HEAT TRANSFER COEFFICIENT COMPARISONS --FLECHT SEASET TESTS

Additional comparisons were made between BART predicted heat transfer coefficients and data. The data presented in figure 4-101 and 4-102 consists of the average of all rods at least one row away from the housing or dead rods and includes  $\pm$  lo uncertainty bands. The heat transfer coefficient is defined in paragraph 4-3.

### 4-7. BART CLADDING TEMPERATURE RISE

The BART predicted cladding temperature rise was compared with several sets of experimental data as shown below:

Α.	FLECHT-SEASET 161-rod tests:	31805
	(Thermocouples between 5.6 and 8.0 feet)	32235 32333
8.	G-2 tests:	538
	(Thermocouples between 5.8 and 7.9 reet)	201
C	FLECHT Skewed power tests:	11618
	(Thermocouples between 8.0 and 10.0 feet)	13303
	(	13404
		13609
		15305
		15713
		16945
0	FLECHT Costne nover tests.	4831
U .	(Thermocouples between 6.0 and 8.0 feet)	5132
	(Thermocoupies becaeen oto and oto rece)	5342
		6638
		7934

A, B, and C are uniform radial power profile tests. The comparisons of BART calculations to data are presented as a composite in figure 4-103.

FLECH: Cosine tests are nonuniform radial power tests and were mentioned to be of particular interest to the NRC regarding BART validation. Cosine tests were compared on a different basis than were the other tests. The method of comparison is described below, and cosine comparisons are identified apart from the other tests comparisons in figure 4-103.

Capture Fractions:

а.	Skew/G-2/FLECHT-SEASET composite	90%
b.	Low flooding rate cosine tests:	
	(per evaluation model procedures)	80%
с.	All-test composite	88%

### FLECHT-SEASET 161-rod tests:

31805 32335 32333

Data from these tests for comparison to BART predictions were taken from the following rods:

levation	Heater kod Locations
5'7"	6J, 6L, 9E, 9I, HI, 11K
5'10"	9K, 10H, 8E, 7E, 9I, 10G
6'0"	7J, 8H, 9I, 9L, 10J, 9C
6'2"	11K, 121
6'4"	11G, 6I, 7K, 11K, 12K
6'6"	10D, 7D, 8H, 8K, 10J, 12I
7'0"	70, 9C, 9L, 10J, 11E, 12I
7'6"	7K, 7M, 9C, 9L, 10J, 121
8'0"	7D, 7M, 8H, 9L, 10J, 12I

These locations reflect an inner channel of rods which are at least three rows back from the housing and more than one row away from failed heater rods. The actual thermocouples used at various elevations in the three tests varied according to thermocouple failure. Figure 4-104 shows the inner channel used for FLECHT-SEASET validation runs.

In a similar manner, inner channels were defined for G-2, FLECHT Skewed, and FLECHT Low Flooding Rate Cosine Tests. Figures 4-105, 4-106, and 4-107 show the rods within the inner channels of these three tests bundles. Instrumented rods are marked, and all available thermocouple data were used to each elevation for comparisons of BART to data in uniform radial power tests.

The low flooding rate cosine tests were compared on a different basis to reflect their nonuniform radial power profile. The procedure was as follows:

- The inner channel was defined.
   (Two rows adjacent to walls and one row of rods around failed rod #7F were excluded.)
- (2) Power profiles and initial temperature profiles were obtained for:(1) An average rod within the inner channel

(2) A hot rod (1.1 power factor)

(Average rod properties were taken from the average of all

instrumented rods in the inner channel. Hot rod properties were taken from the average of instrumented 1.1 power rods only, within the inner channel.)

- (3) BART was run as it will be applied in the ECCS Evaluation Model. That is, average fluid conditions for average rods in the bundle were input, and average bundle heat transfer coefficients were calculated. Then, the average fluid conditions and average bundle heat transfer coefficients were supplied to a separate hot rod model.
- (4) The output of the hot rod calculations using average fluid conditions was used to calculate hot rod temperature rise for comparison to test data (figure 4-107).

BART NODES 22 9 10 11 12 13 14 15 16 17 18 19 20 21 3 4 5 6 7 8 • ... • . . . . . - THERMOCOUPLES - GRIDS 1.8 1.66 60 1.6 1.4 AXIAL PEAKING FACTOR 1.2 1.0 0.8 0.6 0.48 0.4 0.2 0 10 12 6 7 9 14 5 8 0 2 3 4 1 Z (FT)



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Figure 4-2. FLECHT Skewed Power Distribution, Thermocouple and Grid Locations, and BART Noding

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Figure 4-5. FLECHT SEASE1 BART Noding and Initial Conditions

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Figure 4-6. Comparison of FLECHT Cosine and FLECHT SEASET Power Shapes With PWR Calculated Range





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Figure 4-8. Comparison of FLECHT Skewed Test Power Shapes With PWR Calculated Range



Figure 4-9. Peak Linear Power Density Divided by Inlet Velocity: Test Range Versus PWR Range During Reflood (FLECHT Cosine, FLECHT Skewed, and G-2 Tests)


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Figure 4-11. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Cosine Test 5132



Figure 4-12. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Cosine Test 5132



Figure 4-13. Data Comparisons of Bundle Conditions at T = 100 Seconds for FLECHT Cosine Test 5132



Figure 4-14. Comparison of Predicted and Measured Bundle Mass for FLECHT Cosine Test 5132



Figure 4-15. Comparison of Quench Front Elevations for FLECHT Cosine Test 5132



Figure 4-16. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Cosine Test 6638



Figure 4-17. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Cosine Test 6638







Figure 4-19. Comparison of Predicted and Measured Bundle Mass for FLECHT Cosine Test 6638



Figure 4-20. Comparison of Quench Front Elevations for FLECHT Cosine Test 6638



Figure 4-21. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Cosine Test 4831



Figure 4-22. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Cosine Test 4831







Figure 4-24. Comparison of Predicted and Measured Bundle Mass for FLECHT Cosine Test 4831



Figure 4-25. Comparison of Quench Front Elevations for FLECHT Cosine Test 4831



Figure 4-26. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Cosine Test 7934



Figure 4-27. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Cosine Test 7934







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Figure 4-31. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Cosine Test 5342



Figure 4-32. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Cosine Test 5342



Figure 4-33. Data Comparisons of Bundle Conditions at T = 100 Seconds for FLECHT Cosine Test 5342



Figure 4-34. Comparison of Predicted and Measured Bundle Mass for FLECHT Cosine Test 5342



Figure 4-35. Comparison of Quench Front Elevations for FLECHT Cosine Test 5342



Figure 4-36. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 13303



Figure 4-37. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 13303







Figure 4-39. Comparison of Predicted and Measured Bundle Mass for FLECHT Skewed Test 13303



Figure 4-40. Comparison of Quench Front Elevations for FLECHT Skewed Test 13303



Figure 4-4:. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 13404



Figure 4-42. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 13404







Figure 4-44. Comparison of Predicted and Measured Bundle Mass for FLECHT Skewed Test 13404



Figure 4-45. Comparison of Quench Front Elevations for FLECHT Skewed Test 13404



Figure 4-46. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 13609



Figure 4-47. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 13609



Figure 4-48. Date Comparisons of Bundle Conditions at T = 100 Seconds for FLECHT Skinsd Test 13609

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Figure 4-49. Comparison of Predicted and Measured Bundle Mass for FLECHT Skewed Test 13609



Figure 4-50. Comparison of Quench Front Elevations for FLECHT Skewed Test 13609



Figure 4-51. Gata Comparisons of Cladding Temperature at Various Clevilions for FLECHT Skewed Test 11618



Figure 4-52. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 11618



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Figure 4-54. Comparison of Predicted and Measured Bundle Mass for FLECHT Skewed Test 11618



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Figure 4-55. Comparison of Quench Front Elevations for FLECHT Skewed Test 11618



Figure 4-56. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 15305



Figure 4-57. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 15305







Figure 4-59. Comparison of Predicted and Measured Bundle Mass for FLECHT Skewed Test 15305



Figure 4-60. Comparison of Quench Front Elevations for FLECHT Skewed Test 15305



Figure 4-61. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 15713



Figure 4-62. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 15713







Figure 4-64. Comparison of Predicted and Measured Bundle Mass for FLECHT Skewed Test 15713



Figure 4-65. Comparison of Quench Front Elevations for FLECHT Skewed Test 15713



Figure 4-66. Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 16945



Figure 4-67. Data Comparisons of Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 16945



Figure 4-68. Data Comparisons of Bundle Conditions at T = 100 Seconds for FLECHT Skewed Test 16945



Figure 4-69. Comparison of Predicted and Measured Bundle Mass for FLECHT Skewed Test 16945



Figure 4-70. Comparison of Quench Front Elevations for FLECHT Skewed Test 16945



Figure 4-71. Data Comparisons of Cladding Temperature at Various Elevations for G-2 17x17 Test 538



Figure 4-72. Data Comparisons of Heat Transfer Coefficients at Various Elevations for G-2 17x17 Test 538



Figure 4-73. Comparison of Quench Front Elevations for G-2 17x17 Test 538



Figure 4-74. Data Comparisons of Cladding Temperature at Various Elevations for G-2 17x17 Test 561



Figure 4-75. Data Comparisons of Heat Transfer Coefficients at Various Elevations for G-2 17x17 Test 561



Figure 4-76. Comparison of Quench Front Elevations for G-2 17x17 Test 561





Figure 4-77. Data Comparisons of Cladding Temperature at Various Elevations for Semiscale Test S-03-3



Figure 4-78. Data Comparisons of Cladding Temperature at Various Elevations for Semiscale Test S-03-A



Figure 4-79. Data Comparisons of Cladding Temperature at Various Elevations for Semiscale Test S-03-B



Figure 4-80. Data Comparisons of Cladding Temperature at Various Elevations for Semiscale Test S-03-D



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Figure 4-81. Comparison of CAN: Predictions and Cladding Temperature Data Pur Selected Rods in FLECHI SEASET Test 32333



Figure 4-82. Comparison of BART Prediction and Mean Temperature of All Rods More Than Two Rows Away From Tesi Housing in FLECHT SEASET Test 32333





Figure 4-83. Comparison of BART Predictions and Cladding Temperature Data for Selected Rods in FLECHT SEASET Test 32235



Figure 4-84. Comparison of BART Prediction and Mean Temperature of All Rods More Than Two Rows Away From Test Housing in FLECHT SEASET Test 32235



-A-A- FLECHT CORRELATION



Figure 4-85. Comparisons of BART Predicted Heat Transfer Coefficients and Data for FLECHT Cosine Test 4831





Figure 4-86. Comparisons of BART Predicted Heat Transfer Coefficients and Data for FLECHT Cosine Test 5342





Figure 4-87. Comparisons of BART Predicted Heat Transfer Coefficients and Data for FLECHT Cosine Test 6638





Figure 4-88. Comparisons of BART Predicted Heat Transfer Coefficients and Data for FLECHT Cosine Test 7934



--- BART DATA AVERAGE, ±10



Figure 4-89. Detailed Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 13303





Figure 4-90.

