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International Agreement Report

Dispersed Flow Film Boiling

An Investigation of the Possibility to Improve the Models Implemented in the NRC Computer Codes for the Reflooding Phase of the LOCA

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Office of Nuclear Regulatory Research
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Prepared as part of
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NOTICE

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ABSTRACT

Dispersed Flow Film Boiling is the heat transfer regime that occurs at high void fractions in a heated channel. The way this heat transfer mode is modelled in the NRC computer codes (RELAP5 and TRAC) and the validity of the assumptions and empirical correlations used is discussed. An extensive review of the theoretical and experimental work related with heat transfer to highly dispersed mixtures reveals the basic deficiencies of these models: the investigation refers mostly to the typical conditions of low rate bottom reflooding, since the simulation of this physical situation by the computer codes has often showed poor results. The alternative models that are available in the literature are reviewed, and their merits and limits are highlighted. The modifications that could improve the physics of the models implemented in the codes are identified.

CONTENTS

FOREWORD

Work related to improvement of the mechanistic post-dryout heat transfer models implemented in the modern two-fluid codes used for LWR best-estimate accident analysis has been underway for some time at the Thermal-Hydraulics Laboratory of the PSI.

Indeed, it was generally felt that the capabilities of these codes were not fully exploited; that certain correlations and other closure laws used were remnants of two-phase mixture-model calculations and therefore not really compatible with the two-fluid framework; that results obtained this way were often poor; and that certain predicted trends were unrealistic.

The particular subject of this critical-review report is Dispersed Flow Film Boiling. One observes that no real progress has been made in this area for several years now, in spite of the great number of models and publications. A reason may be that a comprehensive attack on the mechanistic modeling and a full synthesis of all available relevant information may not have been undertaken yet. The purpose of this report is to review all aspects of this modeling problem and to establish the requirements for improvements. It will become apparent that some of these (e.g. consideration of two-dimensional effects) may be beyond the capabilities of the two-fluid codes. They are included here for completeness without losing from sight the real purpose, the practical limitations, and the capabilities of the one-dimensional two-fluid codes.

The reader may find some of the criticisms raised excessive. Certain reservations about details in the assumptions imbedded in various models may be valid but not very important. The purpose here was to make an exhaustive critique without leaving out any possible shortcomings, even if these are not expected to be important. Also, some of the suggestions made for improvements may not seem to be practical. The authors are well aware of the need for a realistic and reasonable approach to the problem. On the other hand, one should attempt to find the limits of detailed mechanistic modeling, even if the complexity of the resulting models limits them to benchmark calculations.

The two-fluid models discussed here have raised the level of two-phase flow modeling significantly and have found many successful applications. This review is expected to be a critical one (in the negative sense of the word); for this reason the successes of the various models and codes examined are not stressed, but rather their weaknesses.

1. INTRODUCTION

The accurate prediction of post-critical heat transfer from the fuel cladding to the water-vapor mixture in a nuclear reactor is of crucial importance in calculating the safety margins in a hypothetical loss of coolant accident (LOCA). In both the blowdown and reflooding phases of a postulated LOCA, an important cooling effect that limits the cladding temperature excursion is the heat transfer to a high void fraction mixture, in which droplets are dispersed in the vapor stream. This flow pattern develops at void fractions higher than 80%, according to the widely accepted criterion of Groeneveld (1975) (confirmed experimentally by Kawaji et al., 1983); it is usually called dispersed or mist or liquid deficient flow: the heat transfer mode that is related to it is usually referred to as Dispersed Flow Film Boiling (DFFB).

The structure of this flow regime is highly dependent on the regimes from which it originates (Fig. 1). Since this report is focused on the reflooding phase of a LOCA in a PWR, the typical regimes that develop in such situations are of much interest. Yadigaroglu (1978) reported that at high flooding rates the quality at the quench front is very low or even negative, so that an inverted annular flow regime is established. Dispersed flow (DF) is created from the break-up of the liquid core. At very low flooding rates the quality near the quench front is high and annular flow is expected in the region upstream from the quench front. Between the position at which the critical heat flux is exceeded and the quench front location, a transition regime develops, in which liquid in the form of chunks or drops is ejected into the vapor stream. Since the low flooding rate experiments are the most difficult ones to be predicted by the codes, the main attention will be paid to the physical mechanisms characteristic of this conditions and to the models that attempt to simulate them.

Nevertheless some basic characteristics of DFFB outlined below are common to other physical situations, in which the mist flow originates directly from the dryout location (evaporator tube in Fig. 2, Collier, 1981), so that the large amount of experimental and theoretical work concerning this case will be also taken into account.

The heat transfer coefficients that are typical of the mist flow regime, even though very low if compared with those that are met in nucleate boiling, are much higher than the values that are expected in the case of vapor flowing alone.

A part of the heat that is transferred from the wall to the vapor is used for evaporating the droplets dispersed in the vapor flow, so that the temperature of the steam may remain much lower than in the absence of a liquid phase. As not all the heat that is input to the vapor can be transferred to the liquid phase, significant steam superheating builds up, starting from the point of onset of film boiling. Recent experiments performed with water at low to moderate pressure in a tube (Nijhawan, 1980; Evans et al., 1983) have provided the evidence that very significant thermal non equilibrium can be generated in dispersed flow boiling, with vapor superheats of several hundred degrees.

Minor contributions to the wall cooling are due to the direct wall-to-droplet convection and to the radiative heat transfer to the vapor-droplet mixture.

Dispersed-flow boiling is characterized not only by a noticeable thermal non equilibrium, but also by a certain mechanical non-equilibrium (Ardron, 1981; Lee, 1982): the droplets, starting from their entrainment position, are accelerated by the drag forces created by the higher-velocity steam flow. A terminal velocity is attained very far from the generation point,

so that in a large portion of the channel the velocity ratio varies (see Section 3.3.4).

The mechanical non-equilibrium has to be regarded as an important phenomenon in dispersed-flow boiling, since all the transfer mechanisms at the interface between the phases are affected by the relative velocity. The velocity of the droplets determines the concentration of the liquid phase at the different elevations, and thus the efficiency of the vapor desuperheating mechanism.

Interfacial heat transfer, as well as all other exchange mechanisms at the interface, depend on the driving force (in this case, the temperature difference) and the interfacial area. The last depends not only from the volumetric concentration of the drops N_d (drops per unit volume), but also from the size of the "average" droplet and, therefore, from the spectrum of droplet diameters.

Concentration and spectrum of the droplets are strongly dependent from their previous history and from the generation mechanisms, so that heat transfer in DFFB, even not considering other phenomena (such as inertial deposition of the droplets) can be already regarded as history-dependent.

Any model for DFFB should have at least the capability to take into account all the outlined phenomenological aspects of the associated flow regime, and from this point of view the large computer codes that are used in the nuclear safety field (such as RELAP and TRAC) have the basic capability to meet this request. Indeed, the six-equation (two-fluid) model has the possibility, in principle, to account for both thermal and mechanical non-equilibrium: if a consistent set of adequate closure laws for heat and momentum transfer mechanisms, as well as for droplet diameter distribution, could be used to close the field equations, the model should be capable of a satisfactory simulation of the phenomena involved in DFFB. Actually, a number of simplifying assumptions and the choice of questionable closure laws prevents "a priori" these codes from a faithful modelling of the physics.

These deficiencies result in a major difficulty in predicting the results of the experiments where post-CHF conditions existed. Afifi (1985), Chen (1987) and Akimoto (1987) among others present in their works large discrepancies between the calculated quench time or wall temperatures by means of the TRAC-PF1 code and the experimental data (see Figs. 3, 4 and 5) for reflooding conditions dominated by DFFB.

Analytis (1987) and Hassan (1987) reported analogous difficulties in predicting the wall temperatures in reflooding experiments by the RELAP5/MOD2 code (Figs. 6 and 7).

Even though in many cases the aforementionned researchers were able, by modifying the wall heat transfer and interfacial exchange packages, to match the experimental data within an acceptable level of accuracy, it is still doubtful whether a general solution of the prediction problem can be reached with the actual structure of the codes.

Thus, while such sensitivity studies have the merit to shed more light on the weak points of the codes, the need is felt for a more fundamental investigation of their limits due to the basic assumptions that are implicitly or explicitly contained in the field equations and closure laws.

In the following, the features of the two computer codes TRAC-PF1 (Liles et al., 1986), and RELAP5/MOD2 (Ransom et al., 1985) that are important for the modelling of the DFFB regime are discussed. Certain criticisms may be extreme but are included for completeness.

2. THE RELAP5/MOD2 AND TRAC/PF1 CODES

2.1 Basic Features

In this chapter, the main features of the models for DFFB that are implemented in the two best-estimate computer codes for the evaluation of PWR transients sponsored by the USNRC (TRAC/PF1 and RELAP5/MOD2) are outlined.

Since these codes are designed to simulate the thermal hydraulic response of each component of the primary loop of a nuclear plant under any hypothetical accidental transient, the models that are implemented result from the compromise between the wish to describe in detail the physical phenomena and the need to maintain a general validity.

Moreover, the requirement to maintain the mathematical formulations within a level of complexity that allows a numerical solution of the equations in a stable and fast fashion, is a further restriction to the implementation of very sophisticated models. For instance, in the case of a highly dispersed flow the most exact approach for developing a model would have been the one based on the Interacting Continua Assumption, that is on continuity everywhere for both phases (Ishii and Mishima, 1984), as applied by Soo (1967) to the dynamics of multiphase systems. On the other hand, an approach that postulates that the two phases are flowing in parallel with an imaginary interface separating them, seems to be the most convenient one.

In general, the effects of space and statistical (time) distribution of the phases, velocities and temperatures can become very important. Practically, the only rational approach for obtaining the macroscopic two-phase flow formulation is the application of the time averaging procedure. The mathematical structure of the field equations of both codes originates from the Eulerian time averaging and the so-called two-fluid model formulation (opposed to the mixture model). This yields a set of 6 equations that have basically the same form (Ishii, 1975) in both codes.

In RELAP5 the field equations are reduced to their one-dimensional form by area averaging.

Even though TRAC/PF1 has the possibility to use a three-dimensional component (VESSEL) in which the cross flow through the core can be evaluated, the information on the distribution of variables in the direction normal to the main flow is basically lost, since the intensive quantitites have to be represented by values that result from the average over cross sections that are orders of magnitude larger than the area of the subchannel enclosed between four rods. Thus, in both codes, with reference to heat transfer from a wall to a mist flow, the information concerning velocity, temperature and void fraction (droplet concentration profiles) is lost.

We conclude that, concerning interactions between hydrodynamics and heat transfer in the subchannels of a nuclear core, both codes adopt substantially the one-dimensional approximation.

2.2 One-dimensional Approximation

The one-dimensional approximation of the three-dimensional field equations concerns a) the convective flux terms in the momentum and energy equations,

and b) the proper averaging of the values used in the closure relationships.

a) The convective flux terms have the following basic form:

$$\nabla\cdot(\alpha_K \rho_K V_K V_K) \quad \text{(momentum flux)} \tag{1}$$

$$\nabla\cdot(\alpha_K \rho_K e_K V_K) \quad \text{(energy flux)} \tag{2}$$

where α_K, ρ_K, V_K are the void fraction, density and velocity vector of phase K, and e_K is the enthalpy (RELAP5) or internal energy (TRAC).

By area averaging, the correct form of these terms becomes:

$$\frac{\partial}{\partial z} C_{VK} \langle \alpha_K \rangle \rho_K \langle\langle V_K \rangle\rangle^2, \text{ and}$$

$$\frac{\partial}{\partial z} C_{eK} \langle \alpha_K \rangle \rho_K \langle\langle e_K \rangle\rangle \langle\langle V_K \rangle\rangle$$

where the terms in single brackets are simple area averages, while the terms in double brackets are phase-fraction-weighted mean values. The distribution parameters C_{VK} and C_{eK} appear due to the difference between the average of products of variables and the product of averages. These parameters in general depend from the flow characteristics (phase distribution, velocity profiles, etc.) as shown by Yadigaroglu and Lahey (1976).

In RELAP5 these terms, after some manipulation, take the form:

$$\alpha_K \rho_K V_K \frac{\partial V_K}{\partial z}, \text{ and}$$

$$\frac{\partial}{\partial z}(\alpha_K \rho_K e_K V_K)$$

The covariance effects are thus totally neglected and the nomenclature is simplified by dropping the double-angle-bracket averaging.

As Ishii and Mishima (1984) pointed out, in a one-dimensional model a very careful analysis of transverse distributions of variables and their effects on the field and closure equations is essential, in order to maintain the model consistent and accurate.

The distribution effects in the field equations can be taken into account by the two factors C_{VK} and C_{eK}. C_{VK}, that represents the effect of the void and momentum flux profiles on the cross-sectional area averaged momentum flux of phase K, is in most of the cases different from unity. The parameter C_{eK}, as calculated from velocity and enthalpy profiles in both developing and fully developed flows, under normal conditions (not highly transient cases) is, on the contrary, close to unity.

b) The distribution effects are not affecting only the field equations by the covariance effect discussed under a), but also appear in the averaging of the various local closure laws.

Here the important point (Ishii and Mishima, 1984) is that the averaged interfacial momentum and heat exchanges should be related to a properly weighted and averaged local relative velocity $\langle V_r \rangle$ given by

$$\langle V_r \rangle = \frac{1}{A} \int V_r dA \qquad (3)$$

where $V_r \equiv V_d - V_c$, V_d being the local velocity of the dispersed phase and V_c the local velocity of the continous phase. The interfacial exchanges should not be related to the difference between the area averaged mean velocities of the phases given by

$$\bar{V}_r = \langle\langle V_d \rangle\rangle - \langle\langle V_c \rangle\rangle$$

The difference between $\langle V_r \rangle$ and $\bar{V}_r$ can be quite large. Since in their one-dimensional form the codes necessarily use $\bar{V}_r$, significant errors can be introduced in the calculation of the interfacial exchanges.

2.3 Closure Laws

Under the name closure laws the equations used for calculating the following quantities have to be understood:

1) Interfacial drag
2) Interfacial heat transfer
3) Characteristic diameter of the droplet population
4) Wall heat transfer
5) Wall drag

In the formulation of all the closure laws the assumption is used that steady-state correlations hold also under transient conditions.

Concerning the interfacial mass transfer, both the codes use a thermal-energy jump condition that relates vapor generation to interfacial heat transfer:

$$\Gamma = \frac{-Q_{ig} - Q_{if}}{h_{g,sat} - h_f}$$

where Q_{ig} and Q_{if} are the heat transfer rates per unit volume from the interface to the gas and the liquid respectively.

The RELAP5 code distinguishes between the vapor source terms due to evaporation at the wall and evaporation in the bulk.

Under typical reflooding conditions, the pressure losses associated with dispersed flow are always small, and especially when the velocities are low, negligible pressure drops occur. Thus wall friction will not be considered in the present report. It can be sufficiently well modelled in any case.

One general characteristic of both codes is the use of flow regime maps for the identification of the limits of existence of the different regimes, in order to apply the most appropriate constitutive laws.

The RELAP5 code fixes the beginning of the mist flow regime when the void fraction is higher than 0.75, with the additional condition that the vapor velocity is high enough to suspend a liquid droplet, according to the critical velocity criterion of Wallis (1969)

$$U_{crit} = 1.4 \left[\frac{\sigma g(\rho_f - \rho_g)}{\rho_g^2} \right]^{1/4}$$

The TRAC/PF1 code also selects the constitutive laws relative to annular mist flow when the void fraction is higher than 0.75. In the code, the mist flow regime is entered when the liquid exists only in the form of entrained droplets: this is an extreme condition that requires high vapor velocities. Thus, at low vapor velocities typical of reflooding this regime is usually not entered.

It has been remarked that the selection logic of the wall heat transfer correlations is relatively independent from the flow map that is used for the selection of the closure laws for interfacial drag and heat transfer.

Special techniques of partitioning the wall heat transfer areas between the portions wetted with liquid and in contact with vapor are used in order to try to maintain consistency with the global flow regime model.

In the section concerning wall heat transfer, the consistency of the heat transfer mechanism model with the hypothesized structure of the flow will be discussed.

2.3.1 Interfacial drag

The interphase drag force is expressed in terms of the relative phase velocity:

$$F_D = - C \bar{\alpha}_f \rho_g |\bar{V}_r| \bar{V}_r$$

Independently from the form of the proportionality factor C that contains the standard drag coefficient, this expression implicitly assumes that phase and velocity distributions can be neglected, as already stated earlier.

The drag coefficient is expressed as function of the droplet Reynolds number, defined as:

$$Re_d = \frac{|V_r| d_d \rho_g}{\mu^*}$$

where the viscosity μ^* is the viscosity of the vapor in TRAC and a mixture viscosity in RELAP5. The modification of the viscosity of the continuous phase aims to take into account the additional stresses on the fluid caused by the particles, whose importance grows with their concentration (Ishii and Zuber, 1979).

The drag coefficient used in TRAC is provided by a standard set of formulas for a sphere, while RELAP5 extends the correlation suggested by Ishii and Zuber (1979) for the viscous regime to the full range of Re_d.

All the correlations employed, deduced from steady-state experiments, are applied to the transient case, by neglecting the effect of accelerating flows on the drag coefficient. Moreover, the assumption that the evaporative mass flux does not affect the drag force is used in both the correlations employed in the two codes.

The important point here is that the drag force is calculated as if a monodisperse suspension of droplets were flowing into the vapor stream, and

not a cloud composed of droplets of diameters spanning a wide range.

The droplet diameter that is considered for the calculation of the drag coefficient is the average value determined according to the following criteria:

Common:
- Monodisperse cloud of droplets
- Droplets have spherical shape
- Distribution effects are negligible
- Vaporization does not affect the drag force

TRAC:
- Interaction effects on the drag coefficient are negligible

RELAP5:
- The droplets are in viscous flow ($1 < Re_d < 1000$)

2.3.2 Interfacial heat transfer

In both codes the heat transfer rate between the superheated steam and the droplets is calculated by accounting for the processes of heat convection from the vapor bulk to the drop surface (assumed at saturation temperature) and the heat transfer from the interface to the bulk of the liquid phase.

The heat transfer coefficient on the liquid side is high enough to drive the drops to equilibrium in RELAP5, while the correlation implemented in TRAC takes into account the physical mechanisms, including the internal circulation of the liquid inside the droplet.

Since the droplets are in most of the situations at saturation, the heat transfer on the liquid side does not play any important role. More important is the way of taking into account the heat transfer on the vapor side.

In both codes the heat transfer coefficient derived by Lee and Ryley (1968) for isolated drops is used:

$$Nu_d = 2 + 0.74\, Re_d^{0.5}\, Pr_g^{0.33}$$

where the symbols have their usual meaning.

This equation was obtained by correlating data in the following range of variables:

Pressure:	1-2 bar
Superheat:	3-35 °C
Droplet diameter:	230-1126 μm
Steam velocity:	3-13 m/s.

It has to be remarked that the experimental data range is very narrow, especially regarding pressure and vapor superheat. The droplet diameters that were investigated do not cover the full range of droplet sizes that are expected to show up in most of the practical situations (e.g. reflooding). Moreover, the ability of this correlation to give correct results in the case of very small droplets is questionable (see Toknoka et al., 1982).

In the form of the equation it is also implicitly assumed that there is no effect of the evaporation rate on the heat transfer rate

One further assumption concerns the shape of the droplets: only evaporation rates from ellipsoides and spheres are in the data base of the Lee and Ryley correlation.

Moreover, the distribution effects on the global vaporization rate are neglected. This is correct as long as at least two out of the three

parameters that determine the interfacial heat transfer (interfacial area, relative velocity and temperature difference between the phases) have a flat profile over the cross section.

Finally, the whole spectrum of droplet sizes is characterized by a single representative drop diameter.