1-90. Data Comparisons of Time Histories of Local Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 13303





Figure 4-91. Detailed Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 13404





Figure 4-92. Data Comparisons of Time Histories of Local Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 13404





Figure 4-93. Detailed Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 11618




Figure 4-94. Data Comparisons of Time Histories of Local Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 11618





Figure 4-95. Detailed Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 15305





Figure 4-96. Data Comparisons of Time Histories of Local Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 15305





Figure 4-97. Detailed Data Comparisons of Cladding Temperature at Various Elevations for FLECHT Skewed Test 15713





Figure 4-98. Data Comparisons of Time Histories of Local Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 15713











Figure 4-100. Data Comparisons of Time Histories of Local Heat Transfer Coefficients at Various Elevations for FLECHT Skewed Test 16945



--- BART DATA AVERAGE, ±10



Figure 4-101. Comparison of BART Prediction and Average Heat Transfer Coefficients of All Rods More Than Two Rows Away From Test Housing in FLECHT SEASET Test 32333





Figure 4-102. Comparison of BART Prediction and Average Heat Transfer Coefficients of All Rods More Than Two Rows Away From Test Housing in FLECHT SEASET Test 32235

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Figure 4-105. Cross Section of G-2 Loop Rod Bundle and Baffle, Showing Bundle Instrumentation

ROD BB WITH BONJER SIBILACES AT 1/C 10.411.0MS j. #. 10. 11 5. ELEVATIONS . ANC LEVEL IRANSOUCES \$2..... FIEVAIION 4. 8. 10. 11.5. EIEVALIONS INSTRUMENTED RUD WITH N 1/C'S AT 2' 10' H' HIEVATIONS 145180968160 800 MIN 8 1/C.2 81 3. 1. 4. 6. 9. 11 5. HEARING INSTRUMENTED ROOS WITH 5 1/C 5 41 IN 2. 3/1 E NIIM OOD OHM BADBISMI 11. 616 vA110mS 11. 616 vA110mS 11. 616 vA110mS 201184313 .01 RADIAL POMER DISTRIBUTION SIE MH PROBE 0 .01 . 31841K1 ..... . -0 -1.1. 1.0. LEGIMO \* -5 . • \* --INNER CHANNEL -\$ . : \* . 8 -..... = 2 £ . : \* -: .: -. ... ..... ... . : 8.5 2 • ? 5. ; £ · 2 2.2 --1.0 . . 1 .... • -... -------• • == . . .. 1. -----. . 2.3 £ . 3 •: . . . • : -÷ . : : : 1 .: ~ 1.4 • ] 6.3 • • 6 4 • • : . . \*\* . : ° - : • ... . ... 12 1. (... : • : s.: s . : 1/2 4-3 : 12 • 5 . . . 1 :+: ----. ·• : ÷ ... ... ... \$ 7 -• - 2 7 171 . : : ÷. . 2 101 111 ŝ 2 . • \* -11+ : : -. . . -÷., .. 11.1 - 1 ~ -- 5 . -• . AH + É . . : . . . -2 -

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Rod Bundle Instrumentation for FLECHI Low Flooding Rate Tests (Cosine Test Series) f 1gure 4-107.

## SECTION 5

## PROPOSED PWR ECCS EVALUATION MODEL USING BART

Several changes to the current evaluation model<sup>(37)</sup> are required to incorporate the BARI core heat transfer models into the analysis. These changes affect the way the design calculation is carried out, and affect some of the models used.

## 5-1. ECCS ANALYSIS PROCEDURE

The purpose of the BART code is to improve the reflood portion of the ECCS evaluation model by replacing several correlations and simple thermalhydraulic models with a more integrated, mechanistic model. Alterations were made to the linkages and data transfers among ECCS model codes, to effectively use the improvements available in BART.

BART and LOCTA are linked through the following:

 A one-time data transfer from LOCTA to BART at the beginning of core (BOC) recovery 1

- A one-way transfer of local cladding heat transfer coefficients from a BARI output tape to LOCTA at each timestep
- A conditional one-time data transfer from LOCTA to BART at time of burst (BURST) if it occurs after BOC

BART and LOCTA do not interact in a timestep-by-timestep manner. They interact only through the sequential, alternate runs of the transient (that is, LOCTA - BART - LOCTA). This communication scheme does take advantage of BART's most useful features while leaving undisturbed all but one of the ECCS model's communication paths.

That one modified path is the link between LOCTA and the source of its local heat transfer coefficients. In the current ECCS model, the FLECHT correlation provides the heat transfer coefficients at each LOCTA timestep directly,

calculating the coefficients from data given by LOCTA in the same timestep. This method is interactive, ensuring that the FLECHT correlation is responsive (as responsive as an empirical correlation can be) to the progress of the transient as calculated by LOCTA.

In the proposed interim ECCS model, BART replaces the FLECHT correlation as the source of heat transfer coefficients for LOCTA. In this method the reflood portion of the transient is first calculated using BARi alone, instead of LOCTA. Then the transient is rerun with LOCTA. The BART output tape from the preceding run provides heat transfer coefficients at each timestep and at each LOCTA node. Means are provided for interpolation between BART and LOCTA nodes.

# 5-2. Large-Break LOCA Analysis Methodology

The new LOCA analysis method is compared with current methodology in the following paragraphs.

5-3. <u>Current Method</u> -- The current LOCA analysis method proceeds as shown in figure 5-1 and the step procedure below.

- (1) \* SATAN is run from zero time to the end of blowdown (EUB). SATAN is for Reactor Coolant System (RCS) behavior during blowdown.
  - \* A SATAN output tape is generated which includes the following information at each timestep:
    - System mass
    - -- System energy
    - Accumulator mass
    - -- Accumulator pressure
    - -- Pump speed
    - -- Steam generator secondary mass
    - Steam generator secondary energy

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Figure 5-1. Diagram of Current LOCA Analysis Methodology

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- (2) \* COCO (for containment behavior) is run for the entire transient.
  - \* The SATAN tape provides blowdown mass and energy release rates to COCO at each timestep up to EOB.
  - \* WREFLOOD is initiated at EOB from the SATAN tape data at EOB. (WREFLOOD is for RCS and core behavior during reflood.)
  - \* WREFLOOD runs from EOB to the end of the transient (END).
  - \* REFILL (specific RCS refill models) is run as a subsidiary routine to WREFLOOD from EOB to BOC. (WREFLOOD and REFILL communicate directly.)
  - \* After EOB, WREFLOOD communicates directly with COCO to give, at each timestep, the following:
    - -- Mass release rate into containment
    - -- Energy release rate into containment
    - -- Injection mass overflow rate
    - -- Injection overflow enthalpy
    - -- Spilling injection rate (mass)
    - -- Spilling injection enthalpy
  - \* COCO returns containment pressure and injection enthalpy to WREFLOOD.
  - \* WREFLOOD generates an output tape with the following information available at each timestep:
    - -- Core flooding rate
    - -- Core pressure
    - -- Core fluid inlet enthalpy
    - Quench front location

- (3) \* LOCTA (core thermal and mechanical behavior) is run for the entire transient.
  - \* The SATAN tape gives boundary conditions at each timestep up to EOB as follows:
    - -- Core inlet mass flow
    - -- Core inlet enthalpy
    - -- Core pressure
    - -- Cure power
  - \* The WREFLOOD tape provides boundary conditions from EOB to END.
  - \* After BOC, the FLECHI heat transfer correlation or a steam cooling model is run as a subsidiary routine to LOCTA to generate, at each timestep, local heat transfer coefficients. LOCTA and FLECHT communicate directly.

At this point the analysis using the current ECCS model is complete.

5-4. <u>Proposed Method Using BART as Separate Code</u> -- In the revised ECCS analysis, BART replaces the FLECHT correlation and certain portions of WREFLOOD with more realistic models. Steps 1 and 2 (figure 5-1) of the proposed procedure are identical to the first two steps of the current procedure. Step 3 of the current procedure, however, is modified for the proposed model as illustrated in figure 5-2. Step 3 for the proposed procedure is therefore designated Step 3'. An additional step, Step 4, is required in the revised method and Steps 5 and 6 are used conditionally (if cladding burst occurs after BOC).

- (3') \* LOCTA is run from time = zero to BOC, at which time it generates the following output:
  - -- Hot fuel assembly power profile
  - -- Local fuel rod gap heat transfer coefficients
  - -- Local flow channel blockages
  - -- Fuel rod temperature profile

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5-6. <u>Comparison of Results</u> -- The following comparison of results of ECCS evaluations using, alternately, the tightly coupled and loosely coupled versions of BART and LOCTA core and hot rod models, are based on analyses for a typical PWR. These results must be considered to be preliminary because the BART/LOCTA method has not been completely verified. a,c

Parameter	BART/LOCTA	BART + LOCTA
Peak cladding temperature	1794°F	1916°F
Time of peak cladding temperature	116.6 sec	110 sec
Elevation of peak cladding temperature	7.25 ft	6.5 ft
Rurst time	51.7 sec	45 sec

Figures 5-4 and 5-5 show hot rod cladding temperatures at 6.5 feet (peak cladding temperature elevation for BART + LOCTA) and at 7.25 feet (peak cladding temperature elevation for BART/LOCTA). BART + LOCTA is more conservative with respect to cladding temperature at all times.

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Figure 5-5. BART/LOCTA and BART + LOCTA Cladding Temperatures at a 7.25-Foot Elevation

A key point in the comparisons of peak cladding temperature locations and times is that local peaks occur, for both code versions, at 6.5 feet and about 100 seconds.

Parameter	BART/LOCTA	BARI + LOCTA
Time	96.6 sec	100 sec
Local peak cladding temperature at 6.5 feet and given time	1789°F	1914°F
Difference from global	5°F	2°F

peak cladding temperature

Comparisons at corresponding elevations and times most clearly show the similarity between the transient responses of the two code versions.

Figures 5-6 and 5-7 show hot rod axial cladding temperature profiles around the times of peak cladding temperature occurrence for BART + LUCTA and BART/ LOCTA. Exact time correspondence was not possible because of differences in output printing in the two cases.

## 5-7. Heat Transfer Prior to Entrainment

In the present WREFLOOD code, entrainment from the core is calculated with a correlation based on low flooding rate FLECHT data. As a result, entrainment is calculated to begin early in the reflood transient. When the resulting flow rate is used in BART, entrainment is delayed because of the initially high flooding rates. A situation is then encountered in which low flooding rates and no entrainment exist simultaneously in BART. Although this situation is clearly not physically correct, it is a conservative view of heat transfer from the hot rod.



Figure 5-6. Cladding Temperature Axial Profiles at 96 Seconds (Approximate Time of Peak Cladding Temperature in BART + LOCTA)

HOT ROD CLADDING TEMPERATURE (<sup>O</sup>F)

Figure 5-7. Cladding Temperature Axial Profiles at 115 Seconds (Approximate Time of Peak Cladding Temperature in BART/LOCTA) 8140E-8

The core is free to fill at the maximum rate allowed by the ECCS prior to boiling and entrainment and, at the beginning of entrainment, the liquid droplets will move at a velocity greater than the inlet flow rate. Therefore, the longest possible delay before significant heat transfer begins at a particular location can be found by calculating the time it would take to cold fill the reactor vessel at ECCS rates up to the elevation in question.

The burst location on the hot rod heats up most rapidly during adiabatic conditions because of metal-water chemical reactions, and it will cool as soon as entrainment begins because of the large gap between the cladding and the fuel. Thus temperature of the burst location at the end of the delay period described in the previous paragraph represents a conservative estimate of the peak cladding temperature of the burst location.

Because heatup rates at other locations on the hot rod are determined by rod power, the delay in entrainment calculated by BART is not as critical at these locations, although the peak cladding temperature is still conservatively high. At points on the hot rod away from the burst location, cladding temperature turnaround occurs when the BART calculated heat transfer from the rod is sufficient to overcome the effects of fuel decay heat generation.

## 5-8. CODE MODIFICATIONS

No modifications were required in the SATAN code. The LOCTA code was modified to allow the use of the BART calculated heat transfer coefficients in the hot rod calculation during reflood. Heat transfer coefficients as functions of elevation and time are supplied via a data tape to LOCTA, where linear interpolation is used as required to obtain the heat transfer coefficient at the appropriate LOCTA location and time. WREFLOOD and BART did require code modifications, and these are detailed in the following paragraphs.

## 5-9. WREFLOOD Entrainment Calculation

For a compatible reflood model for use with BART, the core mass entrainment calculation is modified to include the effect of boiling below the quench front. This model is similar to one developed in WCAP-8838.<sup>(10)</sup>

- z = solid water level (level to which mixture would collapse if voids were removed
- $z_a =$  quench front level
- $z_s = saturation$  line (level at which liquids begins to boil)

In the present WREFLOOD calcu'ation,  $z_c$ ,  $z_q$ , and  $z_s$  all coincide. The entrainment rate is calculated by

(5 - 1)

a,c

where

F = mass flow leaving core/mass flow entering core  $Q_m$  = peak linear power (ft) P = pressure (psia)  $\Delta T_s = inlet liquid subcooling (°F)$ V<sub>in</sub> = flooding rate (in./sec)  $z'_q = z_q + (12 - z_{st})(60 - P)/40$ z = quench front location (in.)  $z_{st} = 10 \frac{\overline{v}_{1n} \Delta T_s}{(1000 - T_s)}$ 

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5-19

Let:

$$V_{in} = \frac{1}{2} \int_{0}^{t} V_{in} dt$$

Because it has been assumed above that no liquid is stored above  $z_q$  and that no voids exist below  $z_q$ , F can be expressed as follows:

$$= 1 - \frac{v_q}{v_{10}}$$
(5-2)

where  $V_q = dz_q/dt$ 

Thus the quench front elevation as a function of time can be determined using equation (5-1).

The energy equation for the liquid entering the core is as follows:

$$\frac{\partial H}{\partial t} + U_{L} \frac{\partial H}{\partial z} = \frac{Q^{+++}}{\rho_{L}}$$
 (5-3)

where

The liquid density and velocity are assumed to be constant, so that

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In finite difference for, equation (5-3) becomes

$$\frac{H_{i}^{n} - H_{i}^{n-1}}{\Delta t} + V_{in} \frac{H_{i}^{n} - H_{i-1}^{n}}{\Delta z} = \frac{\vec{Q}^{(1)}}{\rho_{1}}$$
(5-4)

where

n = new time level n-1 = old time level i, i-1 = elevation  $\Delta t$ ,  $\Delta z$  = time, level increments  $\overline{Q}^{\prime\prime\prime}$  =  $Q^{\prime\prime\prime}$  averaged between  $z_{i-1}$  and  $z_i$ 

Equation (5-4) can be solved for H;.

At the location defined by  $z_q$ , equation (5-4) is modified so that

$$\frac{H_q^n - H_q^{n-1}}{\Delta t} + V_{in} \frac{H_q^n - H_j^n}{\Delta z_q} = \frac{\bar{q}'''}{\rho_L}$$
(5-5)

where

 $\begin{array}{l} H_{q} &= \mbox{ liquid enthalpy at quench front} \\ H_{j} &= \mbox{ liquid enthalpy at first elevation below quench front} \\ \Delta z_{q} &= \mbox{ } z_{q} - \mbox{ } z_{j} \end{array}$ 

Liquid enthalpy distribution can be found as a function of time at and below  $z_q$ , from equations (5-4) and (5-5). If H<sub>q</sub> is calculated to be less than H<sub>f</sub>, the saturation liquid enthalpy, it is assumed that

$$z_s = z_q$$

(5-7)

$$\frac{dz_s}{dt} = \frac{dz_q}{dt}$$

If  $H_{q}$  is greater than  $H_{f}$ , the saturation line velocity is given by

a

and  $z_s$  can be determined by integrating equation (5-7). See paragraph 2-9.

If  $z_s$  is less than  $z_q$ , boiling will occur in the region between  $z_s$  and  $z_q$ . The void fraction at a particular location may be found by using Yeh's correlation (see paragraph 2-13).

$$\alpha = 0.925 \quad \left(\frac{\rho_{v}}{\rho_{g}}\right)^{0.239} \quad \left(\frac{j_{v}}{u_{rb}}\right)^{m} \tag{(5-8)}$$

where

$$U_{rb} = 1.53 \quad \left(\frac{\sigma\Delta\rho g}{\rho_{g}^{2}}\right)^{0.25}$$

$$j_v = vapor volumetric flux$$
  
 $\sigma = surface tension$   
 $\rho_v = vapor density$   
 $\rho_{\varrho} = liquid density$   
 $m = 0.67 \text{ if } j_g/V_{bcr} < 1$   
 $= 0.47 \text{ if } j_g/V_{bcr} > 1$ 

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and
If it is assumed that no mass is stored above the quench front, then the mass stored in the core is

$$M_{stored} = p_{g} z_{s} + (1-\alpha) p_{g} (z_{a} - z_{s})$$
(5-9)

where

$$\bar{\alpha} = \frac{1}{z_q - z_s} \int_{z_s}^{z_q} \alpha \, dz \tag{5-10}$$

The collapsed liquid level,  $z_c$ , is related to  $z_s$  and  $z_q$  by

$$z_{c} = z_{q} - \int_{z_{s}}^{z_{q}} \alpha dz \qquad (5-1)$$

The entrainment rate F, which is now defined as  $1 - dz_c/dt/V_{in}$  is given by

$$F = 1 - \frac{1}{V_{in}} \left[ \frac{dz_q}{dt} - \frac{d}{dt} \int_{z_s}^{z_q} \alpha dz \right]$$
(5-12)

The integral in the above equation can be rewritten as follows:

$$\int_{z_{s}}^{z_{q}} \frac{\partial \alpha}{\partial t} dz + \alpha_{q} \frac{dz_{q}}{dt} - \alpha_{s} \frac{dz_{s}}{dt}$$
(5-13)

where  $\alpha_q$  is the void fraction at the quench front and  $\alpha_s$ , the void fraction at the saturation line, is zero.