In summary, the following hypotheses are used in the model of interfacial heat transfer:

1) The functional form of the correlation of Lee and Ryley holds in a range of variables much wider than the data base from which it was originated;
2) Vaporization mass fluxes don't affect the heat transfer process;
3) Droplet concentration and velocity profiles may be assumed flat (uniform heat sink);
4) The droplets are spherical and of uniform size (representative diameter);
5) The interaction between the droplets is negligible.

2.3.3 Droplet diameter

As mentioned before, interfacial transfer modelling requires an average droplet diameter. In both codes this is not calculated on the basis of the physical mechanisms that affect the droplet size distribution (evaporation, aerodynamic break-up, break-up induced by spacer grids and impingement on the wall, coalescence), by following the history of the droplet population since its generation. Instead the average diameter is derived from a local stability criterion. The droplet size depends on the local conditions, namely relative velocity and fluid properties and is calculated from the local critical Weber number

$$We = \frac{\rho_g V_r^2 d}{\sigma}$$

In this fashion the number of droplets changes without any relation to the actual mechanisms that cause this modification, and the average diameter increases above the quench front, since the relative velocity initially decreases, as the liquid is accelerated. The consequence is that the interfacial area decreases very sharply, with an unphysical very rapid degradation of the heat sink.

A more general assumption is made implicitely by the use of a single representative droplet diameter: the hydrodynamics of the droplet flow can be described by using the coordinates of the center of mass of the liquid phase. Without this basic assumption the dispersed flow should be described by several momentum equations, one for each group of droplets whose momentum falls into a range defined by discretizing the continous distribution.

In summary, the codes use the following simplifications:

1) The droplet hydrodynamics can be described in a sufficiently accurate way by considering the motion of the center of mass of the liquid phase.
2) The droplet average size depends only from the local conditions.
3) Break-up and coalescence of the droplets are neglected.

2.3.4 Wall heat transfer

The wall-to-fluid heat transfer process in DFFB is described by taking into account the following three contributions: convective heat transfer to the vapor, convective heat transfer to the liquid (drops) and radiative heat transfer.

Radiative heat transfer In TRAC only the radiative flux to the liquid, while in RELAP5 a complete radiation energy exchange network between wall, vapor and liquid are considered. In both codes the assumption is used that the mixture is optically thin, so that any portion of the fluid exchanges radiation directly with the boundary surfaces.

Convective heat transfer It is not possible here to give the details of the convective heat transfer models in the two codes, but only the most important aspects will be reviewed.

Both codes partition the wall heat flux between the two phases; two heat transfer coefficients are calculated for the heat transfer wall-to-liquid and wall-to-vapor. For the calculation of the contact area, RELAP5 distinguishes between reflooding and other physical situations. In conditions other than reflooding the two areas are simply proportional to the volumetric fractions in the adjacent cell. In the case of reflooding, the total wall heat transfer area is considered available for contact with both the phases, and the partitioning is implicit in the definition of the heat transfer coefficients.

The TRAC code uses a similar technique.

It is important to notice that the way to partition the convective heat flux to the two phases depends only on the average values of the variables in the fluid cell and the wall temperature, and not on parameters that account for the phase concentrations locally near the wall (e.g. droplet mass flux to the wall), that can be derived only from the consideration of the hydrodynamics of the dispersed phase. The analysis of the consistency of the whole heat transfer package is beyond the scope of the present report.

The only remark that is worthwhile to make is that in both codes the heat transfer rate to the liquid phase in film boiling is calculated by using a Bromley-type correlation, that accounts for heat transfer to the bulk liquid by conduction through the vapor-film. While this choice reflects the apparent similarities between film boiling in a pool and inverted annular flow, it is completely arbitrary for dispersed flow, when only a small portion of the entrained liquid phase interacts with the wall.

Even more questionable is the use of the Forslund-Rohsenow equation together with the Bromley correlation in TRAC. The Forslund-Rohsenow correlation, as pointed out by Afifi (1985), was originally developed to account for an additional heat transfer mechanism at low quality and high mass flux, and it should give the total heat flux to the liquid.

This heat transfer rate calculation method is surely imposed by the need to account for a wide range of post-CHF conditions without using specialized models. It is clear, however, that the above mentioned characteristics make the heat transfer packages more suitable for describing inverted annular film boiling rather than DFFB.

The heat transfer rate to the vapor is calculated by using the Dougall-Rohsenow correlation (1963), which has the same form of the Dittus-Boelter correlation, that is strictly valid for single phase turbulent flow at low wall-to-bulk temperature ratio: the modified form (Dougall-Rohsenow) uses a Reynolds number that is corrected to reflect the volumetric flow rate of

the two-phase mixture. Thus use of this last correlation in DFFB implies that the structure of the flow typical of single phase forced convection is not modified by the presence of the dispersed phase, so that the heat transfer rate to the vapor is the same as if it were flowing alone in the channel.

It has also to be taken into account that the Dittus-Boelter equation is valid for fully developed turbulent flow, that is, at some distance from the entry cross section ($L/D > 20$). In case of DFFB the entrance point can be identified as the CHF location, where a continuous vapor film along the channel wall starts to develop. A further assumption is that a correlation valid for fully turbulent flow ($Re > 20\,000$) holds at the low Reynolds numbers typical of reflooding.

Moreover, as already noted earlier, all the correlations were derived from steady-state experiments; this constitutes standard practice: the transient characteristics of the flow are assumed not to affect the heat transfer processes to a large extent.

Finally, the effect of the spacer grids on the structure of the flow and heat transfer is completely neglected.

In summary, it is assumed that:

1) The heat flux partition to the two phases depends on the average flow parameters in the fluid cell and the wall temperature.
2) The heat flux to the liquid phase can be calculated on the basis of the Bromley model.
3) Heat transfer to the vapor is not affected by the presence of the dispersed phase.
4) The flow is fully developed.
5) The flow is fully turbulent.
6) The mixture is optically thin.
7) The effect of spacer grids is negligible.

2.3.5 Final remarks

The heat transfer mechanisms that are taken into account by the codes (wall to liquid, wall to vapor and vapor to liquid) put the DFFB models into the category of the so-called three step models.

The models may be also defined as mechanistic, since the phenomenology is fully accounted for, and the only amount of empiricism is due to the use of correlations for describing the elementary mechanisms of heat, mass and momentum exchange.

The limits of the models are coming from a) some of the assumptions that have been highlighted in the previous sections, and b) from the improper use of some correlations.

In the following only the first of the two categories of possible sources of inaccuracy will be discussed in detail; the validity of every assumption will be discussed on the basis of the experimental evidence or the most recent studies.

3. PHENOMENOLOGICAL ASPECTS OF DFFB

3.1 General

The phenomenology associated to DFFB is quite complex since the coupling of the two phases affects the mechanisms of heat and momentum exchange.

The dispersed phase has flow characteristics (size of the droplets, velocity components) that are statistical in nature and are strongly dependent on their initial values at the point of generation of the droplet cloud.

The turbulence intensity and transport properties of the continous phase are modified by the presence of the liquid particles, to an extent that is depending from the concentration, distribution and spectral composition of the dispersed phase. On the other hand, the motion of the droplets and their heat, mass and momentum exchanges are strongly dependent on the flow characteristics of the continuous phase. Additional complication is brought from the transient nature of the phenomena investigated and from the lack of a well established criterion for the boundary between the laminar and turbulent regimes for a flowing two-phase mixture.

Because of the complexity of the phenomena involved, only some aspects that have been studied most extensively will be treated in detail, and the alternatives to the approach used in the codes will be examined.

3.2 Inadequacy of the One-dimensional Approach

The basic difficulties that stem from the averaging of the conservation equations over a cross sectional area were already mentioned: not only the distribution coefficients C_{VK} and C_{eK} should be calculated, but also the closure laws should employ properly averaged values of the flow variables, that is, the radial profiles have to be guessed in advance. For the calculation of these coefficients, empirical formulas derived from the volumetric-flux-distribution parameter C_o of the drift flux model have been proposed (Ishii and Mishima, 1984).

A major difficulty arises when the effects of phase distribution on closure relations have to be taken into account. In the following, the two examples of interfacial drag and heat transfer are discussed briefly, while the two-dimensional effects on wall heat transfer will be discussed in Section 3.2.5.

Based on the drift flux formulation, Ishii and Mishima (1984) showed that the average relative velocity $\langle V_r \rangle$ (defined from Eq. (3) in Section 2.2) that must appear in the closure laws for bubbly-flow may be approximated (when V_r is much smaller than the phase velocities or is nearly uniform over the cross section) by

$$\langle V_r \rangle \approx \frac{1 - C_{od}\langle a_d \rangle}{1 - \langle a_d \rangle} \langle\langle V_d \rangle\rangle - C_{od}\langle\langle V_c \rangle\rangle \qquad (4)$$

where C_{od} is the distribution coefficient of the dispersed phase and $\langle\langle V_d \rangle\rangle$, $\langle\langle V_c \rangle\rangle$ the average velocities of the dispersed and continous phase, respectively.

Even though in principle one could derive a similar expression for dispersed droplet flow and the possibility of taking into account the two-dimensional (distribution) effects through the factor C_{od} is very interesting, adequate values of C_{od} for all the possible conditions of mist flow are required.

As shown by Lee and Durst (1982) the velocity distribution of monodisperse particles and of the carrying gas are dependent on the size of the particles (Fig. 8), as long as the particles are small.

The experiments of Lee and Durst were conducted in an adiabatic pipe, so that the results cannot be fully applied to dispersed flow in heated channels, but their measurements show that $\langle V_r \rangle$ can be affected by the droplet size through the distribution coefficient C_{od} that depends on the velocity profile.

Since no drift flux correlations have been proposed that include the effect of the droplet diameter, a correction of $\langle V_r \rangle$ based on the drift flux model is not readily available.

A further complication is due to the fact that the particle concentration profile also affects C_{od} (Zuber and Findlay, 1965). As the data of Hagiwara et al. (1980) for adiabatic pipe flow show (Fig. 9), the droplet concentration starts falling in the turbulent core and goes to zero near the wall. The concentration profile is likely to be influenced by the presence of the hot wall also (see 3.2.2).

In conclusion, it is clear from Eq. (4) that $\langle V_r \rangle$ cannot be calculated unless the concentration profile is known.

The concentration profile affects the average drag force not only through its effect on the average relative velocity, but also because the interfacial area concentration depends on it.

The average interfacial heat exchange and the average evaporation rate depend somewhat less from the average relative velocity (since they are proportional to $Re_d^{0.8}$) but are linearly proportional to the average area concentration and the vapor temperature.

Since the temperature profile is modified according to the distribution of heat sinks (droplet concentration profile), it will probably not be flat in the turbulent core, so that the heat sink intensity cannot be calculated by simply multiplying an average temperature difference times the average interfacial area concentration. Thus, only by a two-dimensional analysis of the vapor temperature field it will be possible to account properly for the distributed heat sink.

From these two examples it is already clear that the simulation may be uncertain, if no proper distribution parameters are available, and when the average of the product of variables is substituted by the product of the average values.

3.3 On the Closure Laws Implemented in the Codes

Various assumptions affecting the appropriateness of the closure laws used in the codes are discussed in this section. The subsections of this section correspond to those of Section 2.3.

3.3.1 Interfacial drag

The drag coefficient is calculated by formulas based on steady-state experiments. As Temkin and Mehta (1982) pointed out, the only situations where small spheres are known to move steadily occur when they achieve their terminal velocity in a stagnant fluid or when they are fully carried by a steadily moving fluid. In the actual conditions prevailing in mist flow neither situation occurs.

These authors have experimentally shown that the drag coefficient can be sensibly higher in the case of accelerating flow ($d\langle V_r\rangle/dt > 0$) and lower for decelerating flow ($d\langle V_r\rangle/dt < 0$), and were able to correlate the actual drag with the standard steady-flow value by including a functional dependence on the non-dimensional group

$$\frac{d}{\langle V_r\rangle^2}\frac{d\langle V_r\rangle}{dt}$$

The effect of such a correction could be a few percent. If one assumes that the motion of the droplets can be described in terms of one momentum equation applied to the center of mass of the dispersed phase, the drag force per unit volume can be expressed as (Ishii and Mishima, 1984)

$$\frac{a_d F_D}{B_d} = -\langle a_i\rangle\left[\frac{C_D}{4}\left(\frac{r_{sm}}{r_D}\right)\rho_g\frac{\langle V_r\rangle|\langle V_r\rangle|}{2}\right]$$

where a_i is the interfacial area concentration, r_{sm} and r_D are, respectively, the Sauter mean radius and the mean "drag radius", defined as:

$$r_D = \frac{3B_d}{4A_d}$$

where B_d is the volume of a typical particle and A_d is the area perpendicular to the flow. Kataoka and Ishii (1983) showed that for spherical particles the shape factor, $\frac{r_{sm}}{r_D}$ is equal to 1 regardless of the particle size distribution.

The assumption of spherical particles must be examined next. According to Clift et al. (1978), a droplet of equivalent diameter d_e is spherical if the Eotvos number ($Eo = g\Delta\rho\, d_e^2/\sigma$) is less than 0.4. If this criterion is applied to a system of water and vapor at atmospheric pressure, for $d_e \approx 1$ mm, we find Eo=0.17. Thus, at least when the acceleration forces are not very intense, the shape factor can be assumed to be close to unity.

According to a criterion proposed by Ishii and Zuber (1979), the distorted fluid particle regime is characterized by a viscosity number N_μ

$$N_\mu = \frac{\mu_g}{(\rho_g\sigma\sqrt{\frac{\sigma}{g\Delta\rho}})^{0.5}}$$

higher than a critical value N_μ^*

$$N_{\mu}^{*} = 0.11 \left[\frac{1+\psi}{\psi^{8/3}} \right]$$

with

$$\psi = 0.55 \left\{ \left[1 + 0.08 \frac{r_D^3 \rho_g g \Delta\rho}{\mu_g^2} \right]^{4/7} - 1 \right\}^{0.75}$$

According to this criterion the presence of distorted fluid particles during reflooding should be excluded, at least at distances from the QF where the droplet population has been stabilized. Under such conditions the drag coefficient for the viscous regime (spherical particles) should be appropriate.

Just above a quench front, however, large chunks of liquid are ejected into the vapor stream and are accelerated until break-up occurs. In this region, the drag force should be better calculated using drag coefficients appropriate for the distorted-particle regime (Harmathy, 1969), where C_D depends only on the particle radius and fluid properties, i.e $C_D = \frac{4}{3} r_D \sqrt{g\Delta\rho/\sigma}$ (the equivalent of Newton's regime for solid spheres).

The <u>effect of mass efflux</u> (evaporation or condensation) on the drag coefficient has been investigated among others by Chuchottaworn et al. (1985), who found a dependency on the <u>mass transfer number</u> B_M

$$B_M = \frac{Re_p Sc}{Sh(1-x)}$$

where Sc and Sh are the Schmidt and Sherwood numbers, respectively.

The main shortcoming of using the standard drag coefficient laws is due to the fact that they use mostly the results of measurements based on stationary single particles in a <u>laminar stream</u>, rather than considering a cloud of particles moving <u>with a rather small relative velocity</u> in a higher-velocity turbulent gas flow. The analysis by Lee (1987) of experimental data of Lee and Durst (1982) and Tsuji (1984) demonstrates that particle interactions modify sensibly the dependency of C_D on the Reynolds number in the highly dispersed gas-particle viscous regime. For large particles and large concentrations, the drag coefficient becomes larger, but for small particles and small cooncentrations the reverse is true.

An apparent viscosity correction proposed by Ishii (1977), that takes into account only the particle volumetric concentration, does not account sufficiently for the observed modification (up to an order of magnitude) of the drag coefficient. A relation that takes into consideration also relative velocity and particle size has been proposed by Lee (1987), only for high void fractions (>0.992). Generally theoretical and experimental studies of polydisperse clouds of particles are lacking.

3.3.2 Interfacial heat transfer

The possible insufficiency of the correlation of Lee and Ryley (1968) for <u>heat transfer to droplets in steam</u>, under conditions that are typical of most post-CHF regimes, is mostly due to the lack of high vapor superheating in its data base. As showed theoretically by Hoffmann and Ross (1972), and

confirmed by the experiments of Yuen and Chen (1978), in the range of high Reynolds numbers and high evaporation rates, the mass efflux from the droplet reduces convective heat transfer from the superheated steam. This shielding effect is expressed in terms of the Spalding number

$$B = \frac{c_p (T_g - T_{sat})}{h_{fg}}$$

but the functional dependency of the Nusselt number from B is somewhat controversial (see Harpole, 1979, for further details).

The assumption of uniform droplet concentration has already been discussed in relation to the covariance in the expression of the average heat transfer rate in section 2.3.2. Here it is worthwhile to underline that, while this hypothesis is very convenient, since it allows to simplify considerably the mathematical formulations, the only experimental evidence that is available to justify it is due to Cumo (1973). In his experiments performed with Freon-12 the transversal distribution (measured by dividing the channel in only 6 zones) was measured for $p/p_{crit} > 0.3$, quality > 0.6, mass flux > 280 kg/m^{2}s and average droplet diameters smaller than 50 μm. This range does not cover either the typical conditions of dispersed flow during reflooding ($p = 1$ to 4 bar, $G < 200$ kg/m^{2}s), nor the voidage conditions in most other practical applications. Indeed, up to intermediate pressures of 70 bar, dispersed flow is present already for $x > 0.2$.

The average droplet sizes reported in the literature are larger than those observed by Cumo (e.g. Wong, 1980; Seban, 1980). As can be learned from the equation of motion, the dimension of the drop is crucial in determining its trajectary and therefore its radial position. The coarse partition used by Cumo to investigate the transverse concentration profile (6 zones) does not allow the evaluation of the concentration near the wall which is very important since it influences the temperature profile (local desuperheating), and thus, the heat transfer rate. Therefore the experiments by Cumo cannot definitely answer the question whether any concentration profile develops along a heated pipe.

Notwithstanding this observation, there is, however, general agreement on the fact that the axial velocity of the dispersed phase is radially nearly uniform (see Section 3.3.3).

Any reduction of the total evaporation rate due to overlapping boundary layers around the droplets in a cloud is not well established. Recent studies on the evaporation rates of dense sprays of fuel by Bellan and Harstad (1987) showed that the evaporation time of closely packed fuel particles is weakly dependent on the relative velocity between the cluster of drops and the carrying gas. It is postulated that what controls the evaporation rate is the difficulty of penetration of the outer flow through the dense cluster: at very high droplet concentration, the outer flow bypasses the cluster of drops and only the drops at the periphery "feel it", while the drops at the center evaporate at the rate typical of spheres in a quiescent fluid. The development of this research is still in an early phase and no definitive conclusion or models are available. It appears, however, that sensible reductions of the evaporation rate with respect to the single-drop model prediction may occur when the droplets are not highly dispersed. Indeed, theoretical studies by Labowsky (1978) in the range of very low Reynolds number ($Re_p < 1$) have shown that even when spacing between the droplets is five diameters (corresponding to a void fraction higher than 0.99), a 30% reduction of the evaporation rate is to be expected. The effect of interaction at higher Re_p is quite controversial.

O'Rourke (1981) used in his model an augmentation factor, obtained by a thorough analysis of several sets of experimental results. On the contrary, in a review paper on evaporating sprays, Faeth (1983) concludes that the effect of adjacent drops on vaporization rates is practically negligible. It is evident that more experimental work is necessary to shed light on this aspect of the interaction between the droplets.

3.3.3 Droplet diameter

It has been already remarked that all the closure laws are affected by the size of the droplets, since, according to the particle diameter, the same liquid volume may produce very different interfacial areas.

The characterization of polydisperse sprays of droplets by an average diameter would have been justified only if neither the axial velocity nor the concentration depended from the radial coordinate or from the size of the droplet (effect of averaging of products of variables). Since, as mentioned above, the issue of the radial concentration profile is open, the use of an average diameter in the expression for the interfacial area is highly questionable. The total interfacial area at any elevation along the channel can be anyway properly calculated by using a characteristic droplet diameter, if the axial velocity is not strongly dependent on the size of the droplets.