At any elevation z, the vapor volumetric flux is given by

$$J_{v}(z) = -\frac{1}{Q_{k}} \int_{z_{s}}^{z} Q^{''} dz$$
 (5-14)

Rewriting the integral as in equation (5-13), the rate of change of  $j_v$  with time is

$$\frac{d\mathbf{j}_{\mathbf{v}}}{d\mathbf{t}} = -\frac{1}{Q_{\mathbf{k}}} \int_{z_{s}}^{z} \frac{\partial Q}{\partial \mathbf{t}} dz - Q_{s}^{\prime\prime\prime} \frac{dz_{s}}{d\mathbf{t}}$$
(5-15)

The integral in equation (5-15) represents the contribution to  $dj_y/dt$  from the decrease in power generation in the rod and will be assumed to be small. Thus,

$$\frac{dJ_v}{dt} = Q_s''' \frac{dz_s}{dt} \frac{1}{Q_v}$$
(5-16)

where  $Q_s$  is the power generation rate at the saturation line.

Differentiating equation (5-8) gives

$$\frac{d\alpha}{dt} = 0.925 \quad \left(\frac{\rho_{v}}{\rho_{g}}\right)^{0.239} \quad \frac{m}{U_{rb}m} \quad \left(\frac{J_{v}}{U_{rb}}\right)^{m-1} \quad \frac{dJ_{v}}{dt}$$
(5-17)

Therefore, the integral in equation (5-13) can be evaluated and the entrainment rate can be determined.

The position of the three lines  $z_q$ ,  $z_{sat}$ , and  $z_c$  is thus calculated by using equations (5-1), (5-7), and a core mass balance. The level used in the reactor vessel momentum balance is the quiet water level,  $z_c$ .

During blowdown and reflood, stored energy is released to the fluid from metal components and walls in the downcomer and lower plenum. This heat release raises the temperature of the liquid entering the core, which in turn affects core fluid and heat transfer conditions.

The present WREFLOOD code uses a simple, exponential decay heat release model for two components defined as thin and thick metal. The decay constants used ensure a conservative amount of hot metal heat release to the fluid.

It is desirable to obtain a more accurate measure of hot metal energy release when using the BART code. Accordingly, the following model has been added to WREFLOOD.

The various components in the downcomer and lower plenum are arranged into groups of slabs, cylinders, and hollow spheres. As many as ten different geometries are allowed in both downcomer and lower plenum. Table 5-1 shows a typical breakdown of components for a PWR.

The surfaces of the various components are assumed to remain wet during the transient, if the quality in the downcomer and lower plenum is less than 1.0, so that the heat transfer is conduction limited. When the quality is 1.0 and during refill, the walls are assumed to be adiabatic.

The metal heat flux into the fluid is calculated by numerically solving the conduction equation. An energy balance is then performed in WREFLOOD to calculate liquid temperature in the lower plenum and downcomer.

#### 5-10. BART Blockage Effects

To account for blockage in BART, the basic equations in BART were modified to accept a source term, S, representing the exit of steam from or entry of steam to the flow channel due to flow redistribution. For single-phase flow, the conservation equations are as follows.

$$\frac{\partial p}{\partial t} + \frac{\partial p U}{\partial z} = -S$$

where

p = fluid density
t = time
z = axial location
u = fluid velocity
S = mass leaving flow channel per unit time per unit volume

$$\frac{\partial \rho H}{\partial t} + \frac{\partial \rho U H}{\partial x} = Q^{++} - SH$$
 (5-19)

where

H = fluid enthalpy Q = heat flow per unit volume due to rod heat release

The two-phase equations were modified as follows (see paragraph 2-4):

$$\frac{\partial 1}{\partial z} + \alpha \frac{\partial \ln \rho_{v}}{\partial t} + \alpha U_{v} \frac{\partial \ln \rho_{v}}{\partial z} = \frac{Q_{boil}}{H_{\varrho v} \rho_{v}} \left(1 - \frac{\rho_{v}}{\rho_{\varrho}}\right) - \frac{S}{\rho_{v}}$$
(5-20)

where

j = fluid volumetric flux α = void fraction ρ<sub>v</sub> = gas density ρ<sub>g</sub> = liquid density U<sub>V<sub>1</sub></sub> = gas velocity Q<sub>boil</sub> = heat flow used to generate saturated vapor (Q - Q<sub>boil</sub> = heat used to superheat vapor)

H<sub>av</sub> = heat of vaporization

### TABLE 5-1

MAJOR COMPONENTS IN THE LOWER PLENUM AND DOWNCOMER OF A FOUR-LOOP REACTOR VESSEL

			Thickness or Diameter (in.)	
Area	Component	Туре		
	Vessel wall	Slab	8	
	Thermal shield	Slab	2.8	
Downcomer	Core barrel	Slab	2.3	
	Core formers	Slab	1.6	
	Core baffle plates	Slab	1.0	
	Vessel wall	Slab	в	
Lower plenum	Core barrel	Slab	2.3	
	Lower core support plate	Slab	1.8	
	Lower core plate	Slab	1.1	
	Flow distribution plate	Slab	0.5	
	Core support columns	Cylinder	6	
	Instrument columns	Cylinder	2.3	

If the steam leaving or entering the flow channel is assumed to carry with it the local channel steam enthalpy, the equations describing the rate of change of enthalpy of steam remain unaltered.

The source term, S. is calculated in the same way as it is in the present LOCTA IV code. (38)

#### 5-11. SAMPLE PWR CALCULATION

A typical four-loop PWR was analyzed to demonstrate the methods described in the preceding paragraphs.

Figures 5-8 to 5-10 commare the flooding rate, liquid and quench levels, and entrainment fraction for a typical four-loop plant. Entrainment begins slightly earlier in the proposed calculation. This is because

$$z_{st} = 10$$
.  $\frac{\nabla_{tn} \Delta T_s}{(1000 - T_{sat})}$ 

which has been included in the entrainment correlation. In the old model, the earlier entrainment is accounted for by subtracting  $z_{st}$  from the quench front location in the LOCTA calculation.<sup>(38)</sup> As reflood progresses, boiling commences below the quench front. Subsequently, entrainment is controlled primarily by the movement of this line. The increased entrainment reduces the flooding rate in the proposed model, but this difference tends to disappear because the downcomer driving head remains high because of the lower density in the core.

After the flooding rate was calculated, the BART code was used to calculate fluid conditions in the hot assembly. The heat transfer coefficient obtained from this calculation was then supplied to LOCTA, where the hot rod temperature transient was calculated. Figures 5-11 to 5-14 show the axial temperature distribution from beginning of reflood to turnaround for the typical four-loop plant and compare the proposed methodology with the standard method.











Figure 5-10. Entrainment for a Cold Leg Break Reflood Transient in a Typical Four-Loop Plant

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Figure 5-12. Hot Rod Cladding Temperature During Reflood at Time = 100 Seconds for a Typical Four-Loop Plant With a 15x15 Fuel Array

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igure 5-14. Hot Rod Cladding Temperature During Reflood at Time = 200 Seconds for a Typical Four-Loop Plant With a 15x15 Fuel Array

# SECTION 6

#### BART-WREFLOOD VERIFICATION

To verify that the BART-WREFLOOD combination represents a conservative method for determining peak cladding temperature in a PWR, predictions were made for two FLECHT-SET Phase B tests.<sup>(39)</sup> In these tests the rod bundle was connected to a downcomer and loops which simulate the corresponding components in a PWR. As described in the design procedure (paragraph 5-1), a WREFLOOD calculation was performed first, then the calculated flooding rates were input to BART.

Figures 6-1 to 6-6 compare predicted and measured flooding rate, cladding temperature, and quench front location. These figures show that the procedure predicts higher cladding temperatures than those observed in the data.

The major cause of the large cladding surface temperature overprediction (figures 6-2 and 6-5) is that the WREFLOOD generated flooding rates did not include the large initial injection rates that occurred in FLECHT SET runs J105B and 2714B. The discrepancies are illustrated by the shaded areas in figures 6-1 and 6-4.

To demonstrate this effect, BART was rerun for tests 31058 and 27148. The single modification made to those runs was an increase in the flooding rate to include the high initial flooding rates indicated by the shaded regions in figures 6-1 and 6-4. The improvement in cladding temperature prediction due to the modified flooding rate profiles is shown in figures 6-7 and 6-8 for run 31058 and figures 6-9 and 6-10 for run 27148.

At the 6-foot elevation, graphs are presented for the hottest rod, the average rod (after which the BART calculated rod was modeled and to which the BART rod is most comparable), and the coolest rod. The selections were made from 16 available thermocouple channels at the 6-foot elevation.

At the 8-foot elevation, plots of all available thermocouple channels are presented.

At late times in the transients the inability of BART to calculate rod quenching from the top of the core down retards the temperature drop at unquenched locations along the rod. This effect is more severe at higher elevations.

A discrepancy between the accumulator mass depletion and the measured flow totaling 48 pounds at the end of the FLECHT SET 2714B test has been noted. It translates to a lower calculated flooding rate for the test and a lower input flooding rate for the BART prediction of that test.

As a final exercise, BART was rerun with a flooding rate calculated to include the water unaccounted for in the original test. The further improvement of the BART temperature calculation demonstrates how important an accurate input flooding rate is in determining the accuracy of the entire BART model prediction.

3105B RUN NUMBER 59 PSIA CONTAINMENT PRESSURE 1100<sup>0</sup>F INITIAL CLADDING TEMPERATURE 0.84 KW/FT PEAK POWER 306<sup>0</sup> F AVERAGE HOUSING TEMPERATURE 152°F COOLANT TEMPERATURE 12.3 LBM/SEC FOR 14 SEC INJECTION RATE VARIABLE TO END 12 10 BART PREDICTION 8 FLOODING RATE (IN./SEC) 6 4 2 0 .2 175 100 125 150 0 25 50 75 TIME (SECONDS)

Figure 6-1. Flooding Rate Calculated for Run 31058









6-4

RUN NUMBER3105BCONTAINMENT PRESSURE59 PSIAINITIAL CLADDING TEMPERATURE1100°FPEAK POWER0.84 KW/FTAVERAGE HOUSING TEMPERATURE306°FCOOLANT TE/MPERATURE152°FINJECTION RATE12.3 LBM/SEC FOR 14 SECVARIABLE TO END



Figure 6-3. Quench Front for Run 31058 Showing Maximum and Minimum Times for Each Elevation

RUN NUMBER CONTAINMENT PRESSURE INITIAL CLADDING TEMPERATURE PEAK POWER AVERAGE HOUSING TEMPERATURE COOLANT TEMPERATURE INJECTION RATE 2714B 20 PSIA 1100<sup>o</sup>F 0.84 KW/FT 249<sup>o</sup>F 153<sup>o</sup>F 12.2 LBM/SEC FOR 11 SEC VARIABLE TO END



Figure 6-4. Flooding Rate Calculated for Run 27148



Figure 6-5. Cladding Temperature at 6 Feet (a) and 8 Feet (b) for Run 27148

RUN NUMBER CONTAINMENT PRESSURE INITIAL CLADDING TEMPERATURE PEAK POWER AVERAGE HOUSING TEMPERATURE COOLANT TEMPERATURE INJECTION RATE 2714B 20 PSIA 1100<sup>o</sup>F 0.84 KW/FT 249<sup>o</sup>F 153<sup>o</sup>F 12.2 LBM/SEC FOR 11 SEC VARIABLE TO END



Figure 6-6. Quench Front for Run 2714B Showing Maximum and Minimum Times for Each Elevation

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Figure 6-7. BART Versus FLECHT SET Prediction for Run 31058 Rod Temperatures at a 6-Foot Elevation





2000 FLECHT SET PHASE B **RUN 2714B** AUGMENTED FLOODING RATE 1750 INCLUDING LOST MASS 1500 BART: **ORIGINAL PREDICTION** TEMPERATURE (°F) AUGMENTED FLOODING RATE 1250 (INITIAL PEAK) FLOODING RATE INCLUDING 2714B LOST MASS 1000 DATA 750 ORIGINAL 500 PREDICTION (FIGURE 6.1.5) 250 0 100 200 300 400 500 544 -59 -0 TIME (SECONDS)

Figure 6-9. BART Versus FLECHT SET Prediction for Run 2714B Rod Temperatures at a 6-Foot Elevation

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FLECHT SET PHASE B RUN 2714B ORIGINAL PREDICTION (FIGURE 6-5) AUGMENTED FLOODING RATE INCLUDING "LOST MASS" TEMPERATURE (°F) DATA -59.2 TIME (SEC)



## SECTION 7 BART SENSITIVITY STUDIES

#### 7-1. EFFECT OF FUEL ROD ON REFLOOD HEAT TRANSFER

The mechanistic nature of the quench model in BART makes it possible to use BART to investigate the heat transfer characteristics of fuel rods in reflood conditions.

Calculations were repeated with FLECHT test 5132 using fuel rods rather than heater rods. To perform this calculation, specific heat and conductivity for  $UO_2$  and zircaloy were put into BARI. Gap heat transfer coefficients were also entered.

Table 7-1 shows LOCTA predictions of gap heat transfer coefficients at various elevations and at various times for the reflood phase of a typical PWR large break LOCA.

Initial values for BART are the LOCTA-calculated values at beginning of core recovery. Different PWR LOCA scenarios yield slightly different gap conductance profiles, and the one presented is typical. Figure 7-1 shows both design and LOCTA-predicted gap heat transfer coefficients at beginning of core recovery.

BART holds the input gap heat transfer coefficients constant, although in an actual reflood transient, they decrease from the initial values with time at the important lower elevations. The constant gap heat transfer coefficients retard the advance.

Figures 7-2 to 7-4 compare predicted heat transfer coefficients and cladding temperatures at various elevations, and quench front location, showing the changes which occur because of differences between heater rods and fuel rods.

* *	64.4	10 C	
10	м1		1 - 1
1.04	<b>D</b> I.		

Elevation		T	Time After Core Recovery (Seconds)			5)
(Ft)	0	30	50	100	150	190 <sup>(a)</sup>
0.0	261	325	20.9	211	215	217
1.5	429	412	263	2.0	241	241
3.0	453	125	110	61	54	53
4.0	296	141	132	103	58	55
5.0	178	153	146	128	108	59
6.0	158	155	152	143	129	117
7.0	162	154	155	153	143	135
8.0	227	146	146	149	142	138
10.5	399	459	472	215	130	133
12.0	327	361	395	461	464	417

VARIATION OF GAP HEAT TRANSFER COEFFICIENT (BTU/HR-FT<sup>2</sup>-"F) DURING REFLOOD AS CALCULATED BY THE LOCTA CODE

a. 190 seconds is the end of the transient calculations.

## 7-2. EFFECI OF NODING ON BART CALCULATION

To investigate BART sensitivity to mesh size, the BART calculation for FLECHT SEASET test 32333 (see section 4) was repeated with the axial mesh size reduced by a factor of two.

Figure 7-5 shows that there is an overall increase in calculated temperature of about 100°F. Figure 7-6 shows that the prediction still compares well with data.

8140G-2



Figure 7-1. Variation of Gap Heat Transfer Coefficient During Reflood as Calculated by the LOCTA Code



Figure 7-2. Data Comparisons of Cladding Temperatures at Various Elevations for FLECHT Cosine Test 5132



Figure 7-3. Comparison of Predicted and Measured Bundle Mass for FLECHT Cosine Test 5132



Figure 7-4. Comparison of Quench Front Elevations for FLECHT Cosine Test 5132



Figure 7-5. Comparison of Cladding Temperature for Fine and Coarse Mesh BART Noding for FLECHT SEASET Test 32333



Figure 7-6. Comparison of Cladding Temperature and BART Fine Mesh Prediction for FLECHT SEASET Test 32333

## SECTION 8 CONCLUSIONS

BART Al shows good agreement with reflood data over a wide range of fluid inlet and bundle conditions. Based on these comparisons, physical models developed to describe the phenomena occurring in a rod bundle during reflood are sound and are generally applicable to the problem of reflood heat transfer. Furthermore, the empirical constants required in some of the models were obtained and are used in ways which do not compromise the mechanistic nature of the code. Thus, BART Al may be used with some confidence to determine conditions in a nuclear core during the reflood phase of large loss-ofcoolant accident.

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### APPENDIX A

## NOMENCLATURE -

AF	= channel flow area
A <sub>HT</sub>	= heat transfer surface area per unit volume
a	= vapor generation rate per unit volume
в	= term in equation (3-95)
b	= vapor reduction rate per unit volume
Cdd	= droplet drag coefficient
c, c <sub>1</sub> , c <sub>2</sub> , c <sub>3</sub> , c <sub>n</sub>	= constants or coefficients
C <sub>p</sub>	= specific heat
0 <sub>b</sub>	= bubble diameter
0 <sub>d</sub>	= droplet diameter
Dd	= initial droplet diameter
Dh	= hydraulic diameter
f	= friction factor
F	= grey body factor
G	= mass velocity
g	= gravitational acceleration
н	= fluid enthalpy
Hev	= latent heat of vaporization
h	= heat transfer coefficient from rod surface to fluid
hcond	= condensation heat transfer coefficient at the quench front
hbb	= heat transfer coefficient between vapor bubble and subcooled liquid
hNB	= nucleate boiling heat transfer coefficient
hSF	single-phase forced convection heat transfer coefficient

A-1
t	= volumetric flux
Jvc	= maximum vapor volumetric flux
Jvcond	= vapor flux condensed by liquid
Jvb	= bubble volumetric flux
k	= thermal conductivity
L	= characteristic length for condensation at the quench front
м	= mass
m	= interface disturbance wave number
Nu	= Nusselt number
р	= perimeter
р	= pressure
Pr	= Prandtl number
Q	= total heat flow in boiling zone
q	= heat flow rate
q*	= heat flow rate per unit area
q*'	= heat flow rate per unit volume
qcond	= heat flux due to condensation at quench front
ą	= average heat flow rate
q <sub>ma x</sub>	= maximum heat flux
qnbb	<pre>= heat flow represented by bubble flux j<sub>vb</sub> from rod surface</pre>
r	= radial location from rod center
Re	= Reynolds number
T	= temperature
Tf	= fluid temperature (liquid or vapor)
Tq	= rod temperature immediately above quench front region

Tsat	= fluid saturation temperature
Tu	= rod surface temperature
t	= time
Ug	= liquid velocity
U <sub>ra</sub> , U <sub>rb</sub>	= critical relative velocity in vapor film
Ur	= relative velocity
Ure	= liquid velocity relative to quench front
Urbb	= bubble relative velocity
٧	= rate of change of position z of isotherm on rod surface
٧q	= quench front velocity
W	= mass flow rate
We	= Weber number
Z	= axial position
zf	= liquid-vapor front location
a	= void fraction
α <sub>c</sub>	= void fraction in bulk fluid
αd	= dispersed regime initial void fraction
αŧ	= vapor film void fraction
app	= bubble void fraction on rod surface
ΔT <sub>S</sub>	= T <sub>sat</sub> - T <sub>t</sub>
Δρ	$= \rho_{\xi} - \rho_{V}$
λ	= interface disturbance wavelength
¢	= emissivity
ρ	= density
(ADq) (ADq)	= heat capacity of rod per unit length
(2a)	= heat capacity per unit volume

σ	= surface tension
1	= characteristic time
٢	<pre>= rate of vapor generation due to boiling per unit volume</pre>
μ	= viscosity

Subscripts

c	convection, cladding, channel
d	droplet
f	fuel, film
F	liquid vapor interface
1	axial nod location
8	liquid
nb	nucleate boiling
r	radiation
s, sat	value at saturation conditions
v	vapor
w	wall
w2	wall to liquid
wv	wall to vapor
٤v	liquid to vapor

# Superscripts

n	time step increment number
	flow per unit area
111	flow per unit volume

# APPENDIX B

## REVIEW QUESTIONS AND ANSWERS

Questions generated during the NRC review of a preliminary version of this SCAP are addressed in this appendix.