Measurements of droplet velocities in tubes (the only experiments that offer results free of bundle wall effects) for annular-mist flow (Wilkes, 1983) and dispersed flow above a quench front (Ardron and Hall, 1981) show that small and large droplets travel axially at approximately the same velocity, close to the terminal velocity of the droplets of average size. A possible explanation for this lies in the different radial migration of small and large droplets: the small drops are carried by the turbulent eddies toward the wall where they loose part of their axial momentum, while the large ones, less affected by the turbulent eddies, are continuously accelerated. Momentum exchanges due to collisions may also contribute to render uniform the velocities of the droplets. Therefore, as first approximation, a momentum equation written for the center-of-mass of the liquid phase should give correct information about the average axial velocity and the distribution of the droplets in axial direction.

The droplet size spectrum is difficult to define, since it is dependent upon the droplet generation mechanisms. The proposed distribution function forms are mostly derived from experiments in which the atomization of liquid by air streams was studied (e.g. Nukiyama and Tanazawa, 1938). Implicit in this approach is the dependence of the size distribution from the mechanisms that are responsible for the fragmentation. The maximum diameter is usually calculated using a critical Weber number.

Such fragmentation criteria, according to the studies of Kocamustafaogullari et al. (1983) and Kataoka et al. (1983), overestimate the droplet size: in some experiments it has been observed that downstream from the quench front the drops were too small to have been generated by the standard droplet disintegration mechanism. Therefore the majority of droplets must have been generated at their time of entrainment and not during their flight as droplets.

At gas velocities beyond the inception of entrainment in annular flow (Kataoka et al., 1983), the droplets are generated by the shearing off of roll waves. The study of such a mechanism has yielded a correlation that fits the data of De Jarlais (1983) in IAFB quite well. At lower gas fluxes, like in the case of a slowly advancing quench front, the maximum stable droplet size may be estimated (Kocamustafaogullari et al., 1983) by

considering the relative velocity for churn turbulent flow, as calculated using the criterion of Wallis (1969), referred to in Section 2.3. Most of the data obtained in the FLECHT-SEASET experiments (Lee et al., 1982) could be correlated by the relation (Kocamustafaogullari et al., 1983):

$$d_m = 4 \left[\frac{2\sigma}{g\Delta\rho} \right]^{1/2} N_\mu^{1/3}$$

where N_μ is the viscosity number defined in Section 3.3.1.

A different droplet generation mechanism has been observed in single tube reflooding experiments in glass test sections (Ardron and Hall, 1981). Here very large liquid globules were generated at the quench front, by disintegration of waves formed in the wetted portion of the tube (Fig. 10). A stable droplet size distribution was observed quite far from the generation point.

Following their generation, the population of droplets undergoes evaporation, break-up and coalescence.

In summary, the experimental evidence suggests that the mechanisms of drop generation and break-up should be considered separately; this is not the case in the computer codes.

Moreover, aerodynamic break-up is not the only cause of fragmentation of the droplets, since collisions with the wall and with the spacer grids and among droplets are also effective in changing the droplet size spectrum. Since the various fragmentation processes depend on different critical values of the Weber number, it is clear that the drop stability limit changes along the flow path according to the predominance of the various mechanisms.

The ERSEC experiments (Juhel, 1984) showed an increasing droplet number flux downstream from the quench front. A break-up mechanism is considered responsible for this trend. As shown by Krzeczkowski (1980), the mechanisms of droplet deformation and disintegration as well as the duration of the break-up depend on the Weber number, the Laplace number ($La = \rho_f \sigma d / \mu_f^2$), and the ratio μ_f / μ_g.

Practically only the Weber number is important. Sarjeant (1978) has conducted the most extensive and systematic study on the break-up mechanisms. He found that the critical Weber number is practically independent from the drop Reynolds number, and depends on whether the droplet is suddenly or gradually exposed to a gas stream. Break-up time and number of fragments depend on the Weber number. The size distribution of the fragments has been studied, among others, by Podvysatsky and Shrayber (1984).

Coalescence of drops certainly takes place immediately downstream of the quench front, due to the chaotic motion of the entrained droplets. Ardron and Hall (1981) report that collisions and disintegration processes were still taking place 1 m above the quench front. Clare and Fairbairn (1984) observed that the Sauter mean diameter increased with elevation above the quench front and presumed that drop coalescence was the cause of it. Few theoretical studies and no experimental data were found, concerning collision rates and coalescence mechanisms. The most complete model known to the authors has been developed by O'Rourke (1981) for Diesel engine sprays: advanced statistical methods are used to calculate the local collision and coalescence rates. Simpler mechanistic models can be successfull in situations where the turbulent diffusion is the mechanism responsible for collisions (e.g., Williams and Crane, 1984).

The fragmentation associated with impact of drops onto the spacer grids has been studied in recent years (Lee, for example, Adams and Clare, 1984). A simple break-up model has been presented by Yao et al. (1988).

3.3.4 Wall heat transfer

In the post-CHF regime there is no stable contact between the wall and the liquid, and the partition of the wall contact area between liquid and gas is therefore quite ambiguous. In reality, immediately above a quench front, a very intense droplet flow to the wall is capable of removing a remarkable amount of heat, by strong evaporation with the superheated thermal sublayer or, if the temperature of the wall is lower than a limiting temperature, by direct contact with the hot wall. There is no general agreement regarding this limiting temperature; the highest value found in the literature is 870°K (Nishio and Hirata, 1978). The two heat transfer mechanisms, that according to the Iloeje (1975) terminology can be identified as "dry" and "wet" contacts, are the two main contributors to the enhancement of the heat transfer rate just above the quench front. The relative importance of the wet contacts decreases, obviously, with increasing wall temperature. The recent experimental finding of Yao and Cai (1985) that the wettability of a surface depends not only from the velocity normal to the surface, but also from the tangential component should also be mentioned: therefore the heat transfer effectiveness to a mist flow depends strongly on the hydrodynamics of the droplets.

The hydrodynamic behavior of the droplets above the quench front is a chaotic one, and a mechanistic model that would attempt to calculate the interaction of the liquid phase with the wall needs the droplet size distribution and the transverse momentum distribution. As the details of the generation mechanisms of the droplets are not known up to date, the present approach for taking into account the high heat fluxes immediately above the quench front is highly empirical. The most popular method is to use the highest value of the HTC between that calculated from a transition boiling correlation (e.g., Weisman, 1981) and that calculated from a film boiling correlation (Bromley-like). Alternatively the heat flux can be directly correlated with the distance from the quench front (e.g., Yu and Yadigaroglu, 1979).

Further downstream of the quench front or in the post-CHF zone of an evaporator tube, the droplets lose their initial transverse momentum. In this region, a mechanistic approach for the calculation of the contribution of the dispersed phase to the total wall heat flux must take into account primarily the turbulent deposition. Here the droplets can be transported towards the wall only by turbulent diffusion, and the possibility of reaching the wall is determined by the force balance on the drop. On this basis, Lee and Almenas (1982) have calculated that, in the typical ranges of variables that are of practical interest, the droplets having diameters $> 20\ \mu m$ will not impinge on the surface, unless external forces are applied (e.g. interaction with flow retrictions like spacer grids in a rod bundle). If this result is accepted, all the convective heat flux should go to the vapor. Any Bromley type film boiling correlation is also inapplicable, as the heat transfer process is different from a conductive one through a developing vapor film.

The effectiveness of the presence of the droplets is due to two causes: a) the temperature of the vapor near the wall is reduced because of the strong vaporization of the droplets flying in the thermal boundary layer, and b) the structure of the hydrodynamic boundary layer is modified, with an effect on the thermal boundary layer as well.

The first effect is quite evident, since the migration of droplets towards

the wall, with consequent evaporation in the zone of high vapor temperature, reduces the temperature near the wall and increases the driving temperature difference between the wall and the vapor. In this respect the calculation of the radial droplet concentration profile is crucial: it can be assumed that the droplets undergo a diffusive process in the radial direction, by receiving transverse impulse in the turbulent core and penetrating by inertia in the viscous sublayer, where the force balance controls their motion (see, e.g. El Kassaby and Ganic, 1984).

In order to explain the second contribution, it is convenient to separate the enhancement of the total heat flux due to the evaporation of the droplets from that due to the modification of the heat transfer coefficient to the vapor, because of the above mentioned hydrodynamic effect. In this respect the experimental evidence concerning the modification of the heat transfer coefficient in upflow gas-solid mixtures compared to pure gas flows can be directly transferred to the steam-water mist flow. As Depew (1979) pointed out, a dispersed phase alters convection in several ways:

1) Because of the penetration of the particles in the viscous sublayer, the sublayer becomes thinner.

2) In the core of the turbulent flow, the convective eddy motion is dampened (this mechanism hampers heat transfer).

3) Slip between particles and gas enhances the mixing of the carrier gas.

4) Radial motion of the solids promotes energy exchange between the laminar sublayer and the turbulent core.

All these effects contribute to the alteration of the viscous sublayer thickness and the slope of the gas velocity profile, which control heat transfer in convective flow. Theofanous (1982) has experimentally shown that the structure of the turbulence is strongly affected by the dispersed phase, and Tsuji et al. (1984), by employing LDA techniques that allow the measurement of the radial profiles of the gas and particle velocities, have demonstrated that the gas velocity profile flattens or even becomes concave in the center of the pipe and becomes steeper in the region near the wall, when 500 μm diameter particles are added to the air flow (see Fig. 11). Such an effect was not observed for larger particles but this can be explained by the low loading ratios (i.e. for ratios of mass flow rates of particles to gas < 1).

According to the respective effectiveness of mechanisms 2 and 1,3,4 the heat transfer coefficient HTC may be reduced or increased.

The experimental findings are:

a) According to the ranges of particle sizes below (Shrayber, 1976), and for a density ratio of $\rho_p/\rho_g \approx 10^3$:

$d_p < 30$ μm: only small reductions of the htc are observed for loading ratios (m) lower than 1. For higher m substantial enhancements (up to 500 %) are possible.

$30 < d_p < 500$ μm: The enhancement factor decreases with increasing particle size. Substantial reductions of the htc are observed up to loading ratios of 5 to 10. This reduction may be due to the transition from turbulent

to laminar region (Hasegawa et al., 1983). Slight improvement of the HTC with respect to the single-phase case are possible at higher m.

d_p > 500 μm: Remarkable reductions (at low m) and large enhancements (at high m) are characteristic of this particle size. A collision mechanism is postulated to be responsible for the radial migration of the largest particles.

The minimum value of the HTC is obtained when the characteristic dimension of the particle is comparable to the thickness of the viscous sublayer in the undisturbed flow of the gas alone (Jepson, 1963).

b) The HTC enhancement is reduced at high Reynolds numbers (40 000 to 70 000 in air) and at very high values (10^5) the HTC monotonically decreases for any m (Boothroyd and Hague, 1970).

c) The enhancement is higher for larger diameters of the pipe (Boothroyd and Hague, 1970).

d) The enhancement increases with the ratio between the absolute temperatures of the wall and the gas (Sukomel, 1979).

e) With sufficiently high loadings, the presence of particles results in an increase of the thermal entry length up to 40 - 50 diameters (Kianjak, 1984).

Therefore, the ratio of the HTC in dispersed flow and in single phase flow depends on the loading ratio (quality), droplet size, Raynolds number, hydraulic diameter and wall temperature. To the author's knowledge no empirical correlation has been proposed that takes into account all these parameters.

In an attempt to correlate the heat transfer enhancement in some reflooding experiments, Drucker et al. (1984) proposed that the enhancement factor has to be function of the dimensionless group $\mu Gr/Re^2$, but the large scatter of data in a plane that uses this number as independent variable (see Hassan, 1986), clearly indicates that the correct parameters that control the phenomenon have not yet been identified (Fig. 12).

All the considerations that have been made until now refer to a fully developed turbulent flow. The physical picture is somewhat more complicated.

At the low Reynolds numbers (4 000-10 000) typical of low reflooding a transitional flow is likely to occur. As the experiments show (Lawn, 1969; Gnielinski, 1976) the wall heat flux drops below that calculated by any Dittus-Boelter type correlation (based on Reynolds analogy) at Reynolds numbers lower than 20 000. The use of the Dougall-Rohsenow correlation for the heat transfer to vapor in DFFB may lead to a significant error. Moreover the flow in DFFB is never fully developed since a continuous evaporation process modifies all the flow parameters, including the Reynolds number. It is reasonable to assume a quasi-fully developed flow when the temperature and velocity profiles have reached their characteristic shapes in turbulent flow.

In this respect two different situations are expected. In the case of the post-CHF regime downstream from the dry-out point in an evaporator tube,

the velocity distributions in the liquid annulus and in the vapor core have already developed profiles, so that it can be assumed that the vapor velocity profile downstream of the thermal crisis point is already developed. This is also the case when a high quality mixture is present at the quench front, that is when annular-wavy flow occurs upstream of the quench front position.

The development of the thermal boundary layer, on the other hand, starts from the dryout point, where the vapor temperature is uniformly equal to the saturation temperature (Koizumi, 1979). It is believed that the development of the thermal boundary layer will not be affected very much by the presence of the droplets, as long as they are highly dispersed.

A different physical picture is expected when churn-turbulent flow is present at the quench front. In this case both the velocity and temperature profiles have to develop, and the influence of the turbulence created by the droplets is likely to be very important. However, in the zone immediately above the quench front, the heat transfer by convection to the vapor is not the predominant mechanism, so that a large error in the prediction of the thickness of the hydrodynamic and thermal boundary layers cannot affect very much the total heat flux. Downstream of this highly turbulent zone, when the droplets have lost their initial transverse velocity, it can be assumed again that the boundary layers grow at the same rate as in single phase flow.

The assumption of optically thin mixture used in the models for radiative heat transfer holds only at low pressures (p = 1 bar) and very high void fractions (> 0.99), when the typical geometries of the nuclear cores are considered (D = 10-13 mm).

In Fig. 13 some calculations with a model that employs the optically thin medium assumption (Sun et al., 1975) are presented together with the results obtained by Deruaz and Petitpain (1976) by means of a more sophisticated approach. It is easily recognized that a discrepancy up to 100% is possible in the most unfavorable cases.

The enhanced mist cooling downstream of grid spacers is an important heat transfer mechanism, especially during reflood emergency cooling (Lee et al., 1984). The spacer grid can enhance the fuel rod heat transfer by four mechanisms (Yao et al., 1988): direct radiation from the fuel rods, thermal boundary layer separation and reattachment, spacer grid early rewetting, and break-up of droplets in smaller fragments. Overlooking the presence of the spacer grids results in overprediction of the cladding temperatures.

The last assumption that is always made in any model for DFFB is that the heat transfer coefficients that are measured under steady-state conditions, are applicable also to transient cases, provided that a time-averaging is applied to the flow variables. Actually during the reflooding phase cyclically dispersed flow will take place, due to the discontinuous water entrainment near the quench front. Ghazanfari (1980) studied experimentally unsteady dispersed flow under post-dryout conditions. The experimental results, in the range of low pressure, low mass flux and high quality (>0.5), showed that at equivalent mean vapor quality no noticeable differences in heat transfer exist between a steady-state and a cyclically dispersed flow. These results, relevant to reflooding conditions, do not cover, however, the entire range, and especially data in a lower quality range (0.1 to 0.5) should be necessary to verify the non-susceptibility of heat transfer upon cyclic behavior.

3.3.5 Summary of the deficiencies of the DFFB models implemented in the codes

It has been shown that one of the main shortcomings of the models implemented in the codes is the one-dimensional approximation that does not allow to calculate properly the complex mechanisms of heat and momentum transfer between the phases, since the cross-sectional average values are strongly dependent on the distribution parameters.

The fundamental role played by the droplet size distribution upon the thermal-hydraulics of a dispersed mixture is not properly accounted for, since simplistic criteria based on the local flow variables, are used to calculate the average droplet diameter: generation mechanism and flow-history dependency are completely neglected.

The wall heat transfer package uses heat transfer coefficient correlations that do not consider all the physical processes taking place in DFFB, and the heat flux partitioning between the two phases is defined according to questionable empirical criteria. Moreover, the influence of the dispersed phase on the heat transfer coefficient to the vapor is usually ignored. A full representation of the physical mechanisms that control the heat transfer from the wall to the mixture is possible only within the frame of a <u>two-dimensional</u> approach.

4. MECHANISTIC MODELS FOR DFFB

Reviews of the mechanistic models proposed in the last 20 years are given by Mayinger and Langner (1978), Webb and Chen (1982), and Kawaji (1984). In the following only the most original aspects of the different models, that may constitute a substantial progress with respect to the approach used in the codes, will be highlighted.

The models will be classified in two groups, which, for convenience, will be referred to as local and integral, in relation to the way they consider the dependency of the heat transfer processes on flow history: the former adopt assumptions and/or consider mechanisms concerning the local values of certain variables (typically the relative velocity and the droplet diameter). Integral models consider the evolution of all the flow variables, starting from the point at which the initial values are given (typically at the dryout point).

4.1 Local Models

4.1.1 Models considering the details of wall-droplet interaction

Iloeje et al. (1975) developed the first mechanistic model for the contribution of the droplets to the total heat flux. They distinguished three heat transfer paths: heat transfer from wall to liquid drops that reach the wall (wet contact), from the wall to droplets that have no sufficient transverse momentum to penetrate the thermal boundary layer (dry contact), and from wall to vapor. The range of validity of the model extends all the way from dry-wall film boiling to transition boiling. Iloeje's approach offers also the possibility to calculate the minimum in the heat flux - wall temperature characteristic ("the minimum film boiling temperature"). The original aspect of the model is the attempt to calculate the heat transferred directly to the droplets by analyzing the probability that a droplet reaches the heated surface, using the forces that tend to repell it from the wall. Many hypotheses were formulated, however, in order to calculate the heat transfer to an individual droplet, and three correlation coefficients were left free in order to account for the lack of a solid theoretical or experimental basis for calculating contact area and the total volume of liquid that evaporates during the contact.

For heat transfer to drops that have too little kinetic energy to penetrate the boundary layer, the average minimum thickness of the vapor layer separating the drop from the wall is calculated. Heat is transferred by conduction through this layer.

Heat transfer to the vapor is calculated using a standard correlation for single-phase forced convection. In spite of its merits, Iloeje's method has the shortcoming that too many arbitrary assumptions were made: form of the kinetic energy probability distribution, mechanism of heat transfer from wall to droplets in intimate contact with the surface, and form of the droplet momentum equation in the radial direction where all the forces but the pressure thrust force are neglected.

Contact with the wall is postulated when the distance from the wall becomes equal to the surface roughness. This definition of contact is not consistent with the postulated mechanism of interaction between a truncated sphere and a flat surface. Moreover, the differential evaporation on the two sides of the sphere, that is the reason of the truncated sphere, does not take into account any trend towards symmetry due to the rotation of the

droplet around an axis parallel to the wall.

Kendall (1978) criticized two assumptions of the model by Iloeje. The hypothesis that the drop is rejected from the wall after a brief nucleation period is in opposition to the experimental evidence of complete evaporation and pronounced nucleation of droplets that wet the surface. Moreover, the model of Iloeje does not distinguish between wetting and non-wetting wall temperatures; the minimum film boiling temperature is naturally calculated by the model. Kendall developed a model that replaces the two previous assumptions by a differentiated analysis of the wetting and non wetting regime. For the latter, a dynamic model of the deforming droplet is proposed. The model yields a complicated expression for heat transfer effectiveness in the non-wetting regime in terms of non-dimensional parameters. As Liu and Yao (1982) recognized, the expression does not fit the data in a wide range. From a fundamental point of view, the analysis of Kendall may hold only when a droplet impacts the wall almost perpendicularly, that is when the axial component of the velocity is negligible with respect to the radial one: usually, in dispersed flow film boiling the situation is the reverse.

Ganic and Rohsenow (1977, 1979) modified the theory of Iloeje by considering lift, drag, gravity, buoyancy and reaction forces in the droplet momentum equation to calculate the mimimum drop size that allows the droplet to reach the wall. They assumed that the same initial transversal velocity is given to all droplets.