B-1. REVIEW QUESTIONS ON SECTION 2

B-2. Questions on Paragraphs 2-1 through 2-10

1. How sensitive are the BART results to axial mesh size?

See section 6

2. In the numerical solution how is  $Q_{\rm K} = \rho^2 \frac{\partial H}{\partial \rho}$  in the ratio  $q^{\prime \prime \prime}/Q_{\rm Kn}$  determined?

BART calculates water properties with polynominal fits of the steam tables. These polynominals express the density and temperature as functions of pressure and enthalpy. To calculate the quantity  $\partial p/\partial H$  a partial derivative is taken on the polynominal equation. Thus:

$$Q_{K_{1-1}} = \rho^2 (p, H_{1-1}) \left[ \frac{\partial \rho (p, H_{1-1})}{\partial H} \right]^{-1}$$

3. The saturated water/two-phase interface is defined by  $H = H_{sat}$  where  $H = H(\rho)$  and  $\rho$  is determined for a hydraulic control volume bounded by  $z_1, z_{1-1}$ . It would appear that the interface would have to be at the hydraulic node boundary and then the saturation line taken at the top of the volume  $z = z_1$ . Specify how the interface calculations are initialized. How does the code discern the initial locations of these boundaries?

Appendix C contains a more detailed description.

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4. It is inferred that equation (2-17) is integrated over the full axial dimension to determine new vapor temperatures. Is that the case? If so, what is done below the subcooled water/two-phase fluid interface and how does the nodalization fit in with the location of that interface? If it is not so, where does the integration start and how does it fit the nodalization?

Appendix C contains a more detailed description.

5. In the single-phase region, the new enthalpy values are determined from  $H = \gamma/\rho$ . Because the pressure is constant  $H = H(\rho, p)$  also. How do these different determinations of H compare?

In single-phase conditions BART performs the following steps:

- (1) Solve the continuity equation, Eq. (2-1)
- (2) Solve the energy (pH) equation Eq. (2-2)
- (3) Recalculate  $\rho$  from the equation of state

It is felt this procedure gives a more accurate solution of the energy equation.

6. The determination of the two-phase fluid/single-phase vapor boundary is quite unclear. Elaborate on this interface determination; e.g., how does it get started?

Appendix C contains a more detailed description.

 How does the code decide the location of the droplet acceleration section of the two-phase region?

See paragraph 2-15.

8. Provide clear definitions of the three flow regimes described in the two-phase region. How are these regimes distinguished and separated? Do these three regions encompass all the expected flow regimes; e.g., churn turbulent, bubbly, etc? Do these three regimes exhaust the phenomena in the two-phase region?

See paragraph 2-15.

9. In the derivation of Equation (2-8), is it assumed that  $\alpha$  is constant? If not, provide the derivation of Equation (2-8).

The liquid continuity equation [equation (2-5)] is as follows:

$$\frac{\partial(1-\alpha) \rho_{\varrho}}{\partial t} + \frac{\partial(1-\alpha) U_{\varrho} \rho_{\varrho}}{\partial z} = -\Gamma$$
(B-1)

where  $\Gamma$  = rate of vapor generation (1b/ft<sup>3</sup>-sec).

The vapor continuity equation [equation (2-4)] is as follows:

$$\frac{\partial \alpha \rho_{v}}{\partial t} + \frac{\partial \alpha U_{v} \rho_{v}}{\partial z} = \Gamma$$
(B-2)

Expand equations (B-1) and (B-2) and divide by  $\rho_{\rm Q}$  and  $\rho_{\rm V}$ . Also assume that  $\rho_{\rm Q}$  = constant. This leads to

$$\frac{\partial(1-\alpha)}{\partial t} + \frac{\partial(1-\alpha)}{\partial z} = -\Gamma/\rho_{g}$$
(B-3)

$$\frac{\partial \alpha}{\partial t} + \frac{\partial \alpha U_{v}}{\partial z} + \frac{\alpha}{\rho_{v}} \frac{\partial \rho_{v}}{\partial t} + \frac{\alpha U_{v}}{\rho_{v}} \frac{\partial \rho_{v}}{\partial z} = \Gamma/\rho_{v}$$
(B-4)

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Adding equations (8-3) and (8-4) gives

$$\frac{\partial J}{\partial z} + \alpha \frac{\partial \ln \rho_{v}}{\partial t} + J_{v} \frac{\partial \ln \rho_{v}}{\partial z} = \Gamma \left(\frac{1}{\rho_{v}} - \frac{1}{\rho_{g}}\right)$$
$$= \frac{Q_{boll}}{H_{gv}} \left(\frac{1}{\rho_{v}} - \frac{1}{\rho_{g}}\right)$$

where

$$j = (1 - \alpha) U_0 + \alpha U_y$$

$$j_v = \alpha U_v$$

## 8-3. Questions on Paragraphs 2-11 Through 2-17

 How are the "below quench front," "at quench front," "above quench front" flow regimes defined and how do they integrate into the flow regime models introduced in the hydraulic model section?

See appendix D for a more detailed description.

2. What size axial meshing does BART need to be able to resolve and separate all the various flow regimes and local void fractions?

See section 6.

3. Where and how is  $\alpha_c$  of equation (2-72) defined?

See paragraph 2-15.

4. Where and how is  $j_{vc}$  determined for the evaluation of equation (2-77)?

There is no term  $j_{vc}$  in equation (2-77).

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8-4

5. In equation (2-91), it would seem that there is an area problem. Do both h<sub>c</sub> + h<sub>NB</sub> function over the same wall surface areas? Justify this assumption if so.

See appendix D.

6. The reference for equation (2-96) has √f/2 rather than √f/8. Why is there a difference?

The friction factor in the reference is defined as follows:

$$f = \frac{1}{4} \begin{pmatrix} 0 \\ 1 \end{pmatrix} \frac{\Delta p}{\frac{1}{2} \rho v^2}$$

where

D = pipe diameter L = pipe length Ap = pressure drop across the pipe

This leads to the following expression, for example, for laminar flow:

$$f = \frac{16}{Re}$$

In BART, following mechanical engineering practice, the following definition is used:

$$f = \left(\frac{D}{L}\right) \frac{\Delta p}{\frac{1}{2} \rho v^2}$$

This definition leads to friction factors for laminar flow of

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

Because equation (2-96) contains f based on the first definition, the f calculated in BART must be divided by 4.

7. What happens to the calculation if We = 7 is varied?

. Figure B-1 shows how the calculated Weber number varies with elevation for FLECHT tests 5342 and 6638. The figure shows that the Weber number is much smaller than 7. Therefore, varying the Weber number at which droplet breakup occurs will not have any effect on the calculation unless the Weber number is greatly reduced.

#### 8-4. Questions on Paragraph 2-19

1. The calculation  $\frac{C}{U_r n} = D_{d_0}$  is used to determine initial droplet size. From data comparisons with various FLECHT tests, C = 1.26, n = 1.73 were determined. The data comparisons are done with fixed droplet sizes sizes  $D_{d_0} = 0.0035$ , 0.005 feet, not C and n. What size droplets do the specified values of C and n predict and what values of C and n predict droplets in the close range of 0.0035 feet?

The equation above for  $D_{d_0}$  is only a function of pressure and hydraulic diameter. Since these parameters do not change in BART during a calculation, the initial droplet size remains constant. The word "initial" is used here to mean at the spatial location where drops are first formed (that is; just above the quench front).

2. The  $D_{d_0} = 0.0035$ ,  $\alpha_d = 0.90$  seems to give a better calculation. Why choose  $\alpha_d = 0.99$ ,  $D_{d_0} = 0.0035$ ?

It was desirable to maintain some conservatism in comparison to a larger body of data. The main purpose of this study was to demonstrate that the model was not overly sensitive to these parameters.



Figure 8-1. Calculated Weber Number for FLECHT Tests 6638 and 5342

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## B-5. REVIEW QUESTIONS ON SECTION 3

- 8-6. Questions on Paragraphs 3-5 Through 3-7
- 1. Define  $Z_{1,1}^n$  on page 3-19.

 $Z_{1,j}^n$  is the finite difference approximation to  $z_{1,j}^n$  in the equations on pages 3-18 and 3-19. In essence, the error introduced by dropping higher order terms is now contained in the Z's.

 Elaborate on how the quench front isotherm migration two-dimensional heat conduction analysis is coupled with the overall one-dimensional rod model.

See paragraph 3-19

3. How are the approximations for K and pc determined (see pages 3-19 and 3-20)?

The approximations for K and  $\rho c$  are defined in the same sense as the approximation for z (see question 1 above)

### B-7. Questions on Paragraph 3-9

1. How is j cond developed for the turbulent flow model or is the whole component H Cond Ts substituted for h A J Cond in equation (3-60)? How is the turbulent model incorporated into equation (3-61)? There seems to be some inconsistency in the whole J Cond Calculation. The q<sup>#</sup> Cond is heat flux from vapor to liquid which results in vapor condensation at the interface and heating of the liquid, equation (3-62). Does all the vapor condense? It appears that all of the heat flux goes into the water. How is the vapor film maintained? Clarify the allocation of the heat fluxes and describe the physical phenomenas.

In turbulent flow the condensation heat flux is assumed to be given by

$$q_{cond} = C_5 (\rho cu)_{\varrho} Re_{\varrho}^{-2} \Delta T_s$$

$$j_{cond} = \frac{4C_5 \Delta T_s}{D_h H_{\varrho v} \rho_v} \frac{(\rho cu)_{\varrho} \lambda}{Re^{-2}}$$

This is the volumetric flux condensed in turbulent flow and is used in equation (3-60). Equation (3-60) describes the physical situation shown in figure B-2 below



Figure B-2. Sketch of Physical Equivalent of Equation (3-60)

As the quench front is approached, the heat flux builds rapidly and a vapor film begins to form. Of the heat released by the rod, a fraction generates vapor (leading to  $j_{vc}$ ) and the recarder is condensed  $(q'_{cond})$ . It is postulated in this model that the volumetric flux of vapor  $j_{vc}$  controls the overall flow of heat from the rod surface in the shaded region in figure B-2. If some of the vapor generated is condensed, more heat can flow from the wall before the maximum  $j_{vc}$  is reached and the film stability is affected.

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How is it that the area of condensation is the flow channel area rather than the liquid slug surface area?

Equation (3-64) is basically a simplification of equation (2-15):

$$\frac{dj}{dz} = -\frac{q_{cond}}{q_{k}}$$

where:

$$Q_{cond} = Q_{cond} P/A_F$$

= condensation per unit fluid volume

$$Q_{K} = \frac{\rho_{v} H_{\varrho v}}{1 - \rho_{v}} / \rho_{\varrho}$$
$$\sim \frac{\rho_{v} H_{\varrho v}}{1 - \rho_{v}} + \frac{1}{2} + \frac{1}{2}$$

Over some length L, the amount of vapor condensed is then

$$\Delta j_v = -j_v \text{ cond} = \frac{q_{cond}}{\rho_v} = \frac{q_{cond}}{A_F} = \frac{q_{c$$

The surface area over which condensation is assumed to take place is PL, which is the surface area of the rod. Since the flow regime at the quench front is envisioned as inverted annular, the rod surface area is a reasonable first approximation of the actual interface area.

 Justify using a semi-infinite solid model for a relatively small fluid volume in the flow channel when developing equation (3-62). In developing the condensation models, a relationship was sought between the amount of vapor condensed and the subcooling,  $\Delta Ts$ , and liquid mass velocity,  $(\rho U)_{g}$ . It is believed that using a more detailed model such as a slab or cylinder would not substantially change the relationship.

$$Q_{cond} = f(\Delta T_s, \forall \rho U_s)$$

4. Is the vapor flow area  $A_F$  of equation (3-50) the entire channel cross-sectional area? This seems contradictory to the notion that the heat transfer is controlled by the vapor flow out of the film generation regime.

The quantity  $j_{vc} A_F$  is the total volumetric flux of vapor. If the vapor film area were to be used, the proper relationship would be

$$Q^{A}_{HT} = \rho_{V} H_{QV} U_{V} A_{film}$$

where

U = vapor velocity

Afilm = film cross-sectional area

5. The quench front is supposed to be in the two-phase regime above H = H sat What is the mechanism by which the subcooled fluid is transported into the quench region to condense the vapor?

The saturation line is allowed to move above the quench front, if the mass velocity is high enough. Thus subcooled fluid may reach the quench front and condense vaport there.

6. How does one get the use of the U of equation (3-63)?

The liquid velocity  $U_{g}$  in equation (3-63), as well as equations (3-67), (3-68), and (3-70) is calculated at the beginning of the quench region (see appendix D) using equations (3-75) and (3-76).

 It appears that the quench front heat release model overlaps the onedimensional heat release model. Show that there is not a double accounting in this heat flow calculation.

See response to question 2, paragraph B-6.

8. What is the u in equation (3-70)?

The u in equation (3-70) is the liquid velocity (see appendix D).

9. The quench front is supposed to be in the two-phase regime above H = H<sub>sat</sub>. What is the mechanism by which the subcooled fluid is transported into the quench region to condense the vapor?

The saturation line is allowed to move above the quench front if the mass velocity is high enough. Thus subcooled fluid may reach the quench front and condense vapor there.

### B-8. Questions on Paragraph 3-12

 The thermal and hydraulic modeling is referred to as "near the quench front." Quantify this statement.

Appendix D describes how the calculations are performed near the quench front.

2. If the control volume boundaries velocities are allowed to be different, how can the mass velocity in one and not the other change? Granted that  $\partial G/\partial z = 0$  is an assumption, quantify this assumption. Consider the control volume shown in figure B-3 below.



1

Figure B-3. Sketch of Control Volume

 $V_1$ ,  $V_{1-1}$  = control volume boundary velocities  $U_1$ ,  $U_{1-1}$  = fluid velocity (relative to fixed frame of reference)

A mass balance yields

$$\frac{d}{dt} (\rho_1 \Delta z_1) = \rho_{1-1} (U_{1-1} - V_{1-1}) - \rho_1 (U_1 - V_1)$$

Expanding the derivative and rearranging, and noting that  $d\Delta z_1/dt = V_1 - V_{1-1}$ .

$$\Delta z_{1} \frac{d\rho_{1}}{dt} = \rho_{1-1} U_{1-1} - \rho_{1} U_{1} - V_{1-1} (\rho_{1-1} - \rho_{1})$$

If density gradients are small or if the boundary velocities are small, the mass balance above reduces to the normal fixed-frame of reference mass balance:

$$\Delta z_{1} \frac{d\rho_{1}}{dt} = \rho_{1-1} U_{1-1} - \rho_{1} U_{1}$$

The further assumption that the rate of change of density is small leads to a uniform mass velocity.

The assumption that  $\partial G/\partial Z = 0$  near the quench front is a reasonable approximation in view of the above and the generally good agreement between predictions of this model and data.

3. The local vapor flux  $j_{vi}$  is treated in the same way as the global vapor flux in that  $j_{vi} = F_1 \alpha_i$ . Does the vapor film generation enter into this  $\alpha_i$  and hence into  $j_{vi}$ ?

Equation (3-75) is used below the quench region, where the flow regime is bubbly or churn turbulent. As described in appendix D, fluid flow calculations in the quench model are terminated at the beginning of the vapor film.

4. Why is  $\Delta z_1$  in equation (3-79) not time dependent? Does this infer that the isotherm spacing is constant?

Equation (3-79) is in error and has been corrected. The correct derivation appears below:

$$\frac{d}{dt} \left[\rho \varepsilon_{1} \left(z_{1} - z_{1-1}\right) H_{1}\right] = \rho \left(U_{1-1} - V_{1-1}\right) \varepsilon_{1-1} H_{1-1}^{-1}$$

$$-\rho \left(U_{1} - V_{1}\right) \varepsilon_{1} H_{1}^{-1} + q_{1}^{+1} \left(z_{1} - z_{1-1}\right)$$
(B-5)

The liquid continuity equation is

$$\frac{d}{dt} \left[\rho \varepsilon_{1}(z_{1} - z_{1-1})\right] = \rho \left(U_{1-1} - V_{1-1}\right) \varepsilon_{1-1} - \rho \left(U_{1} - V_{1}\right) \varepsilon_{1}$$
(B-6)

where  $\varepsilon_1 = (1 - \alpha_1)$ . Assume that the liquid density is constant.

Equation (B-5) becomes:

$$\epsilon_{1} (z_{1} - z_{1-1}) \frac{dH_{1}}{dt} + H_{1} \frac{d}{dt} \epsilon_{1} (z_{1} - z_{1-1})$$

$$= (U_{1-1} - V_{1-1}) \epsilon_{1-1} H_{1-1} - (U_{1} - V_{1}) \epsilon_{1} H_{1} + (B-7)$$

$$+ q_{1}^{"'} (z_{1} - z_{1-1}) / \rho$$

Substitute equation (B-6) into equation (B-7):

$$\epsilon_{1} (z_{1} - z_{1-1}) \frac{dH_{1}}{dt} = (U_{1-1} - V_{1-1}) \epsilon_{1-1} H_{1-1}^{-1}$$

$$- (U_{1-1} - V_{1-1}) \epsilon_{1-1} H_{1} + \frac{H_{1}}{2} (z_{1} - z_{1-1}) / \rho$$

$$\epsilon_{1} (z_{1} - z_{1-1}) \frac{dH_{1}}{dt} = (U_{1-1} - V_{1-1}) \epsilon_{1-1} (H_{1-1} - H_{1}) + \frac{H_{1}}{2} (z_{1} - z_{1-1}) / \rho$$

5. It is implied that the volumetric flux of equation (2-15) is the  $j_1$  of equation (3-72) which is used to determine  $\alpha_1$  in equation (3-76). Why is the  $\alpha$  from the same set of equations, (2-15) through (2-29), not used for the local void fraction?

It was desirable to simplify the equations because of the addition of the condensation term  $b_1 \alpha_1$ . Furthermore, an exact treatment with the complete continuity equations would have required accounting for the moving mesh which would have required more involved mathematics. In view of the slow movement of the isotherms, such detail is unwarranted.

# B-9. Questions on Paragraph 3-13

 With reference to the isotherms in figure (3-10), and the rod heat transfer to the liquid, what is used for the wall temperature between the i<sup>th</sup> isotherm and the i+1<sup>th</sup> isotherm? In general, how closely spaced are the isotherms between T<sub>sat</sub> and T<sub>min</sub>?

Figure B-4 shows the calculated isotherm mesh at T = 100 seconds for FLECHT test 4831. The calculated heat flux is also drawn in a way which indicates how the heat flux is treated numerically (that is, the temperature between isotherms i and i + 1 is the temperature given by isotherm i).

2. In the rod-to-liquid heat transfer, is there a mechanism by which the generated voids which are not recondensed are swept downstream of the control volume?

The flux of vapor leaving a particular control volume is given by

jv, = F1 a1"

Basically, the above equation describes the flux of vapor upwards due to its buoyancy.

3. Is it significant that there is no reduction of the bubble dimension in the condensation model in equation (3-87)?

It is true that condensing bubbles will reduce in volume as they flow through the liquid. However, new bubbles are continually being formed on the rod. An accurate bubble model would have to track the change in volume of each individual bubble. As described in appendix D, a mean bubble size is used to represent the entire bubble population.

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4. The expression  $q_{nb}$  does not apppear in paragraph 3-14. Is  $q_{nb}$  actually the value Q determined by integrating up the elevation of the control volume? If so, should not  $q_{nb}$  in that integral be a function of z? Clarify the definition of  $q_{nb}$ .

See appendix D.