A droplet size distribution P(d) was assumed and the liquid mass flux to the wall was calculated in terms of a cumulative deposition factor:

$$f = \frac{\int_{d_c}^{d_m} d^3 P(d)\,dd}{\int_{0}^{d_m} d^3 P(d)\,dd}$$

where d_m is the diameter of the largest droplet and d_c is the diameter of the smallest droplet that can deposit onto the wall, respectively.

In their analysis of the drop trajectories, the existence of "dry" contacts is not taken into consideration, and the droplets can either reach the wall or bounce over the thermal boundary layer.

The heat that is transferred to the droplets that strike the wall is calculated on the basis of the effectiveness (ε) of the contact, defined as the fraction of the total amount of vaporization heat that could be absorbed in the case of a perfect contact (total vaporisation) between wall and droplet.

The experiments of Pederson (1970) showed that the contact effectiveness is exponentially decreasing for increasing wall temperature. Use of such a heuristic parameter represents a more realistic approach, since the details of the contact wall-droplet are far from being known.

The model proposed by Ganic and Rohsenow, with some corrections for the form of the thrust force and the boundary conditions at the edge of the thermal boundary layer (see Lee and Almenas, 1982), is an interesting alternative to the often used empirical formula of Forslund and Rohsenow (1963) for estimating direct contact heat transfer.

4.1.2 Models including radiation

Sun, Gonzales and Tien (1975) focused on the calculation of combined radiation and convection heat transfer from the wall to a mixture flowing in laminar flow. The interesting aspect of their work is that for the first time the role played by the so-called "distributed heat sink" was recognized: the heat transfer rate from the wall to the vapor depends on the vapor temperature profile, which in turn depends on the radial distribution of the evaporation rate. It follows that the use of a standard heat transfer coefficient for single-phase flow is not valid when this distributed heat sink modifies the thermal boundary layer. The heat transfer rate has to be calculated by integrating the energy equation in the radial direction.

Radiation heat transfer is fully accounted for: wall, vapor and droplets are the three nodes of a network (electrical analogy) where the radiative currents are calculated. Further developments of this approach will be given in the following section.

The convective part of the model of Sun et al. must be complemented by additional equations, since it was assumed that the vapor bulk temperature has reached the asymptotic value ($\frac{\partial T_g}{\partial z} = 0$). It can be considered the first rational local approach for the calculation of heat transfer to a dispersed flow.

4.1.3 Models considering thermal non-equilibrium

Most of the remaining models that calculate the wall heat flux by using only the local flow variables, are based on the estimation of the thermal non-equilibrium or the mass evaporation rate Γ_g: once the vapor temperature is known, the wall temperature is calculated by a standard single phase HTC correlation. The possibility of calculating Γ_g without the need to integrate the field equations step-by-step downstream from the dryout or the quench front position is very attractive for possible application to computer codes, since the calculation of the interfacial area concentration, the most difficult parameter to predict, is no longer necessary.

It is easy to show (Jones and Zuber, 1977) that thermal non-equilibrium (expressed by the difference or the ratio between the equilibrium quality x_e and the actual quality x) and the actual vapor generation rate Γ_g are related by:

$$\frac{dx}{dx_e} = \frac{\Gamma_g}{\Gamma_e}$$

where $\Gamma_e = 4q''_w/Dh_{fg}$ is the equilibrium evaporation rate. Many researchers have proposed correlations for Γ_g or functional relations for dx/dx_e, mainly derived from simplified one-dimensional models.

Plummer (1976) proposed the relation dx/dx_e = constant, depending on the mass flux and dryout quality.

Jones and Zuber (1977) criticized this relation, since it implies that the rate of return to equilibrium is constant, at variance with the true situation. Instead they showed that the degree of non-equilibrium ($x_e - x$) behaves like a first-order relaxation process having a forcing function proportional to the local rate of heat input and a local relaxation length

depending from quality, void fraction, mass flux, and droplet average diameter. They proposed:

$$\frac{dx}{dx_e} = N_{sr} (x - x_e)$$

where N_{sr}, the non-equilibrium number, represents the ratio of actual heat transfer per unit volume between the superheated vapor and the liquid to that which would be required to evaporate all the liquid in the length between the onset of non-equilibrium and the location where $x_e = 1$. N_{sr} is determined from a correlation of experimental data and depends from the droplet diameter and the velocity ratio at the dryout point, so that the flow history is taken into account to a certain extent.

A recent evolution of the same basic idea is the model by Yoder and Rohsenow (1983), that uses only dryout conditions and local equilibrium fluid conditions to predict the surface temperature under constant heat flux. Wall-to-drop heat transfer, and radiation are considered to play a minor role in most situations and are thus neglected.

Major assumptions are that the drop size distribution is characterized by one average drop size and liquid and vapor velocities are uniform across the tube. The calculation method for initial droplet size takes into account two different situations: annular type dryout ($x > 0.1$) and inverted annular type dryout ($x < 0.1$). It is recognized that, when dispersed flow forms from annular flow all droplets are formed before the dryout point.

Using the conservation equations, ignoring the variation of fluid properties and using the observation that the product of the slip ratio and the void fraction is roughly equal to one, a first order differential equation is obtained:

$$K \frac{dx}{dx_e} = f(x, x_e) (x_e - x)$$

where K, the non-equilibrium constant, contains the group of fluid parameters that dictate departure from equilibrium.

If no break-up occurs, K is a constant that may be calculated using the condition at the dryout point only. This is a peculiar aspect of this calculation method. It is not clear whether the model could be applicable to the case of a non-uniform heat flux.

All the models described above have been assessed against experimental data obtained in stationary dryout joint experiments. The possibility of developing a similar conceptual approach for the calculation of thermal disequilibrium just above a moving quench front has not been explored. Indeed the peculiarity of the region above the QF lies in the strong vapor generation rate at the wall, which yields very high accelerations and velocity differences between the two phases, as well as violent break-up of liquid globules and filaments.

4.2 Integral Models

These models employ the same calculation method that is implemented in the codes: at any elevation z the heat transfer processes are calculated by taking into account the local values of the flow variables, as calculated by a stepwise integration of the conservation equations downstream starting from the point of onset of mist flow. A significant difference between

these models is the way they treat the heat transfer between the wall and the vapor: a first class of models uses standard heat transfer correlations, while in a second group the multidimensional vapor energy equation is integrated. Some examples of both classes of models are presented below and their distinguishing features are highlighted. All models use similar methods for calculating heat transfer from the vapor to the droplets; thus this item is not discussed in this Section.

4.2.1 Models employing heat transfer coefficient correlations

Many models can be included in this class, since practically all experimental investigations were analyzed by a model employing the method "of the four gradients," that was simultaneously developed at MIT (e.g. Forslund and Rohsenow, 1968) and at AERE (UK) (e.g. Bennet et al., 1967).

The basic equations that are solved are the vapor mass and energy, and the liquid momentum conservation equations, as well as the liquid mass continuity (written as a droplet average diameter variation). The liquid energy equation is not necessary since the droplets are assumed to be saturated.

Since the pressure gradient in mist flow is not relevant (as shown by Groeneveld (1972), the pressure losses play a role only in the case of freons), the vapor momentum equation is not usually included in these models. The models take into account the history of the average droplet starting at its generation point, so that the average droplet diameter is related to the mass loss during the flight, and not to the local relative velocity as is done in several large codes: this should be an important improvement for the computer codes as discussed in Section 2.3.3.

Wall-to-vapor heat transfer

Most of the proposed models assume that the heat transfer from the wall to the vapor can be described by the same relations used to describe heat transfer to single-phase gas flow. As highlighted before, this assumption is not justified: flow models tried to take into account the enhancement due to the presence of the dispersed phase.

An empirical approach has been proposed by Varone and Rohsenow (1984). These authors observed that using a HTC correlation valid for pure vapor, even the shape of the wall temperature profiles obtained by Era (1966) could not be reproduced. From an extensive sensitivity analysis aimed at the parameters that could produce such a big discrepancy, they realized that only a modification of the wall-to-vapor heat transfer coefficient could bring the calculated wall temperatures close to the experimental values. Modification in a wide range of any other parameter used in the closure laws could only modify the wall temperature level, but not the shape of the axial temperature profile.

The correction factor γ that Varone and Rohsenow applied to the Hadaller correlation to fit the data passes through a minimum as the quality increases. The same behaviour was experimentally found for gas-particle mixtures. Varone and Rohsenow correlated γ only with quality and the ratio between the vapor viscosities at wall and bulk temperature. The last dependence may account for the wall temperature effect observed in gas-particle dynamic flows. The fact that the correction factor does not depend from the droplet diameter and the Reynolds number (two quantities that have been shown to influence the heat transfer rate to solid dispersions) could be due to the fact that it includes not only the effect of the droplets on the vapor velocity profile but also the "distributed heat sink effect" which is absent in the case of gas-particle mixtures. It

could also include any error in the prediction of the vapor temperature. It is clear then that the γ factor, since not based on a phenomenological basis, cannot be used out of the range of the experimental data that were used for its correlation.

Spencer and Young (1980), starting from the analysis of the FLECHT experiments that showed a significantly higher heat flux to the vapor than could be accounted for by standard correlations, concluded that the physical mechanism responsible for the excess heat transfer was the increased level of vapor turbulence due to the droplets. Extending the analogy between heat and momentum transport, they assumed that any mechanism that creates shearing stresses in the continous phase enhances the turbulent diffusion of heat. One such mechanism could be the drag imposed by the droplets on the vapor, since the local pressure gradient produced by droplet drag is of the same order of magnitude as the wall shear stress induced pressure gradient. The effect of these additional shear forces can be accounted by modifying the local friction factor; from a semianalytical model that predicts the local heat transfer coefficient on the basis of the friction factor (Kays, 1966), the modified heat transfer coefficient can then be calculated.

Styrikovich (1982) observed that the heat transfer coefficient was lower than that expected from a forced convection correlation for single phase vapor, and attributed this anomaly to the change of the velocity profile, that was presumably "elongated" because of the presence of the droplets.

It has been shown in Chapter 3 that both an enhancement or a decrease of the heat transfer coefficient are possible, depending on the quality (or the solid loading ratio for gas-particle flows). Styrikovich derived the correction factor using the analogy between heat and momentum transfer and by modifying the shape of the velocity profile.

Clare and Fairbairn (1984) correlated the enhancement of the skin friction coefficient with turbulence intensity, an effect observed experimentally by Theofanous and Sullivan (1982), by replacing the wall shear in the expression of the friction velocity by the sum of wall shear and interfacial shear on the continous phase.

Three-Step Models: Wall-to-Droplet Heat Transfer

The most complete models consider DFFB as a three-path heat exchange process (three-step model), the third path being direct heat transfer from the wall to the droplets. Above a certain rewetting temperature it is commonly assumed that a droplet cannot touch the wall. However, the enhanced evaporation of droplets that penetrate the thermal boundary layer adds a non-negligible contribution to the total heat transfer rate.

Chen (1984) on the basis of the evident existence of a region of enhanced vapor generation close to the CHF point, concluded that the direct wall-to-liquid heat transfer is negligible only at downstream regions reasonably far from the inception point of film boiling.

Forslund and Rohsenow (1968) used a heat transfer coefficient to the droplets, based on the experiments of Baumeister with nitrogen sessile drops on a horizontal heated plate: the effects of droplet velocity and concentration were correlated.

Plummer (1976) employed the technique proposed by Groeneveld (1972) and calculated the wall-to-droplet heat transfer by a simple heat conduction term assuming a linear temperature profile between the wall and the drop across the vapor film that separates the droplet from the wall. Even though, as shown by Iloeje (1975), the thickness of this film is strongly dependent on the wall temperature, good results may be obtained if an

optimal value is chosen.

The most recent models account for the wall-droplet interaction by expressing the wall-to-droplet heat flux as:

$$q''_d = \varepsilon \dot{m}_d h_{fg}$$

where ε is the heat transfer effectiveness (see 4.1) and $\dot{m}_d$ is the droplet mass deposition rate (droplet mass transfer rate per unit area).

In the model of Koizumi (1979) ε is a constant and $\dot{m}_d$ is given by the product of an empirical constant, derived by Ueda (1978), and the mean concentration of droplets:

$$\overline{C} = \rho_g \frac{\dot{M}_d}{\dot{M}_g} \frac{U_g}{U_d}$$

where $\dot{M}_d$, $\dot{M}_g$ are the phase mass flows and U_g, U_d the average axial velocities.

Mastanajah and Ganic (1981) considered the dependency of ε from the wall temperature and used the experimental results of Styrikovich (1978) for defining an exponential relationship. The number of droplets per unit time and unit surface area that impinge on the wall is expressed as:

$$\dot{N}_d = K_d f \overline{C}$$

where $\overline{C}$ has the same meaning as before, K_d is the deposition velocity obtained by Mastanajah (1980) for a wide range of Reynolds numbers ($6 \cdot 10^3$ to $2 \cdot 10^5$) and f is the penetration factor defined by Ganic and Rohsenow (1979) (see 4.1).

Moose and Ganic (1982) assumed an exponential decay of the heat transferred to an impinging droplet with increasing wall temperature, as considered by Ganic and Rohsenow (1977), but calculated the maximum effectiveness by the correlation of Holmann et al. (1972) who investigated the maximum heat flux to a droplet impinging on a hot surface using several fluids.

The droplet deposition velocity was calculated by the relation $K_d = 0.17 U^*$, where U^* is the friction velocity: this proportionality was found by Ganic and Mastanajah (1981) in their studies on turbulent deposition of droplets from an air-droplet mist flow onto the wall of an adiabatic tube.

Varone and Rohsenow (1984) included in their model a wall-to-droplet heat transfer path, where the deposition velocity is calculated according to Liu and Ilori (1974), and ε is estimated by the model of Kendall (1978): wet and dry contacts are distinguished, and in the second case it is considered that the space between the wall and the drop is occupied by a flowing gas.

Multi-field approach

Two recent models reject the usual assumption that the droplet hydrodynamics can be described in terms of the center-of-mass motion, and propose a multi-field approach for the calculation of the axial concentration of the liquid.

Williams (1983) considers the effects of a distribution of droplet sizes: the small droplets carry most of the interfacial area and travel faster

than the larger ones that carry most of the mass. The global effect on the axial distribution of the interfacial area concentration and on the evaporation rate is claimed to be different than that of a monodisperse spray. The droplets are therefore divided in two groups and an interfacial area transport equation* is written for each group. There is no interchange of number of particles between the small and large droplet groups. Models for the droplet volume mean diameter, upper-limited-log-normal size distribution function, and entrainment rates developed by Ishii and Mishima (1981) are used.

The model is insensitive to the choice of the arbitary boundary between the two groups and was assessed against experimental reflooding data with good results. The fact that no interchanges are allowed between the two droplet groups causes a droplet to stay in the larger diameter group even after significant vaporization, so that the spectrum shift cannot be taken into account.

Kawaji (1984) used in his model eleven groups of droplets, and assumed that the droplet size distribution remains unaltered. The maximum droplet diameter is determined from a local critical Weber number.

Kawaji included in his program the possibility of using only one group of droplets, so that the evaluation of the benefits that are possible by a multifield approach can be appreciated. The numerical results were compared with the experimental data obtained in five selected University of California-Berkeley (UC-B) inconel tube reflooding experiments (Seban et al., 1978). Since both the single-drop formulation and the multi-field approach resulted in fairly good agreement, it is not possible from the limited assessment work presented by Kawaji to draw any conclusion about the need to complicate to such extent the model for mist flow.

Lee, Reyes and Almenas (1984), postulated the existance of two distinct droplet generation mechanisms leading to a bi-spectral droplet population, to explain the exponential decay of heat transfer just above the quench front. A population of small droplets (10-50 μm) is produced by the shattering of the bubbles that grow within the thin liquid layer below the quench front. The size of these droplets depends from the thickness of the liquid film covering the bubbles. A population of much larger droplets (400-1200 μm) is formed from the water between the bubbles. The calculations show that the small droplets evaporate entirely within a distance of 10-50 cm above the quench front, while the large droplets change relatively little. By taking an appropriate weighted average, a characteristic droplet diameter is defined for the two distinct evaporation regimes. The characterictic diameter of the small droplets is determined according to the experimental data of Newitt (1954), and for the large droplets the measurements in the FLECHT experiments were used. From geometrical considerations, the ratio between the number of droplets belonging to the two groups is evaluated.

Cigarini (1987) implemented this droplet population model in the German code FLUT, and obtained a very satisfactory prediction of the cladding temperatures for two FLECHT tests; these could not be correctly calculated using a standard mono-spectral droplet size distribution.

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* Incidentally, use of an area transport equation could be an improvement with respect to numerical stability, since it imposes a gradual reduction of the interfacial area by evaporation.

4.2.2 Models that integrate a multidimensional vapor energy conservation equation

All the models presented in this section are characterized by the assumption that the vapor velocity profile is not affected by the presence of the droplets, so that the enhancement of the heat transfer rate to the vapor is only due to the change of the vapor temperature profile (distributed heat sink effect).

Yao and Rane (1979-1981) presented the extension of the model of Sun et al (1975) to a developing two-phase flow. Both laminar and turbulent flow have been taken into consideration. The basic difference between the two analyses is in the vapor velocity profile: parabolic for laminar flow, two-zone turbulent for turbulent flow.

In the most recent formulation, Rane and Yao (1981) make the following assumptions:

1) Radiative heat transfer and direct-wall contacts are negligible
2) The dispersed droplets have uniform size and their concentration is not function of the radial coordinate
3) The quality is sufficiently high ($x > 0.5$) to neglect interactions between droplets.
4) Constant fluid properties
5) Radial variations of the turbulent-velocity and diffusivity are not affected by the presence of droplets
6) The droplets have the same velocity as the vapor

The effect of droplets is equivalent to that of a heat sink (HS) distributed in the vapor. Referred to a unit volume:

$$HS = n\pi d^2 h_d (T_g - T_{sat})$$

where h_d is the vapor-droplet heat transfer coefficient for an evaporating droplet and n is the droplet number concentration. The steady-state energy equation for the vapor phase can then be written in the form:

$$\rho_g c_p U_g u(r) \frac{\partial T_g}{\partial z} = \frac{1}{r}\frac{\partial}{\partial r}\left[e(r)\, r \frac{\partial T_g}{\partial r}\right] - HS$$

where u(r) is the non-dimensional velocity profile and e(r) is the thermal diffusivity of turbulent flow (including molecular and eddy transport) according to a turbulence model developed by Deissler (1955) for single-phase flow. The averaged velocity of the undisturbed flow U_g is allowed to increase due to evaporation. The local Nusselt number is found to be affected by three parameters: the heat sink paramater (S), the liquid loading parameter ($A_f = \alpha_f \rho_f / \rho_g$) and the wall superheat parameter ($C_w = c_p q''_w R / k h_{fg}$, where R is the radius of the tube and k is the thermal conductivity of the vapor).

The heat sink parameter,

$$S = n_o \pi d_o R^2 h_p d_o / k$$

(where h_p is the heat transfer coefficient for non-evaporating spheres and n_o and d_o are, respectively, the droplet density number and diameter at the inlet) incorporates the droplet concentration and the heat transfer between

vapor and droplets at the entrance of the heated length.

The analysis for laminar flow (supposed to be valid at low flooding rates) gave correct results, as compared with the experimental data from the UC-B single-tube test section, in the region some distance downstream from the quench front. For the zone near the quench front, an empirical correlation was established (Yao and Sun, 1982) for wall heat transfer augmentation due to droplet-to-wall interaction.

Wong and Hochreiter (1980) developed a three-dimensional laminar vapor temperature field equation using the energy balance for non-equilibrium dispersed droplet flow. The vapor temperature distribution is calculated taking into account the effects of conduction, radiation and droplet evaporation.

The improvement with respect to the models of Yao and Rane (beyond consideration of radiation) is the allowance for some velocity difference between the phases; the droplet velocity is assumed to be equal to the local terminal velocity everywhere along the channel.

On the other hand, turbulent mixing is neglected, and this is justified by the fact that the FLECHT experiments that were analyzed were in the transition region between laminar and turbulent flow (Re = 2000 to 5000). The analysis of the discrepancies between calculated and experimental heat fluxes, the latter being systematically unaffected by the Reynolds number, supported the conclusion that turbulent mixing is not important for very low reflooding conditions (2 to 2.5 cm/s). Throughout the analysis an average droplet diameter corresponding to the experimentally observed value (780 μm) was used.

An analogous model (in two-dimensions), including radiation heat transfer from a hot wall to an optically thick medium (high pressure system or large characteristic length or both) was developed by Chung and Olaffson (1984). Both turbulent conductivity and vapor velocity profiles were taken from the theory of turbulent fully-developed single-phase flow (two-layer turbulent conductivity model by Travis, 1971, and three-layer turbulent velocity distributions by Kays, 1966). A uniform droplet diameter was input, and considered as a fitting parameter.

Substantially different is the model proposed by Webb and Chen (1982), who did not assume a vapor velocity profile, but calculated the radial velocity radial distribution by considering, together with the vapor energy equation, the two-dimensional momentum conservation equation. Three important assumptions are still present: negligible direct wall-to-liquid heat transfer, radially uniform vapor generation rate and negligible influence of the dispersed phase on the thermal diffusivity and viscosity. The peculiar feature of this model is the way the vapor generation source function is considered. Convection (conv) and radiation (rad) components are included, so the total (t) source function is:

$$\Gamma_t = \Gamma_{conv} + \Gamma_{rad}$$

where Γ_{rad}, the radiative portion, is assumed to be constant in the radial direction. The convection source function Γ_{conv}, usually related to the droplet size and a given heat transfer correlation, lumps in this model these parameters into a variable σ_Γ, such that

$$\Gamma_{conv} = \sigma_\Gamma (T_g - T_{sat}) (1 - \alpha)^{2/3}$$

σ_Γ is a function of system and operating conditions and is deduced by a best-fit analysis of the experimental vapor superheat data. The formulation in terms of a lumped vapor generation source function σ_Γ is a pragmatic approach circumventing the need for good estimates of droplet size and vapor-to-droplet heat transfer.

Implicit in the form of Γ_{conv} is the assumption that the number of droplets is constant. The model necessitates, however, the specification of a droplet size, in order to calculate the void fraction. The droplets are assumed to travel at their terminal velocity. Knowing the quality and velocity difference, the void fraction can be determined.

The model has been evaluated against the experimental data of Nijhawan (1980) at low pressure and low-to-moderate CHF qualities. In the comparison, the strength of the vapor generation source function σ_Γ was varied to match the experimental vapor temperature. The resulting wall temperatures compared favorably with the experimental wall temperature profiles of Nijhawan (1980). The form of σ_Γ based on the experimental data of Nijhawan and on the "inferred" vapor superheats in the experiments of Bennet (1967) is given by Webb et al. (1982). An extension of the data base by two series of experiments carried out by Evans et al. (1983) and Gottula et al. (1984) under steady-state or slow-reflooding conditions, allowed a refinement of the correlation. The standard deviation in the prediction of the vapor superheat is, however, still 50% (Webb and Chen, 1984). The revised form of σ_Γ includes the effect of the enhanced vapor generation immediately above the quench front due to liquid-wall contacts (near field vapor generation rate).

The same basic idea of accounting for the distributed heat sink in the calculation of the wall-to-vapor heat transfer lead to the use of a correction coefficient in a previously developed phenomenological correlation (Webb and Chen, 1984). It is worth to remark the similarity of this coefficient with the one used by Varone and Rohsenow (1984) to match the data with their model: in both cases the discrepancy between data and prediction by a correlation valid for single-phase flow is attributed to the presence of the droplets. However, while Webb and Chen consider only the effect of the distributed heat sink (and correlate the correction factor with pressure, quality and Reynolds number), Varone and Rohsenow (1984) correlate the correction factor with the quantities that are supposed to modify the level of turbulence of the mixture (quality and viscosity ratio), as discussed in 4.2.1. Both approaches are thus partial and empirical in nature.

Another interesting result of the analysis of Webb and Chen (1982) is that a fully developed velocity profile may be used, since only slight differences in the calculation were detected between the present model and a previous formulation (Webb and Chen, 1981) that assumed no radial component of the vapor velocity.

The assumption of uniform droplet concentration (uniform heat sink distribution) is removed in the recent model by Kirillov et al. (1987).

Their two-dimensional analytical formulation of annular-dispersed flow involves mass, momentum and energy conservation equations for the vapor and the droplet flows, as well as for the liquid film. The advanced feature of this model is the consideration of the two-dimensional motion of a population of droplets: a mass transfer equation for a flow of droplets is written for all the droplet size groups in which the entire spectrum is divided.

The convective component in the advection-diffusion equation (in terms of

the concentration C_i of the i-th group) is determined by the steady-state equation for a moving spherical liquid droplet in a vapor flow (analysis similar to that of Ganic and Rohsenow, 1977).

The boundary conditions for the mass transfer equations incorporate the radial distribution of the concentration of droplets in the initial cross-section. The value of the most probable droplet diameter in the initial cross-section is calculated by using an empirical correlation. The entrainment rate is calculated according to a model that is referenced in the Russian literature.

The diffusion coefficient for a group of droplets incorporates the effect of the interaction of the droplets with the turbulent eddies and is taken equal to that of particles having no inertia. A correction for their inertia is, however, applied since a substantial effect on the velocity fluctuation amplitudes is observed for large particles. The physical situation that is modelled is that of annular-dispersed flow, followed by dry-out of the liquid film. The vapor axial velocity profile is assumed not to be affected by dryout: it is assumed to adjust instantaneously to the fully developed value, including any liquid evaporation.

An effectiveness ε, similar to the one used by Ganic and Rohsenow (1977), is employed to take into account the fraction of evaporation due to the wall deposition.

The model of Kirillov et al. is the most complete among those proposed up to date, and the only simplifying assumptions are the absence of radiative heat transfer (justified by the fact that the model is developed for the analysis of annular-dispersed flow in steam-generator tubes), an equilibrium condition for the droplet size distribution (no break-up), and the disregard of the influence of the dispersed phase on the vapor velocity field.

Moreover, since the theory was not developed for DFFB under reflooding conditions, the models for entrainment rates and droplet size distribution are not necessarily applicable to such cases. Computational experiments conducted by the authors have revealed some specific features of post burnout heat and mass transfer: the maximum of the droplet concentration curve is located in the central part of the channel, and smoothly decreases to zero at the wall. This result is supported by experimental data in annular-dispersed flow. The calculations have shown that the effect of the most probable droplet size on the results increases with a decrease of the droplet diameter. For highly dispersed systems departures of the most probable droplet size from the real one affect appreciably the calculations.

It has also been shown that the use of the monodisperse approximation alone may lead to errors in the prediction of temperatures and heat fluxes of up to 30%.

5. CONCLUSIONS

The models implemented in the reactor safety codes have in principle the capability to account for all the important wall and interfacial mass, momentum and energy transfer mechanisms that take place in DFFB.

The tight coupling between heat transfer processes and topological characteristics of the flow (distribution of the phases over the cross-section) imposes some caution in defining and using the cross-section-averaged quantities that are required in the one-dimensional formulation of the codes. This averaging problem is common to any two-phase flow pattern: what is special in DF is the fact that a very high temperature gradient exists in the vapor phase, so that the distribution of the evaporating water droplets plays a fundamental role in the heat transfer process.

The full representation of droplet hydrodynamics is far beyond the possibilities of the codes: the introduction of distribution coefficients in the conservation equations and closure laws is a candidate method for improving the physics of one-dimensional models.

The importance of direct wall-to-droplet heat transfer has not yet been fully assessed, also because of the lack of a reliable criterion for the wettability of the wall. In any case, the partition of the contact area between the wall and the two phases on the basis of the average void fraction is definitely wrong and poses a serious limit to the realism of the entire procedure.

Enhanced heat transfer to the liquid present in proximity of (or in contact with) the wall can only be calculated if the concentration of droplets near the wall and their radial velocity are known: within the frame of a one-dimensional model the deposition velocity could in principle be correlated to the average characteristics of the flow and the conditions at the onset of DFFB.

Of special importance during reflooding is the initial radial velocity of the droplets that determines the deposition rate just above the quench front. An empirical parameter such as the effect[illegible]s of the contact, relating the heat flux to the droplet to the dr[illegible] mass flux could be derived from experiments (much experimental work has been already accomplished, and more would be welcome).

A critical parameter in modelling DFFB is the characteristic droplet diameter, since it affects the distribution of the liquid phase both in the axial and radial directions. The experimental evidence shows that within a short distance from the generation point, droplets of very different sizes attain the same axial velocity. This behaviour indicates that the introduction of a multi-field approach cannot be beneficial to the realism of the model.

Nevertheless, the choice of a representative diameter that characterizes the multi-sized population of droplets affects the deposition velocity, the average axial drag, interfacial heat transfer, and radiation heat transfer from the wall. In this respect, the droplet size distribution at the generation point is of crucial importance: in the codes, a fixed distribution is implied, independently from the flow regime from which the droplets originated. The experiments show, on the contrary, the dependency of the size of the droplets from the flow pattern upstream of the DF zone. On the other hand, a general relation between droplet size and initial conditions has not yet been experimentally established. An improvement of the codes in this area is required. A flow pattern dependent droplet size at the inception of DFFB is, however, useful only if the evolution of the characteristic droplet size according to the mechanisms that control it

(break-up, evaporation and coalescence, in that order of importance) is accounted for.

The actual criterion used by the codes to define the maximum droplet diameter, based on the local critical Weber number is completely unphysical. It should be replaced by a transport equation considering the fact that break-up does not occur instantaneously: large drops can indeed be carried over in spite of the fact that their existance would not have been allowed by aerodynamic stability criteria.

Interaction between the drops affects interfacial drag and (possibly) heat transfer. The codes overlook this effect, which, in the range of void fractions that is typical of DFFB might be important. More experimental work is needed in this area, since the basic interaction mechanisms are poorly understood.

Droplet-vapor interaction mechanisms that affect the wall-to-vapor heat transfer cannot be taken into account by the one-dimensional models: these are a) the modification of the thermal boundary layer due to the evaporation near the wall, and b) the alteration of the vapor velocity profile by changes in the structure of turbulence due to the droplets. In principle, a correction factor to the single-phase forced convection correlation (Dittus-Boelter or other) could take into account both effects. Unfortunately such a correction factor considering all the parameters that are known to affect the heat transfer coefficient is not yet available, and partial corrections must be discouraged, because of the complex and not fully understood variation of the HTC (strong enhancements and reductions are both experimentally observed).

More attention should be paid to radiation heat transfer, which can give an important contribution to the total heat flux, often neglected by the modeller.

The presence of spacer grids in the fuel rod bundles should be considered in the DFFF model, as it affects locally the wall heat transfer and, what is more important, results in a shift of the droplet size distribution to smaller droplets.

In summary the basic deficiencies of the models are due to a simplified description of the droplet hydrodynamics and the disregard for the wall-to-vapor heat transfer coefficient modification due to the dispersed phase.

The literature survey reveals, however, that most of the one-dimensional models are based on the same assumptions used in the codes, with the only exception of an improved consideration of the droplet size evolution. Interesting improvements to specific sub-models have been proposed, but the importance of these modifications is always difficult to assess because of the many simplifications in other areas (e.g. neglect of wall-to-drop heat transfer, of radiation heat transfer, predetermined velocity ratio etc., in various combinations). A few two-dimensional models have, however, been proposed: their main limitation is the rough modelling of the droplet hydrodynamics or the structure of the flow.

In conclusion, a model that accounts for all the important phenomenological aspects of DFFB that have been discussed in the present report and are not considered in the codes is not available in the literature.

However, the need is felt for a complete model that could help to point out and understand the importance of the several sub-processes under various conditions: this could be helpful for defining the priorities for new theoretical and experimental work.

The influence of the conditions at the incipience of DFFB (such as droplet population size spectrum and droplet initial transverse momentum) has also

to be studied, as any improvement in the mechanistic description of DFFB could be useless if the results are too sensitive to the initial conditions. At the present time these cannot be easily defined because of the lack of systematic experimental investigations.

NOMENCLATURE

A	flow area
A_d	projected area of the droplet
A_f	liquid loading parameter
B	Spalding number
B_d	volume of the droplet
B_M	mass transfer number
c_p	vapor specific heat capacity
C	proportionality factor
C_e	distribution parameter of energy (or enthalpy) flux
C_o	distribution parameter of volumetric flux
C_V	distribution parameter of momentum flux
C_w	wall superheat parameter
$\bar{C}$	mean concentration of droplets
d	droplet diameter
D	hydraulic diameter
e	total heat diffusivity (molecular+turbulent)
e_K	specific internal energy or enthalpy of phase K
Eo	Eotvos number
f	deposition factor
F	drag force
g	gravity acceleration
G	mass flux
Gr	Grashof number
h	enthalpy
HS	heat sink
HTC	heat transfer coefficient
k	thermal conductivity
K	non-equilibrium constant
K_d	droplet deposition velocity
La	Laplace number
m	loading ratio
$\dot{m}_d$	droplet mass deposition rate
$\dot{M}$	axial mass flow
n, N_d	droplet number concentration
$\dot{N}_d$	droplet flux
N_{sr}	non-equilibrium parameter
N_μ	viscosity number
Nu	Nusselt number
p	pressure
Pr	Prandtl number
q''	heat flux
Q	volumetric interfacial heat transfer rate
r	droplet radius, radial coordinate
R	tube radius
Re	Reynolds number
S	heat sink parameter
Sc	Schmidt number
Sh	Sherwood number
t	time
T	temperature
U	average axial velocity

U^*	friction velocity
V	velocity
x	quality
W	mass flow
We	Weber number
z	axial coordinate

Greek symbols

α	void fraction
Γ	volumetric vapor generation rate
$\Delta\rho$	density difference
ε	wall-to-droplet contact heat transfer effectiveness
μ	viscosity
ρ	density
σ	surface tension
σ_Γ	vapor generation source function
ϕ	radiative heat flux

Subscripts and symbols

c	continous phase
conv	convection
crit	critical
d	discontinous phase, droplet
D	drag
e	equilibrium
f	liquid
g	vapor
i	interfacial
K	K phase
l	liquid
m	maximum
o	inlet or dryout conditions
p	particle
r	relative
rad	radiation
sat	saturation
sm	Sauter mean
v	vapor
w	wall
$\langle\ \rangle$	cross-sectional average
$\langle\langle\ \rangle\rangle$	phase cross-sectional average (over area occupied by phase only)

REFERENCES

Adams, J.E. and Clare, A.J. (1984) "Droplet Break-up and Entrainment at P.W.R. Spacer Grids," The First Int. Workshop on Fundamental Aspects of Post-Dryout Heat Transfer, Salt Lake City, Utah, April 2-4, 1984, NUREG/CP-0060.

Afifi, J. (1985) "Evaluation of Some Large "Best Estimate" Computer Codes for the Prediction of Loss of Coolant Accidents in Light Water Reactors," Ph.D. Thesis, University of London.

Akimoto, H. and Murao, Y. (1987) "Current Experiment Status and Future Plan for Reflood Tests at Japan Atomic Energy Research Institute," paper presented at ICAP Post-CHF Meeting, June 22-25, 1987, Winfrith, England.

Analytis, G., Richner, M. and Aksan, S.N. (1987) "Assessment of Interfacial Shear and Wall Heat Transfer of RELAP5/MOD2/36.02 During Reflooding," paper presented at the Europ. Two-Phase Flow Group Meeting, June 1-4, 1987, Trondheim, Norway.

Ardron, K.A. and Hall, P.C. (1981) "Droplet Hydrodynamics and Heat Transfer in the Dispersed Flow Regime in Bottom Flooding," CEGB, RD/B/5007N81, Berkeley.

Bennett, A.W., Hewitt, G.F., Kearsey, H.A. and Keeys, R.F.K. (1967) "Heat Transfer to Steam-Water Mixtures Flowing in Uniformly Heated Tubes in which the Critical Heat Flux has been Exceeded," AERE-R-5373.

Bellan, J. and Harstad, K. (1987) "Analysis of the Convective Evaporation of Nondilute Clusters of Drops," Int. J. Heat Mass Transfer, 30, 1, 125-136.

Boothroyd, R.G. and Hague, H. (1970) "Fully Developed Heat Transfer to a Gaseous Suspension of Particles Flowing Turbulently in Ducts of Different Size," J. Mechanical Engineering Science, 12, 3, 191-200.

Chen, J. (1987) "TRAC Assessment with Rod Bundle Data," paper presented at ICAP Post-CHF Meeting, June 22-25, 1987, Winfrith, England.

Chuchottaworn, P. and Asano, K. (1985) "Calculation of Drag Coefficients on an Evaporating or Condensing Droplet," J. of Chem. Engineering of Japan, 18, 91-94.

Chung, J.N. and Olafsson, S.I. (1984) "Two-Phase Droplet Flow Convective and Radiative Heat Transfer," Int. J. Heat Mass Transfer, 27, 6, 901-910.

Cigarini, M. (1987) "Thermohydraulische Untersuchungen zu den Vorgaengen waehrend der Flutphase nach einem Kuehlmittelverlust bei einem forgeschrittenen Druckwasserreaktor," KFK 4302, EUR 10538d.

Clare, A.J. and Fairbairn, A. (1984) "Droplet Dynamics and Heat Transfer in Dispersed Two-Phase Flow," The First Int. Workshop on Fundamental Aspects of Post-Dryout Heat Transfer, Salt Lake City, Utah, April 2-4, 1984, NUREG/CP-0060.

Clift R., Grace, J.R., and Weber, M.E. (1978) Bubbles, Drops and Particles, Academic Press, New York.

Collier, J.G. (1981) Convective Boiling and Condensation, 2nd Ed., McGraw-Hill, New York.

Cumo, M., Farello, G.E., Ferrari, G. and Palazzi, G. (1973) "On Two-Phase Highly Dispersed Flows," ASME Paper 73-HT-18.

De Jarlais G. (1983) "An Experimental Study on Inverted Annular Flow Hydrodynamics Utilizing on Adiabatic Simulation," NUREG/CR-3339.

Depew, C.A. (1979) "Heat Transfer to Flowing Gas-Solid Mixtures," Advances in Heat Transfer, 9, 113-180.

Deruaz, R. and Petitpain, B. (1976) "Modelling of Heat Transfer by Radiation During the Reflooding Phase of LWR," Proc. of the Specialist Meet. on the behaviour of the Fuel Elements under Accident conditions, Spatind, Norway, September 13-16, 1976.

Dougall, R.S. and Rohsenow, W.M. (1963) "Film Boiling on the Inside of Vertical Tubes with Upward Flow of the Fluid at Low Qualities," MIT report 9079-26.

Drucker, M.I., Dhir, V.K., and Duffey, R.B. (1984) "Two-Phase Heat Transfer For Flow in Tubes and Over Rod Bundles with Blockages," J. Heat Transfer, 106, 856-864.

El Kassaby, M.M. and Ganic, E.N. (1984) "A Further Study in Drop Deposition," Multiphase Flow and Heat Transfer III, Part A, ed. Veziroglu and Bergles, Elsevier Science Publishers B. V., Amsterdam.

Era, A., Gaspari, G.P., Hassio A., Milani A. and Zavattarelli, R. (1966) "Heat Transfer Data in the Liquid Deficient Region for Steam-Water Mixtures at 70 kg/cm^2 Flowing in Tubular and Annular Conduits," CISE-R-184, June 1966.

Evans, D.G., Webb, S.W., and Chen, J.C. (1983) "Measurement of Axially Varying Non-Equilibrium in Post-Critical-Heat-Flux Boiling in a Vertical Tube," NUREG/CR-3363 June 1983.