5. Which a is the a of equation (3-87)?

 $\alpha_1$  is calculated using equation (3-76).

6. It is not clear that the heat fluxes of equation (3-88) function over the same areas. Do forced convection and nuclear boiling occur over the same surface? If so, how?

See appendix D.

 In the condensation model, equation (3-87), what, if any, mechanism accounts for the bubbles getting smaller as they condense?

A mean bubble size is assumed as representative of the bubble population in the channel.

B-10. Questions on Paragraph 3-14

 It appears that region IV was omitted from table 3-2. Should it be included? If not, explain why.

Table 3-2 has been corrected.

2. More clearly define the boundaries between region III and region II.

See appendix D for clarification.

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3. More clearly define the boundaries between region II and region I.

See appendix D for clarification.

4. It is inferred that the rod Liedenfrost temperature is used to determine a quench region isotherm. How is this temperature determined? How is the minimum rewetted rod surface temperature determined? It appears that one of these is an input value -- how is it determined and what are the effects of variation in this value?

The temperature at which the minimum heat flux occurs on the rod is used to define the location of the quench front. As shown in figure 3-11, this temperature varies with the value of B. It is not an input quantity. The only requirement is that the isotherm mesh define this region reasonably well.

5. Due to the general upward propagation of energy released during reflood, doesn't the linearized apportioning of energy favor the quench front advance. That is, at  $z_q = z_1$ , all the heat is realized into the lower node reducing the subcooling there rather than generating more steam at the quench plane and thereby slowing the advance of the quench plant.

Sufficient detail exists in the isotherm mesh to avoid the severe distortion suggested above as figure B-2 shows.

6. On page 3-45, the definition of the "approximate" boundary between region I and II appears to be missing. Is that the case? If so, what is the definition?

The boundary between region I and II is defined as the point where the heat flux equals or exceeds

q = py Hey Jyb

8-19

where:

 $J_{vb} = U_{rbb} \alpha_{bb} \qquad [(see equation (3-89)]$   $\alpha_{bb} = \frac{\pi D_b^2}{2 \lambda^2} \qquad [(see equation (3-90)]$   $U_{rbb} = 1.53 \qquad \left[\frac{\sigma \Delta \rho q}{\rho k^2}\right]^{0.25}$ 

 What happens to the bubbles formed in region I, page 3-45? The implication is that they are swept downstream. How does the condensation model account for them?

The flux of vapor is determined from equation (3-75). The change in void fraction as a result of condensation, the flux of vapor from below, and the flux of vapor leaving the region is calculated from equation (3-76).

#### B-11. Review Questions on Section 5

 Is there any particular reason for not incorporating BART into LOCTA to take advantage of the LOCTA gap conductance calculations as well as eliminate the final stand-alone LOCTA calculation?

See paragraph 5-5.

 A comparison of the current and proposed WREFLOOD calculation is not presented. This is needed, in particular, a comparison of the entrainment fractions.

See paragraph 5-11.

 A comparison of the surface heat transfer coefficients from both BART and the current FLECHT correlation should be provided.

See section 4.

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4. Is there any means of comparing the levels of  $z_q$  and  $z_s$  produced by WREFLOOD and BART for the same reflood analysis? The WREFLOOD/BART incompatibility is not clear. The level calculations in WREFLOOD are done in BART, why not use them?

The levels calculated by BART cannot be directly compared with WREFLOOD because BART calculates conditions in the hot assembly and WREFLOOD calculates conditions in the average assembly.

 Provide the comparisons of the peak cladding temperatures using both the current ECCS evaluation model package and the proposed model.

See paragraph 5-11.

## B-12. Review Questions on Section 6

 It was understood that BART required a flooding rate as input. Describe how the BART flooding rate prediction in figures 6-1 and 6-4 was developed.

The flooding rates predicted in figures 6-1 and 6-4 were calculated using the WREFLOOD code. The procedure described in paragraph 5-4 was used.

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 The cladding wall temperature comparison graphs in figures 6-2 and 6-5 do not include elevations or rod identification. A more complete and detailed data presentation is needed.

See section 6.

 How does one account for the substantial cladding surface temperature overprediction of figures 6-2 and 6-5?

See section 6.

4. Provide comparison calculations of the PWR FLECHT SEASET unblocked bundle, forced, and gravity reflood tests 32333 and 32234.<sup>(36)</sup> These comparisons should include, but not be restricted to, calculations modeling rod 6L in test 32333 and 6K in test 32235; also rods 7J, 8K, 9K and 11E for both tests. Include the input data in the calculation summary.

See paragraph 4-5.

# APPENDIX C ADDITIONAL INFORMATION ON FLUID INTERFACES

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# C-1. INTRODUCTION

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. 8 Fluid interfaces are used in BART to model the initial stages of reflood and to define the boundary between boiling and subcooled flow. The sequence of events is shown in figure C-1 below.





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# C-2. LIQUID-VAPOR INTERFACES

This interface exists at the beginning of the reflood transient and is located initially at the bottom of the core. The numerical scheme is shown in figure C-2 below.



Figure C-2. Sketch of the Numerical Scheme

Initial conditions are as follows:

$$z_F = z_1$$
$$\frac{dz_F}{dt} = U_1$$
$$\rho_F = \rho_1$$

Transient calculations are shown below.

Similarily, the numerical form of the energy equation is

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## C-3. SATURATION LINE

As the liquid-vapor interface  $z_F$  moves into the core, the enthalpy at the interface  $H_F$  increases until it reaches saturation. At this point, the saturation line is formed. The movement of this line is calculated in the following manner.

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The initial conditions are

 $z_{sat} = z_F$  where  $\gamma_F / \rho_F$  first reaches saturation.

 $\frac{dz_{sat}}{dt} = \frac{dz_F}{dt}$ 

To solve equation 2-28 for  $dz_{sat}/dt$ , the following scheme (figure C-3) is used.

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## C-4. CALCULATIONS ABOVE THE SATURATION LINE

Above the saturation line, the liquid-vapor interface z<sub>F</sub> continues to exist, except that now the liquid is part of a two-phase mixture. The situation is shown in figure C-4 below.



Figure C-4. Sketch of Liquid-Vapor Interface Interface Above the Saturation Line

The initial conditions are as follows:

$$z_F = z_{sat}$$

Transient calculations are shown below. The  $z_F$  interface is defined as the location of the liquid front. In contrast to the subcooled liquid interface, vapor can now flow through the interface. The interface velocity is

$$\frac{dz_F}{dt} = U_{g}(z_F)$$
$$= J_{z_F} - \alpha_F U$$

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Integrating from  $z_{sat}$  to  $z_F$  yields

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The above equation can be put in numerical form and rearranged to give equation (2-52) for  $\alpha_{\rm F}$ . After boiling has begun,  $z_{\rm F}$  moves rapidly up the channel until liquid begins to exit the core. Figure C-5 shows the movement of these interfaces for FLECHT test 5342.

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Figure C-5. Graph of Interface Movement in FLECHT Test 5342

#### APPENDIX D

#### ADDITINAL INFORMATION ON THE QUENCH FRONT MODEL

#### D-1. HEAT TRANSFER CALCULATIONS NEAR THE QUENCH FRONT

Figure D-1 illustrates a typical isotherm mesh and the region within which detailed calculations are made. The quench region shown is one BART node in length below the quench front (defined later) and extends 3 inches above the front. Below the quench region, the fluid temperature and heat transfer coefficient at each isotherm location is determined by interpolating between the values calculated in BART. The conditions at the last isotherm below the region (in this case, the 260°F isotherm) serve as the starting point for detailed calculations within the region. Equations (3-76) and (3-79) are used to calculate the liquid velocity and enthalpy.

If the wall temperature is below the saturation temperature (for example, the 260°F isotherm) the heat transfer regime is subcooled forced convection, and equations (2-89) and (2-90) are used. If the wall temperature is above saturation, a boiling component is added, using the Rohsenow correlation, so that the total heat flux is

$$q_1 h_{wlc} (T_w, -T_l) + h_{wlnb} (T_w, -T_{sat})$$
 (D-1)

At relatively low heat fluxes the functional form of  $h_{wlnb}$  reflects the small number of nucleation sites available. At high heat fluxes,  $h_{wlnb}$  is much greater than  $h_{wlc}$ ; thus area weighting is not needed in the above equation. (40)

The forced convection component of  $q_1$  is assumed to generate vapor which is released in the form of bubbles and is then recondensed, as is illustrated in figure D-2. Although the bubbles released into the fluid shrink as they recondense, new bubbles are continually being formed. It is therefore reasonable to use a mean bubble size [eq. (3-83)] in calculating the condensation rate.

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Figure D-1. Typical Isotherm Mesh and Quench Region

8140J-10
8140J-11





At a heat flux defined by

qb = py Hey abb Urbb

where  $U_{rbb}$  and  $\alpha_{bb}$  are defined by equations (3-89) and (3-90), the vapor film is assumed to begin forming. This is defined as the boundary between region I and region II.

(0-2)



the value of Q calculated in equation (D-3). If the total heat flow is greater than Q, the parameter B is reduced. This has the effect, shown in figure D-3, of reducing the integrated heat flow to the value required by equation (D-3).





8140J-12

Beyond the region III/region IV boundary, the heat transfer is calculated using Berenson's equation. The location of the quench front is defined as the isotherm with the minimum heat flux. In figure D-1, this is the 800°F isotherm. As previously mentioned, calculations are continued within the quench region at a distance of 3 inches above the quench front. Thus the 900°F isotherm heat transfer will also be calculated. Beyond the quench region, isotherm heat transfer is a value obtained by interpolating between BART nodes.

Beginning with the first isotherm within the quench region (in figure D-1, the 260°F isotherm), and ending at the last isotherm below the quench region (the 900°F isotherm), the total heat release from the rod is summed. This total heat release is then applied in a smoothly varying manner over the two BART nodes within the quench region (figure D-1) as described in paragraph 3-19.