Faeth, G.M. (1983) "Evaporation and Combustion of Sprays," Prog. Energy Comb. Sci., 9, 1-76

Forslund, R.P. and Rohsenow, W.M. (1968) "Dispersed Flow Film Boiling," J. Heat Transfer, 90, 399-407.

Ganic, E.N. and Rohsenow, W.M. (1977) "Dispersed Flow Heat Transfer," Int. J. Heat Mass Transfer, 20, 855-866.

Ganic, E.N. and Rohsenow, W.M. (1979) "On the Mechanism of Liquid Drop Deposition in Two-Phase Dispersed Flow," J. Heat Transfer, 101, 288-294.

Ganic, E.N. and Mastanajah, K. (1981) "Investigation of Droplet Deposition from a Turbulent Gas Stream," Int. J. multiphase Flow, 7, 401-422.

Ghazanfari, F., Hicken, E.F., and Ziegler, A. (1980) "Unsteady Dispersed Flow Heat Transfer Under Loss of Coolant Accident Related Conditions," Nucl. Technology, 51, 21-26.

Gnielinski, V. (1976) "New Equations for Heat and Mass Transfer in Turbulent Pipe and Channel Flow," Int. Chem. Eng., 16, 2, 359-367.

Groeneveld, D.C. (1972) "The Thermal Behaviour of a Heated Surface at and Beyond Dryout," AECL-4308, Nov. 1972

Groeneveld, D.C. (1975) "Post-Dryout Heat Transfer: Physical Mechanisms and a Survey of Prediction Methods," Nucl. Eng. Des., 32, 283-294.

Hagiwara, Y., Suzuki, Y., and Kato, T. (1980) "An Experimental Investigation on Liquid Droplets Diffusion in Annular-Mist Flow," Multiphase Transport, Ed. Veziroglu, Hemisphere Publishing Corp., Washington.

Harmathy, T.Z. (1960) "Velocity of Large Drops and Bubbles in Media of Infinite and Restricted Extent," AIChE Journal, 6, 281.

Harpole, G. (1979) "Water Droplet Evaporation in High Temperature Surrounds," ASME Paper 79-WA/HT-6.

Hasegawa, S., Echigo, R., Kanemaru, K., Ichimiya, K. and Sanui, M. (1983) "Experimental Study of Forced Convective Heat Transfer of Flowing Gaseous Solid Suspension at High Temperature," Int. J. Multiphase Flow, 9, 131-145.

Hassan, Y.A. (1986) "Dispersed-Flow Heat Transfer During Reflood in a Pressurized Water Reactor After a Large Break Loss-of-Coolant Accident," Trans. ANS, 53, 326.

Hassan, Y.A. (1987) "Assessment of a Modified Interfacial Drag Correlation in Two-Fluid Model Codes," Trans. ANS, 54, 211.

Hoffmann, T.W. and Ross., L.L. (1972) "A Theoretical Investigation of the Effect of Mass Transfer on Heat Transfer to an Evaporating Droplet," Int. J. Heat Mass Transfer, 15, 599-617.

Holmann, J.P., Jenkins, P.E. and Sullivan, F.G. (1972) "Experiments on Individual Droplet Heat Transfer Rates," Int. J. Heat Mass Transfer, 15, 1489-1495.

Iloeje, O.C., Rohsenow, W.M., and Griffith, P. (1975) "Three-Step Model of Dispersed Flow Heat Transfer (Post CHF Vertical Flow)," ASME Paper 75-WA/HT-1.

Ishii, M. (1975), Thermo-Fluid Dynamic Theory of Two-Phase Flow, Publisher Eyrolles, Paris.

Ishii, M. (1977) "One-Dimensional Drift-Flux Model and Constitutive Equations for Relative Motion Between Phases in Various Two-Phase Flow Regimes," ANL-77-47.

Ishii, M. and Zuber, N. (1979) "Drag Coefficients and Relative Velocity in Bubbly, Droplet or Particulate Flows," AIChE Journal, 25, 843-855.

Ishii, M. and Mishima, K. (1981) "Correlation for Liquid Entrainment in Annular Two-Phase Flow of Low Viscous Fluid," NUREG/CR-2885.

Ishii, M. and Mishima, K. (1984) "Two-Fluid Models and Hydrodynamic Constitutive Relations," Nucl. Eng. and Design, 82, 107-126.

Jepson, G., Poll, A. and Smith, W. (1963) "Heat Transfer from Gas to Wall in a Gas/Solids Transport Line," Trans. Instn. Chem. Engrs., 41, 207-211.

Jones, O.C. and Zuber, N. (1977) "Post-CHF Heat Transfer: A Non-Equilibrium, Relaxation Model," ASME Paper 77-HT-75.

Juhel, D. (1984) "A Study on Interfacial and Wall Heat Transfer Downstream a Quench Front," The First Int. Workshop on Fundamental Aspects of Post-Dryout Heat Transfer, Salt Lake City, Utah, April 2-4, 1984, NUREG/CP-0060.

Kataoka, I., Ishii, M., and Mishima, K. (1983) "Generation and Size Distribution of Droplet in Annular Two-Phase Flow," J. of Fluids Engineering, 105, 230-238.

Kawaji, M., Ng, Y.S., and Banerjee, S. (1983) "Flow Regime and Heat Transfer Measurements During Reflooding with Steady and Oscillatory Coolant Injection," Thermal Hydraulics of Nuclear Reactors, M.Merilo ed., ANS, Vol. 1, pp. 262-270 (2nd Int. Topical Meeting on Nuclear Reactor Thermal-Hydraulics, Santa Barbara, CA, Jan 11-14).

Kawaji, M. (1984) "Transient Non-Equilibrium Two-Phase Flow: Reflooding of a Vertical Flow Channel," Ph.D. Thesis, University of California, Berkeley.

Kawaji, M. and Banerjee, S. (1987) "Application of a Multifield Model to Reflooding of a Hot Vertical Tube: Part 1 - Model Structure and Interfacial Phenomena," J. Heat Transfer, 109, 204-211.

Kendall, G.E. (1978) "Heat Transfer to Impacting Drops and Post-Critical Heat Flux Dispersed Flow," Ph.D. Thesis, M.I.T., Cambridge.

Kianjah, H., Dhir, V.J., and Singh, A. (1984) "An Experimental Study of Heat Transfer Enhancement in Disperse Flow in Rod Bundles," The First Int. Workshop on Fundamental Aspects of Post-Dryout Heat Transfer, Salt Lake City, Utah. April 2-4, 1984, NUREG/CP-0060.

Kirillov, P.L., Kashcheyev, V.M., Muranov, Yu.V., and Yuriev., Yu.S. (1987) "A Two-Dimensional Mathematical Model of Annular Dispersed and Dispersed Flows," Int. J. Heat Mass Transfer, 30, 791-806.

Koizumi, Y., Ueda, T., and Tanaka, H. (1979) "Post-Dryout Heat Transfer to R-113, Upward Flow in a Vertical Tube," Int. J. Heat Mass Transfer, 22, 669-678.

Kocamustafaogullari, E., De Jarlais, G., and Ishi, M. (1983) "Droplet Generation During Reflood," Trans. ANS, 45, 804-805.

Krzeczkowski, S.A. (1980) "Measurement of Liquid Droplet Disintegration Mechanisms," Int. J. Multiphase Flow, 11, 227-239.

Labowsky, M. (1978) "A Formalism for Calculating the Evaporation Rates of Rapidly Evaporating Interacting Particles," Comb. Science and Techn., 18, 145-151.

Lawn,C.J. (1969) "Turbulent Heat Transfer at Low Reynolds Number," J. Heat Transfer, 91, 532-536.

Lee, K. and Ryley, D.J. (1968) "The Evaporation of Water Droplets in Superheated Steam," ASME Paper 68-HT-11.

Lee, N., Wong, S., Yeh, H.C., and Hochreiter, L.E. (1982) "PWR FLECHT SEASET Unblocked Bundle, Forced and Gravity Reflood Task, Data Evaluation and Analysis Report," EPRI NP-2013, NUREG/CR-2256, WCAP-9891.

Lee, R. and Almenas, K. (1982) "Droplet Deposition Above a Quench Front During Reflooding," Trans. ANS, 39, 787-788.

Lee R., Reyes J.N. and Almenas K. (1984) "Size and Number Density Change of Droplet Population Above a Quench Front During Reflooding," Int. J. Heat Mass Transfer, 27, 573-585.

Lee, S.L. and Durst, F. (1982) "On the Motion of Particles in Turbulent Duct Flows," Int. J. Multiphase Flow, 8, 125-146.

Lee, S.L. (1987) "Particle Drag in a Dilute Turbulent Two-Phase Suspension Flow," Int. J. Multiphase Flow, 13, 247-256.

Lee, S.L., Sheen, H.J., Cho, S.K. and Issapour, I. (1984) "Measurement of Droplet Dynamics Across Grid Spacer in Mist Cooling of Subchannel of PWR," The First Int. Workshop on Fundamental Aspects of Post-Dryout Heat Transfer, Salt Lake City, Utah, April 2-4, 1984, NUREG/CP-0060.

Lilles, D.N. and Mahaffy, J.H. (1986) "TRAC-PF1/MOD1: An Advanced Best-Estimate Computer Program for PWR Thermal-Hydraulic Analysis," NUREG/CR-3858.

Liu L. and Yao, S.C. (1982) "Heat Transfer Analysis of Droplet Flow Impinging on a Hot Surface," 7th Int. Heat Transfer Conf., Munich, BRD.

Mastanajah, K. (1980) "Experimental and Theoretical Investigation of Droplet Deposition and Heat Transfer in Air-Water Dispersed Flow," Ph.D. Thesis, Univ. of Illinois at Chicago.

Mastanajah, K. and Ganic, E.N. (1981) "Heat Transfer in Two-Component Dispersed Flow," J. Heat Transfer, 103, 300-306.

Mayinger, F. and Langner, H. (1978) "Post-Dryout Heat Transfer," 6th Int. Heat Transfer Conf., Toronto.

Moose, R.A. and Ganic, E.N. (1982) "On the Calculation of Wall Temperatures in the Post-Dryout Heat Transfer Region," Int. J. Multiphase Flow, 8, 525-542.

Nijhawan, S., Chen, J.C., Sundaram, R.K., and London, E.J. (1980) "Measurement of Vapor Superheat in Post-Critical-Heat-Flux Boiling," J. Heat Transfer, 102, 465-470.

Nishio, S. and Hirata, M. (1978) "Direct Contact Phenomenon Between a Liquid Droplet and High Temperature Solid Surfaces," 6th Int. Heat Transfer Conf., Toronto, Canada.

Nukiyama, S. and Tanazawa, Y. (1938) "An Experiment on the Atomization of Liquid by Means of an Air Stream," Trans JSME, 4, 86.

O' Rourke, P.J. (1981) "Collective Drop Effects on Vaporizing Liquid Sprays," Ph.D. Thesis, Univ. of Princeton.

Pedersen, C.O. (1970) "An Experimental Study of the Dynamic Behavior and Heat Transfer Characteristics of Water Droplets Impinging Upon a Heated Surface," Int. J. Heat Mass Transfer, 13, 369-381.

Plummer, D.N., Griffith, P. and Rohsenow, W.M. (1976-77) "Post-Critical Heat Transfer to Flowing Liquid in a Vertical Tube," Trans. CSME, 3, 151-158.

Podvysotsky, A.M. and Shrayber, A.A. (1984) "Coalescence and Break-Up of Drops in Two-Phase Flows," Int. J. Multiphase Flow, 10, 195-209.

Rane, A.G. and Yao, S. (1980) "Heat Transfer of Evaporating Droplet Flow in Low Pressure Systems," ASME Paper 79-WA/HT-10.

Rane, A.G. and Yao, S. (1981) "Convective Heat Transfer to Turbulent Droplet Flow in Circular Tubes," J. Heat Transfer, 103, 679-684.

Ransom, V.H., Wagner, R.J., Trapp, J.A., Feinauer, L.R., Johnsen, G.W., Kiser, D.M. and Riemke, R.A. (1985) "RELAP5/MOD2 Code Manual," NUREG/CR-4312.

Sarjeant, M. (1973) "Drop Break-Up by Gas Strams," CEGB, R/M/N1005.

Seban, R., Greif, R., Yadigaroglu, G., Elias, E., Abdollahian, D., and Peak, W.T. (1978) "UC-B Reflood Program: Experimental Data Report," EPRI Report NP-743.

Shrayber, A.A. (1976) "Turbulent Heat Transfer in Pipe Flows of Gas Conveyed Solids," Heat Transfer - Soviet Research, 8, 3, 60-67.

Soo, S.L. (1967) Fluid Dynamics of Multiphase Systems, Blaisdell Publishing Company, Waltham, Massachussets.

Spencer, A.C. and Young, M.Y. (1980) "A Mechanistic Model for the best Estimate Analysis of Reflood Transients (The BART Code)," 19th National Heat Transfer Conference, Orlando, Florida, ASME publication HTD-7.

Styrikovich, M.A., Baryshev, Yu.V., Tsiklauri, G.V., and Grigorieva, M.E. (1978) "The Mechanism of Heat and Mass Transfer between a Drop and a Heated Surface," 6th Int. Heat Transfer Conf., Toronto.

Styrikovich, M.A., Baryshev, Yu.V., Grigorieva, M.E., and Tsiklauri, G.V. (1982) "Investigation of Heat Transfer Processes during Film Boiling in Steam Generating Channel," 7th Int. Heat Transfer Conf., Munich, 1982.

Sukomel, A.S., Koryukin, S.I., Tsvetkov, F.F., Kerimov, R.V. and Spirina, Z.N. (1979) "Experimental Study of Heat Transfer to Aerosol Flows at High Wall Temperatures," Heat Transfer-Soviet Research, 11, 4, 137-141.

Sun, K.H., Gonzales, J.M. and Tien, C.L. (1975) "Calculations of Combined Radiation and Convection Heat Transfer in Rod Bundles Under Emergency Cooling Conditions," ASME Paper 75-HT-64.

Temkin, S. and Mehta, H.K. (1982) "Droplet Drag in An Accelerating and Decelerating Flow," J. Fluid Mech., 116, 97-313.

Theofanous, T.G. and Sullivan, J. (1982) "Turbulence in Two-Phase Dispersed Flows," J. Fluid Mech., 116, 343-362.

Toknoka, N., Kobayashi, T., and Sato, G.T. (1982) "The Evaporation of a Small Droplet in a Hot Air Stream," 2nd Int. Conf. on Liquid Atomization and Spray Systems, Madison, Wisconsin, U.S.A.

Travis, J.R., Buhr, H.O., and Sesonske, A. (1971) "A model for Velocity and Eddy Diffusivity Distributions in Fully Turbulent Pipe Flows," Can. J. Chem. Eng., 43, 14-18.

Tsuji, Y., Morikawa, Y., and Shiomi, H. (1984) "LDV Measurements of an Air-Solid Two-Phase Flow in a Vertical Pipe," J. Fluid Mech., 139, 417-434.

Ueda, T., Tanaka H. and Koizumi, Y. (1978) "Dryout of Liquid Film in High Quality R-113 Upflow in a Heated Tube," 6th Int. Heat Transfer Conf., Toronto.

Varone, A.F. and Rohsenow, W.M. (1984) "Post Dryout Heat Transfer Prediction," The First Int. Workshop on Fundamental Aspects of Post-Dryout Heat Transfer, Salt Lake City, Utah, April 2-4, 1984, NUREG/CP-0060.

Wallis, G.B. (1969) One-Dimensional Two-Phase Flow, McGraw-Hill, New York.

Webb, S.W. and Chen, J.C. (1981) "A Non-Equilibrium Model for Post-CHF Heat Transfer," 3rd CSNI Specialist Meeting on Transient Two-Phase Flow, Pasadena, CA, March 23-25.

Webb, S.W. and Chen J.C. (1982) "A Numerical Model for Turbulent Non-Equilibrium Dispersed Flow Heat Transfer," Int. J. Heat Mass Transfer, 25, 325-335.

Webb, S.W., Chen, J.C., and Sundaram, R.K. (1982) "Vapor Generation Rate in Non-Equilibrium Convective Film Boiling," 7th. Int. Heat Transfer Conf., Munich.

Webb, S.W. and Chen, J.C. (1984) "A Two-Region Vapor Generation Rate Model for Convective Film Boiling," The First Int. Workshop on Fundamental Aspects of Post-Dryout Heat Transfer, Salt Lake City, Utah, April 2-4, 1984, NUREG/CP-0060.

Wilkes, N.S., Azzopardi, B.J., and Willetts, I. (1983) "Drop Motion and Deposition in Annular Two-Phase Flow," Thermal Hydraulics of Nuclear Reactors, M.Merilo ed., ANS, Vol. 1, pp. 202-209 (2nd Int. Topical Meeting on Nuclear Reactor Thermal-Hydraulics, Santa Barbara, CA, Jan 11-14).

Williams, K.A. (1983) "Numerical Fluid Dynamics of Non-Equilibrium Steam-Water Flows with Droplets," Ph.D. Dissertation, Univ. of New Mexico.

Williams, J.J. and Crane, R.I. (1983) "Particle Collision Rate in Turbulent Flow," Int. J. of Multiphase Flow, 9, 421-435.

Wong, S. and Hochreiter, L.E. (1980) "A Model for Dispersed Flow Heat Transfer during Reflood," Proc. 19th Nat. Heat Transfer Conf., Orlando, Fl., USA.

Yadigaroglu, G. and Lahey, R.T., Jr. (1976) "On the Various Forms of the Conservation Equations in Two-phase Flow," Int. J. Multiphase Flow, 2, 477-494.

Yadigaroglu, G. (1978) "The Reflooding Phase of the LOCA in PWRs. Part I: Core Heat Transfer and Fluid Flow," Nucl. Safety, 19, 20-36.

Yao, S.C. (1979) "Convective Heat Transfer of Laminar Droplet Flow in Thermal Entrance Region of Circular Tubes," J. Heat Transfer, 101, 480-483.

Yao, S.C. and Rane, A. (1980) "Heat Transfer of Laminar Mist Flow in Tubes," J. of Heat Transfer, 102, 678-683.

Yao, S.C. and Rane, A. (1980) "Numerical Study of Turbulent Droplet Flow

Heat Transfer," ASME Paper 80-WA/HT-48.

Yao, S.C. and Sun K.H. (1982) "A Dispersed Flow Heat Transfer Model for Low-Flow Bottom Reflooding Conditions," Heat Transfer in Nuclear reactor Safety, ed. Bankoff and Afgan, Hemisphere Publishing Corp., Washington.

Yao, S.C. and Cai, K.Y. (1985) "The Dynamics and Leidenfrost Temperature of Drops Impacting on a Hot Surface at Small Angles," ASME Paper 87-WA/HT-39.

Yao, S.C., Hochreiter, L.E. and Cai, K.Y. (1988) "Dynamics of Droplets Impacting on Thin Heated Strips," J. Heat Transfer, 110, 214-220.

Yoder, G.L. and Rohsenow, W.M. (1983) "A Solution for Dispersed Flow Heat Transfer Using Equilibrium Fluid Conditions," J. Heat Transfer, 105, 10-17.

Yu, K.P. and Yadigaroglu, G. (1979) "Heat Transfer Immediately Downstream of the Quench Front During Reflooding," ASME paper 79-HT-48.

Yuen, M.C. and Chen, L.W. (1978) "Heat Transfer Measurements of Evaporating Liquid Droplets," Int. J. Heat Mass Transfer, 21, 537-542.

Zuber, N. and Findlay, J.A. (1965) "Average Volumetric Concentration in Two-Phase Flow Systems," J. Heat Transfer, 87, 453-468.

FIGURES

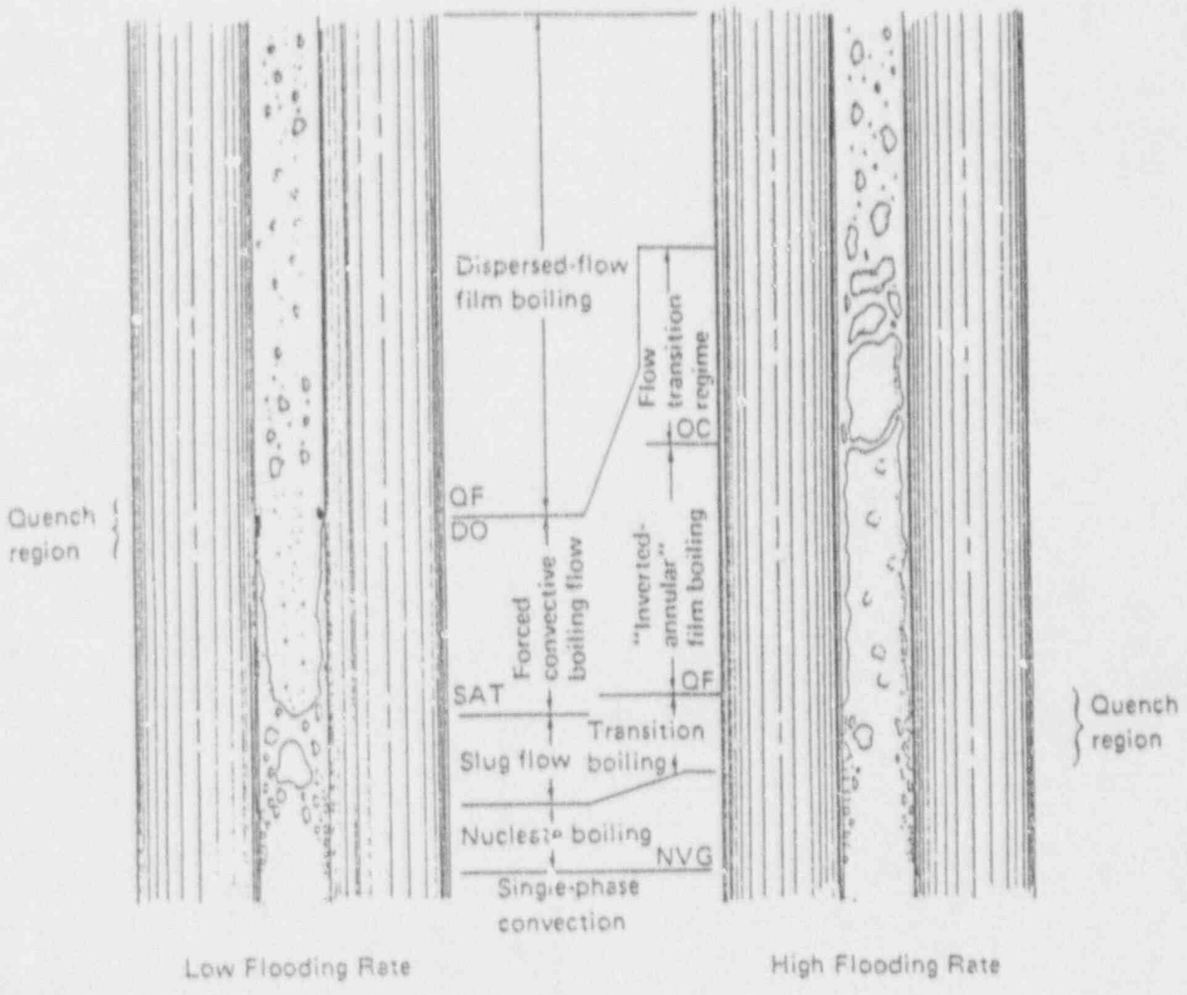

Fig. 1 Flow and heat transfer regimes observed during reflooding (Yadigaroglu, 1979)

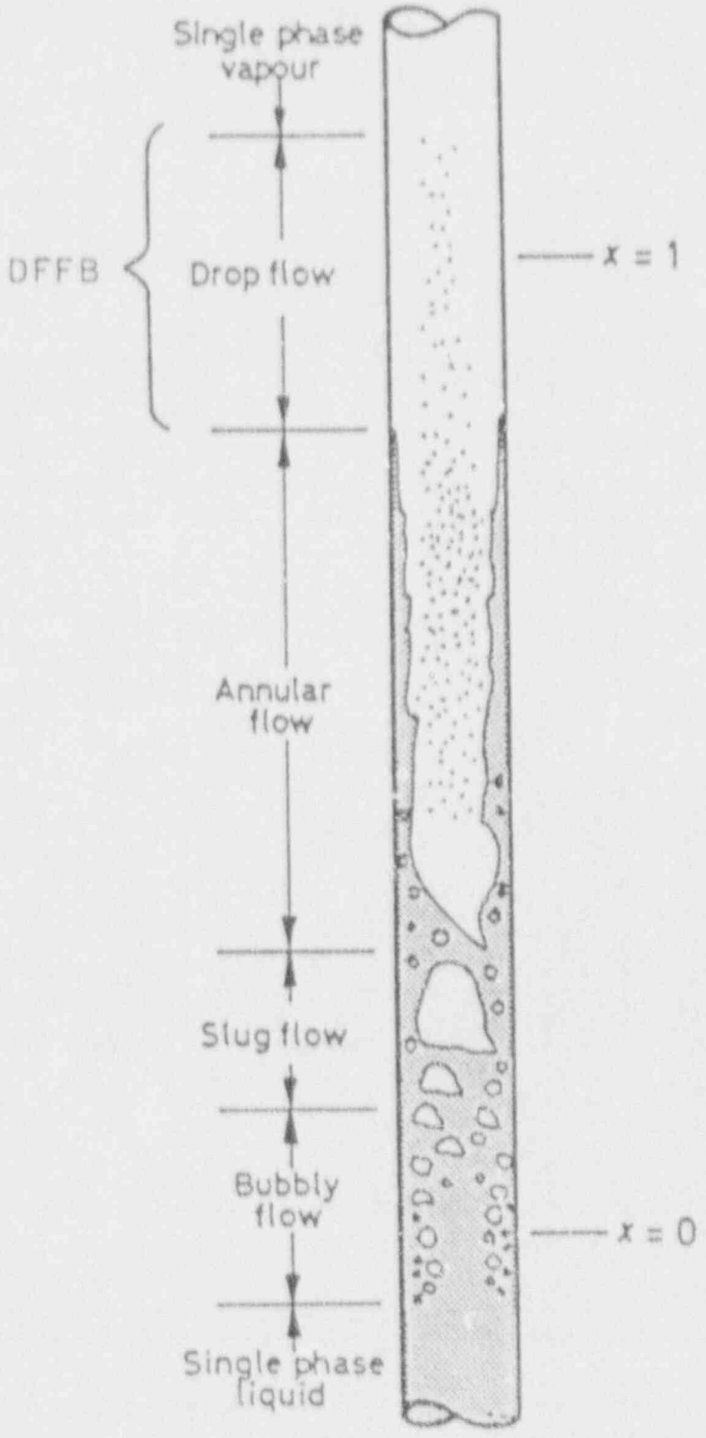

Fig. 2 Flow patterns in a vertical evaporator tube (Collier, 1981)

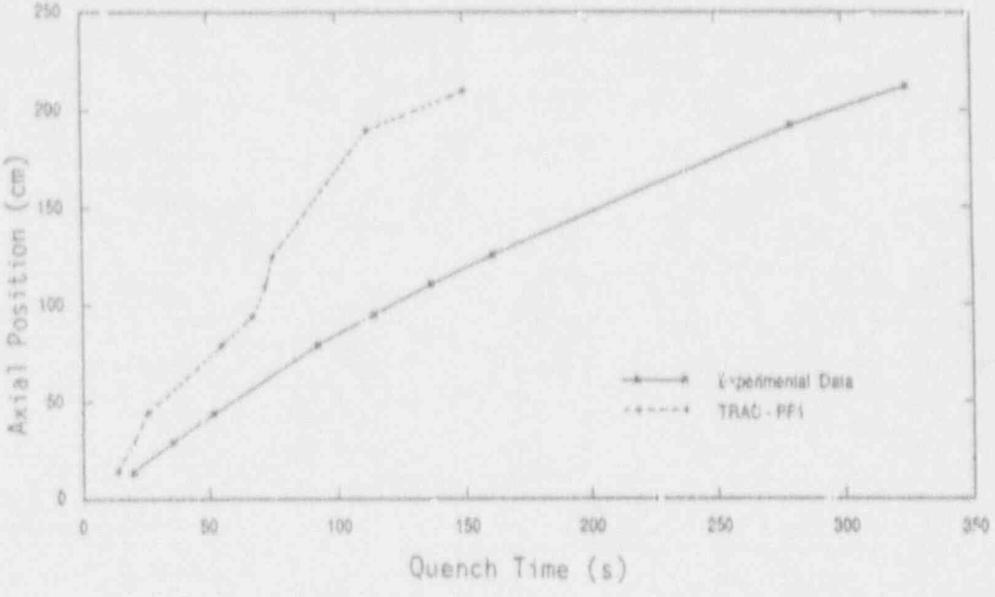

Fig. 3 Comparison of experimental quench front progression with TRAC-PF1/MOD0 prediction. Reflex test 92 (Afifi, 1985)

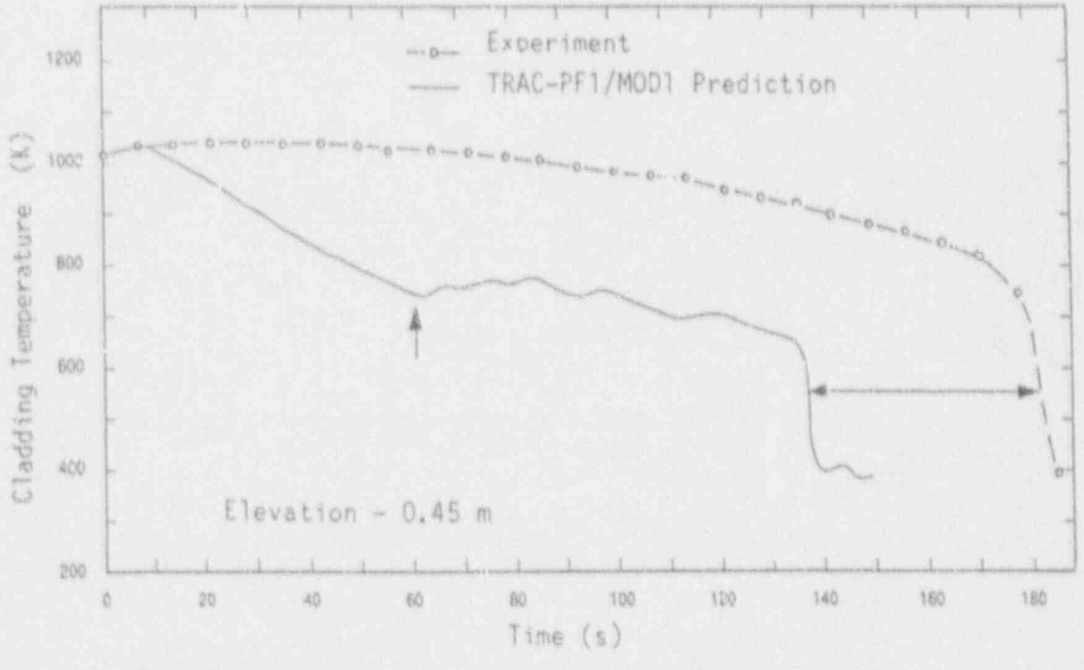

Fig. 4 Comparison of measured wall temperature with TRAC-PF1/MOD1 prediction. Lehigh Univ. rod bundle test 02/24/85-6 (Chen, 1987)

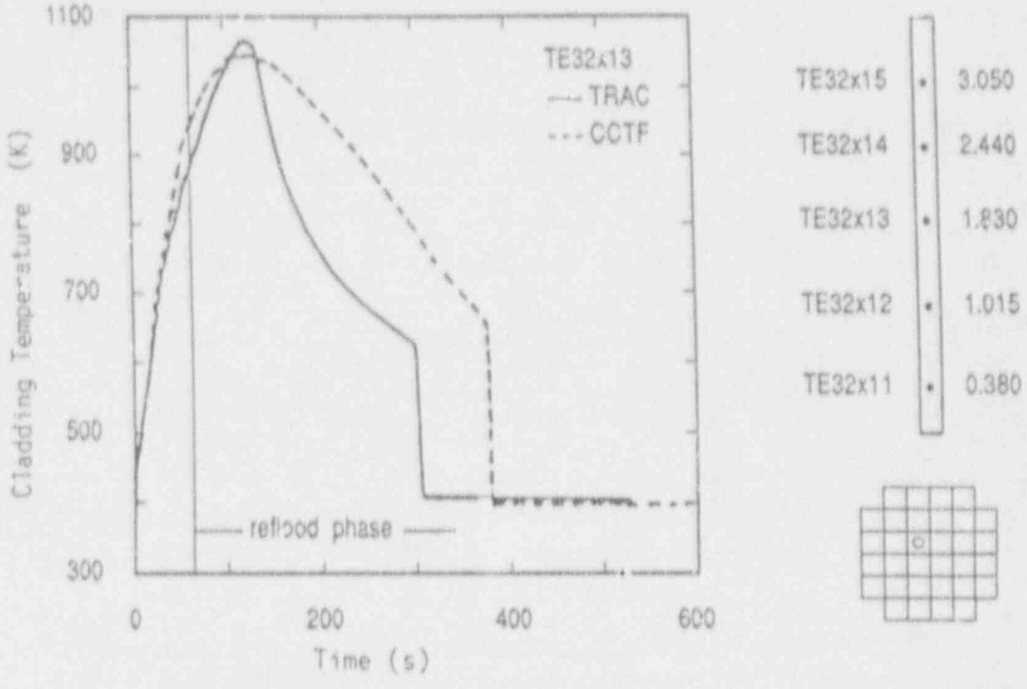

Fig. 5 Comparison of measured wall temperature with TRAC-PF1/MOD1 prediction. CCTF radial power profile test (Akimoto and Murao, 1987)

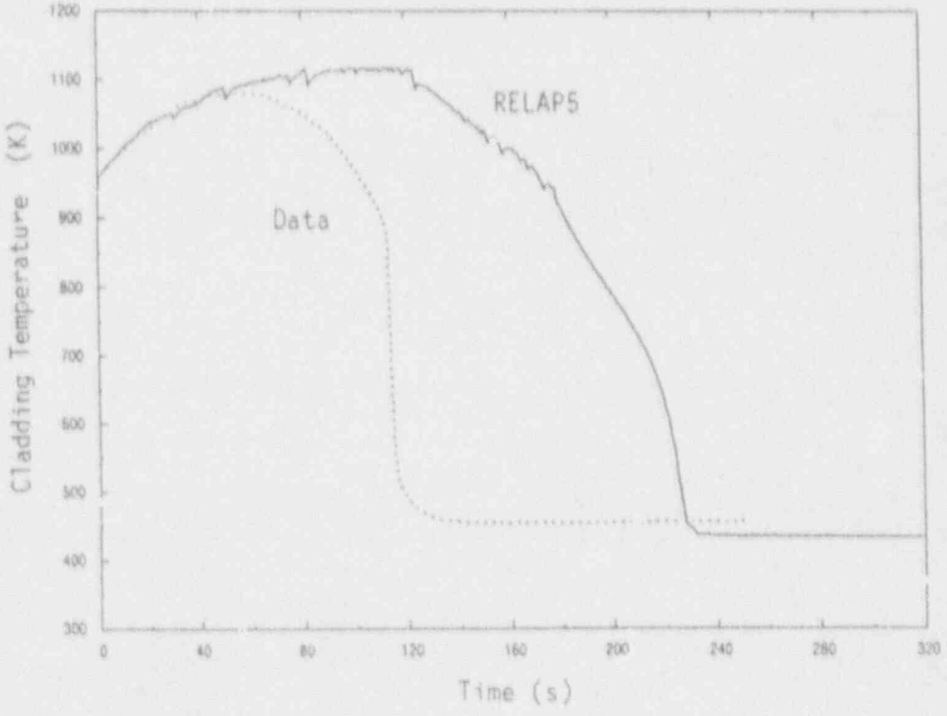

Fig. 6 Comparison of measured wall temperature with RELAP5/MOD2 prediction. NEPTUN test 5036 (Analytis et al., 1987)

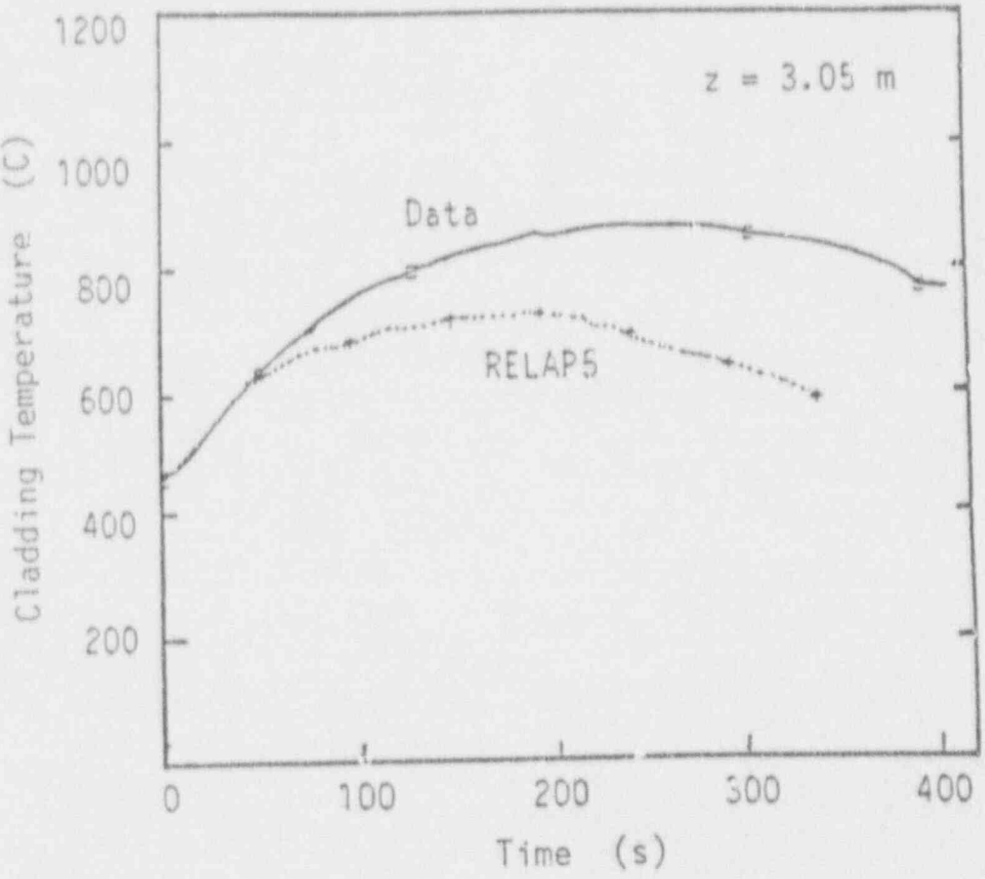

Fig. 7 Comparison of measured wall temperature with RELAP5/MOD2 prediction. FLECHT-SEASET test 31504 (Hassan, 1987)

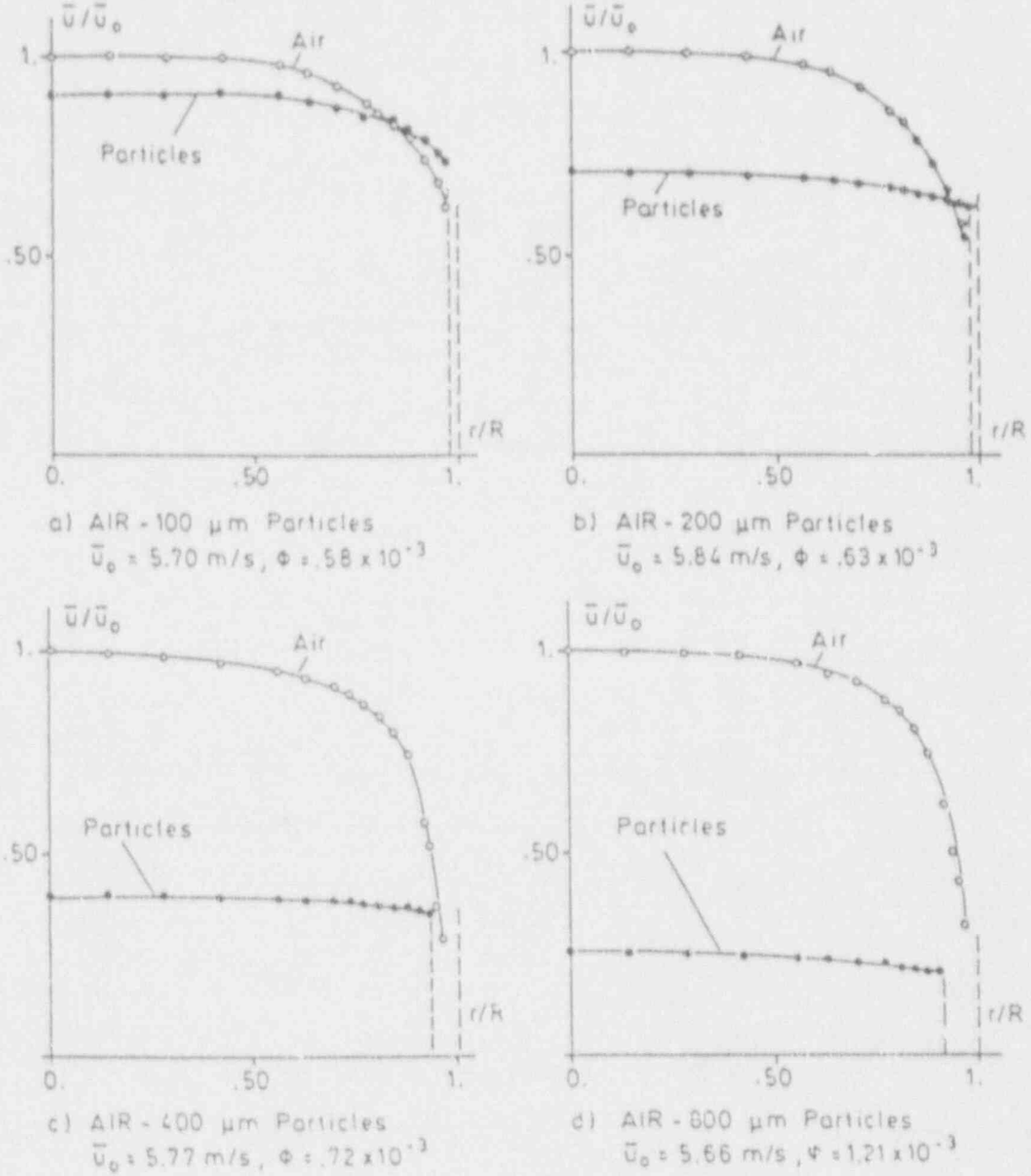

Fig. 8 Radial profiles of axial velocity of air and particles in highly dispersed (low particle concentration Φ) tube up-flow (Lee and Durst, 1982)

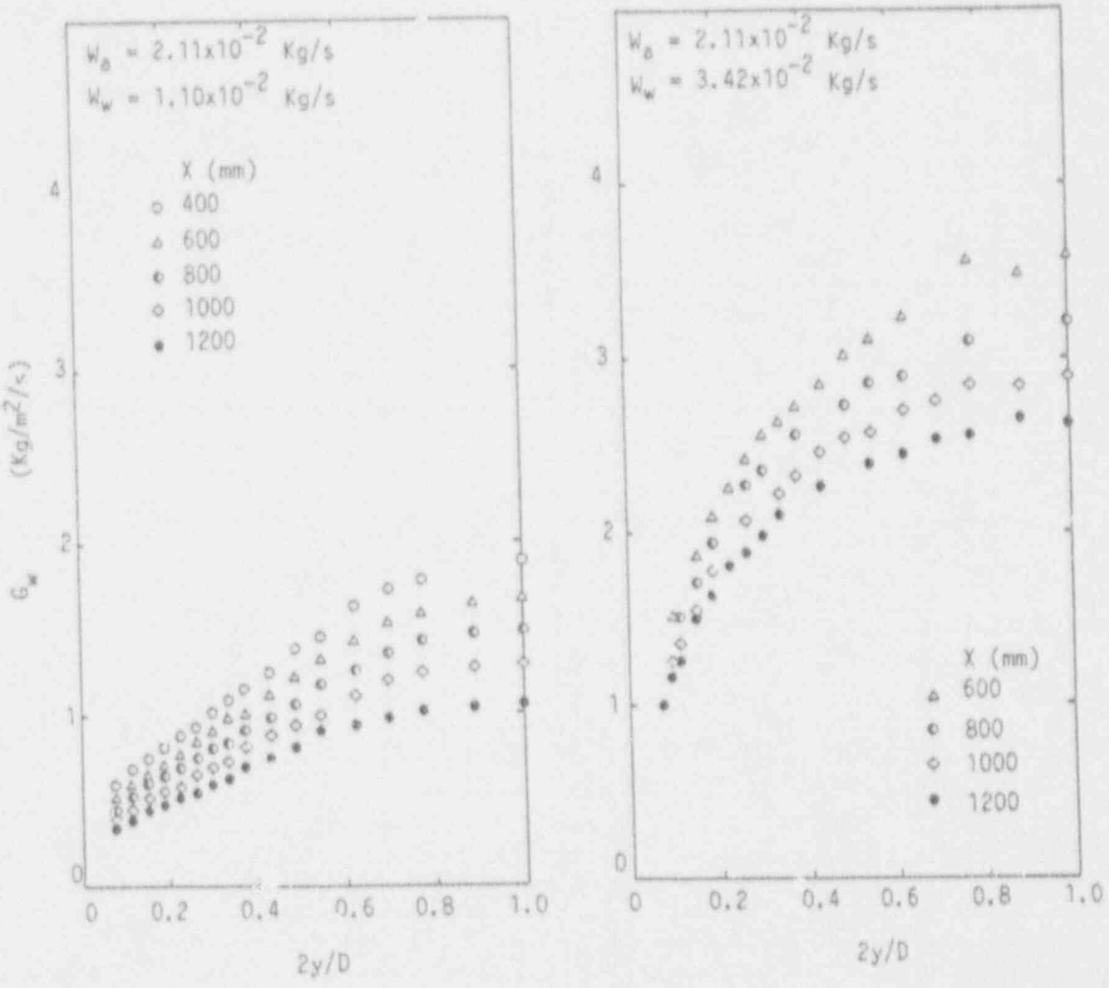

Fig. 9 Radial droplet mass flux distribution in vertical down-flow of air (a) and water (w). Re_a = 55 000 (Hagiwara et al., 1980)

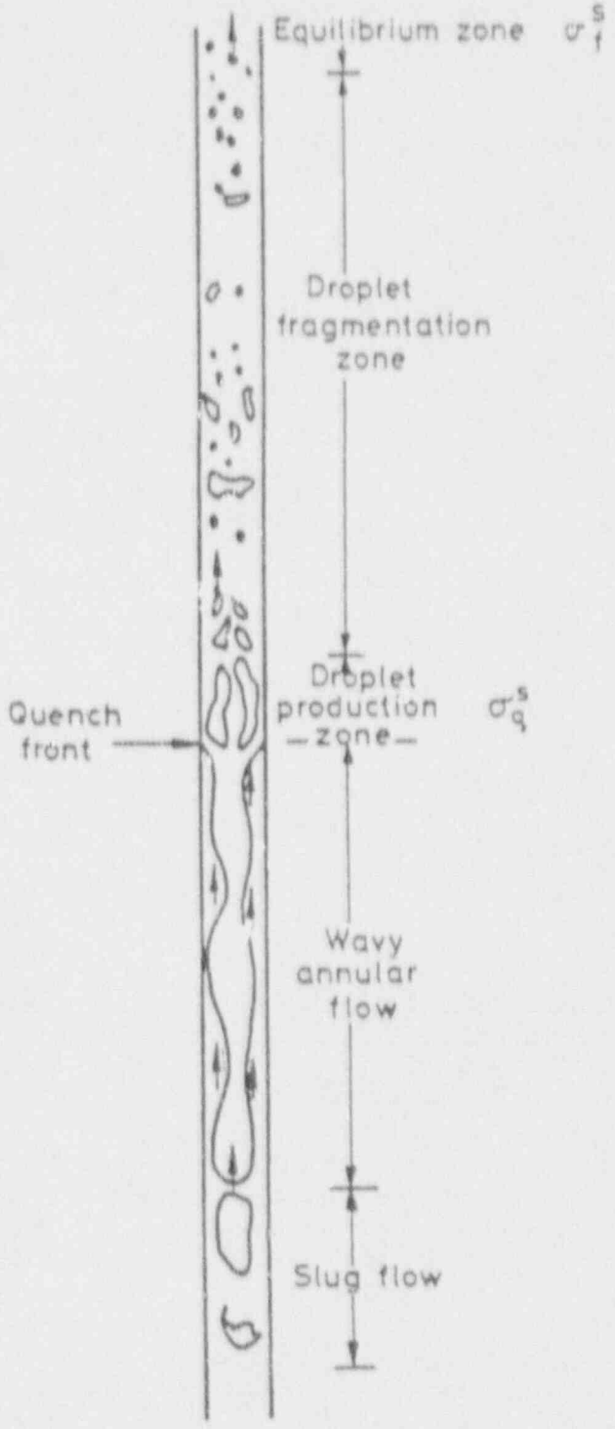

Fig. 10 Flow pattern observed in single tube reflooding tests (Ardron and Hall, 1981)

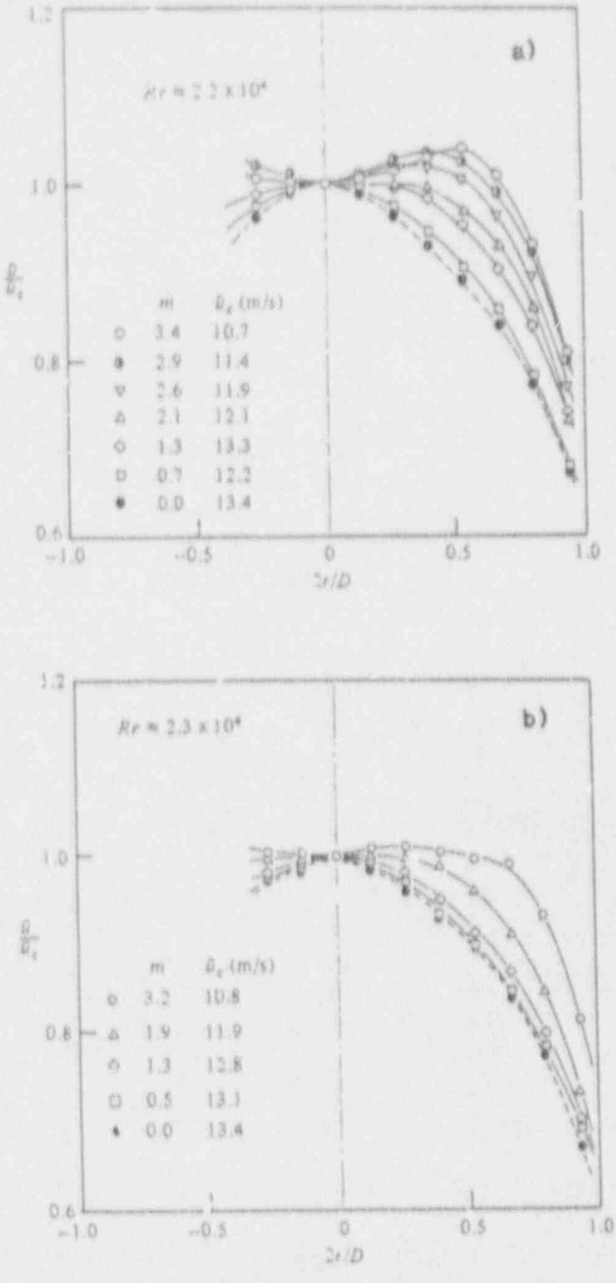

Fig. 11 Mean air velocity distribution in the presence of particles. a) d = 500 μm, b) d = 200 μm (Tsuji et al., 1984)

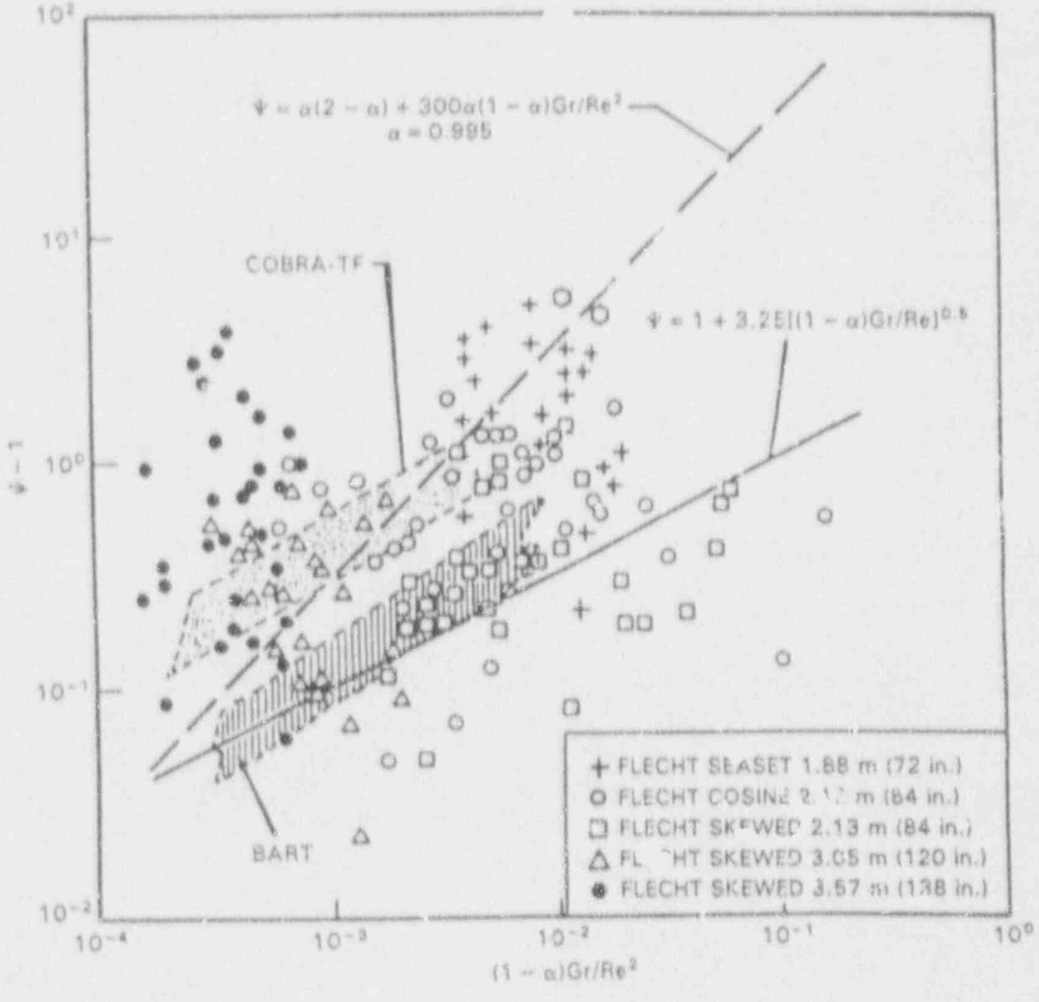

Fig. 12 Comparison of heat transfer coefficient enhancement models (ψ = two-phase HTC/single phase HTC) for dispersed flow and reflood data (Hassan, 1986)

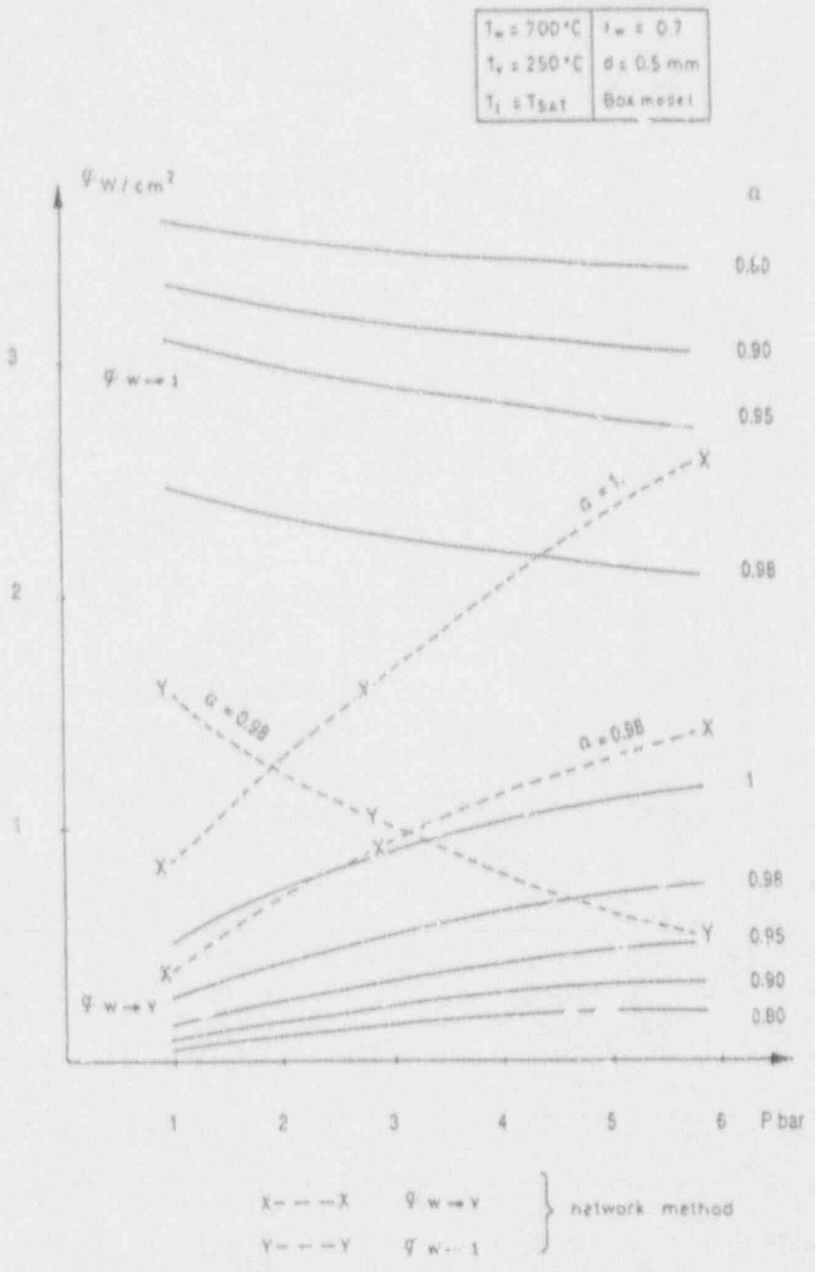

Fig. 13 Comparison of radiative heat fluxes to vapor and droplets calculated by two different models (Box model results from Deruaz and Petitpain, 1976)

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11. ABSTRACT (200 words or less)

Dispersed Flow Film Boiling is the heat transfer regime that occurs at high void fractions in a heated channel. The way this heat transfer mode is modelled in the NRC computer codes (RELAP5 and TRAC) and the validity of the assumptions and empirical correlations used is discussed. An extensive review of the theoretical and experimental work related with heat transfer to highly dispersed mixtures reveals the basic deficiencies of these models: the investigation refers mostly to the typical conditions of low rate bottom reflooding, since the simulation of this physical situation by the computer codes has often showed poor results. The alternative models that are available in the literature are reviewed, and their merits and limits are highlighted. The modifications that could improve the physics of the models implemented in the codes are identified.

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