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Recommendations for Modeling Upper Plenum Injection System for a Pressurized Water Reactor

by

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ABSTRACT

The present work examines the unique heat transfer modeling problems associated with analysis of performance of an upper plenum emergency core cooling injection system during a postulated Loss-of-Coolant Accident (LOCA) in a pressurized water reactor.

The particular system under study consists of four 4-inch pipes conveying emergency core cooling water from outside the pressure vessel through four spare control and rod mechanism housings and internal pipings, discharging the water directly over the top of the reactor core.

Previously, it has been assumed that the water injected into the upper plenum passes through the reactor core to the lower plenum without any heat interaction with the fuel rods during a LOCA. This simplified model neglects the many beneficial and adverse effects that accompany the upper plenum injection.

This study is undertaken to examine the following items

- heat transfer to the emergency core coolant before reaching the core
- flow and heat transfer through the reactor core
- reflood heat transfer.

The phenomena occurring will be delineated and recommendations made for calculating both conservative and best estimate values. The method of solving these problems will be illustrated in a series of examples given in the appendices.

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NOMENCLATURE

SYMBOLS	DEFINITION	UNITS
A	Area	ft^2
a	Thermal Diffusivity	ft^2/hr^2
C_d	Discharge Coefficient	
C	Specific Heat	Btu/lbm-^oF
d,D	Diameter	ft
g	Gravitational Acceleration	ft/hr^2
G	Mass Velocity	lbm/ft^2-hr
h	Heat Transfer Coefficient	Btu/hr-ft^2-^{o} F
H	Height of Pool Depth	ft
h_{fg}	Heat of Vaporization	Btu/lbm
j_f^*	Dimensionless Volumetric Liquid Flux $j_f^* = G(1-X)[\, g\ D\ \rho_f(\rho_f - \rho_g)]^{-\frac{1}{2}}$	
j_g^*	Dimensionless Volumetric Vapor Flux $j_g^* = G\ x\ [\, g\ D\ \rho_g(\rho_f - \rho_g)]^{-\frac{1}{2}}$	
k	Thermal Conductivity	Btu/hr-ft-^oF
L	Axial Length	ft
$\dot{m}$	Mass Flux	lbm/hr
P	Pressure	lb/ft^2
Pr	Prandtl number	
q	Heat Rate	Btu/hr
q/A	Heat Flux	Btu/hr-ft^2
$\dot{Q}$	Volume Flow Rate	gmp
r,R	Radius	ft
Re	Reynolds Number	

NOMENCLATURE (continued)

SYMBOLS	DEFINITION	UNITS
T	Temperature	^oF
ΔT	Temperature Difference	^oF
U	Overall Heat Transfer Coefficient	Btu/hr-ft^2-^oF
V	Velocity	ft/sec
x	Quality	
y	Local Value	ft

GREEK SYMBOLS	DEFINITION	UNITS
α	Void Fraction	
ρ	Density	lbm/ft^3
μ	Absolute Viscosity	lbm/hr-ft
Γ	Condensation Rate	lbm/hr
ε	Droplet Heat Transfer Effectiveness	
ε_*	A Dimensionless Constant	
ϕ	Contraction Coefficient of a Jet Stream	

SUBSCRIPTS

b	Bulk Fluid
d	Droplet
direct	Direct Contact Condensation
DB	Dittus-Boelter Correlation
ECC	Emergency Core Coolant
f	fluid
h	Hydraulic
i	Inside

NOMENCLATURE (continued)

SUBSCRIPTS	DEFINITION	UNITS
inlet	Inlet	
o	Outside	
pitch	Pitch	
rod	Rod	
ref	Reference	
lower	Lower	
Sat	Saturation	
tube	Tube	
turb	Turbulent	
upper	Upper	
v,g	Vapor	
w	Wall	
y	Local	
∞	Fully Developed Value	
1	Initial	

Chapter 1

Introduction

The Loss of Cooling Accident

In the safety evaluation of a nuclear power reactor, all credible accidents involving risks of release of radioactivity to the environment have to be considered. One of the most serious accidents is a major loss-of-coolant accident (LOCA), the design basis accident, which is postulated to occur when there is a double-ended (guillotine) break in one of the largest pipes connected to the reactor vessel.

Consider the sequence of events following an instantaneous double-ended break located near the reactor vessel of a cold leg (inlet) pipe in a Pressurized Water Reactor (PWR) system, as shown in Figure 1. Immediately, high-pressure, subcooled coolant is expelled from the break, rapidly reducing the pressure in the reactor vessel. During this period, local voiding causes burn-out to occur and cladding temperature begins to increase rapidly. The energy generated by the decay of the fission products after reactor shutdown causes the fuel cladding temperatures to rise even further. Finally, at a high enough temperature a chemical reaction between the cladding, the vapor, and the molten fuel occurs and results in severe damage to the core.

In order to prevent such a catastrophe from occurring, numerous, redundant safety systems are designed into each nuclear reactor. These systems, called Emergency Core Cooling Systems (ECCS), are designed so that the reactor can be safely shut down and the essential heat transfer geometry of the core preserved following the accident.

The ECCS Description

There are numerous different ECCS designs in the present commercial nuclear reactors. However, in this study, a particular emergency core cooling system designed for a four-reactor coolant loop Pressurized Water Reactor (PWR) will be examined. This particular ECCS is comprised to two separate systems, namely the High Pressure Safety Injection System (HPSI), and the Low Pressure Safety Injection (LPSI) system. These systems automatically deliver borated water to the reactor vessel for cooling the core during a loss of coolant accident.

The HPSI is designed to inject water at a rated flow of 1750 gpm per pump at a pressure of 470 psig into each of the four cold legs. The LPSI, on the other hand, consists of four tubes conveying emergency core cooling water from outside the pressure vessel through four spare control rod mechanism housings and internal piping, discharges the water directly over the top of the reactor core. The Low Pressure Safety System pumps are rated to deliver 5500 gpm at a pressure of 350 psig. Both HPSI and LPSI systems are activated automatically after low pressurizer pressure and low water level signals are detected. Operation of the LPSI and HPSI systems can be activated within 3 seconds and 13 seconds, respectively, after sensing the ECCS signal, provided outside electrical power is available. If outside power is not available, there is an additional 10 seconds delay in starting up the plant diesels to provide the backup power. Figure 2 illustrates the internals of a PWR vessel and the respective location of the LPSI (core deluge) tubes in the upper plenum. The design parameters of a four-loop ECCS are presented in Table 1.

Statement of Problem

Up until now in the analysis of the ECC upper plenum injection system, it has been assumed that the water injected into the upper plenum from the LPSI passes through the core without any heat interaction with the fuel rods, and reaches the lower plenum in zero delay time [16]. This simplified model clearly neglects the many beneficial and some adverse effects that accompany upper plenum injection. For this reason, the present work was undertaken. To be precise, we want to address the problem of how does the ECC make it from the upper plenum where it is injected to the lower plenum. There are a number of unique reflood heat transfer modeling problems associated with the LPSI (core deluge) system. We will proceed to describe these problems in the following chapters and sketch the solution techniques, each with a tangible physical basis. The method of solving these problems will be presented in a series of example calculations. It is clear that the complete solution to the problem of how ECC water gets to the lower plenum must incorporate the results of

Table 1

Typical Four-Loop Plant ECCS Design Parameters

High Pressure Safety Injection	Four-Loop
Number of Pumps	2
Design Pressure (Psia)	950
Design Temperature (^{o}F)	200
Design Flow Rate/Pump (gpm)	1750
Maximum Flow Rate (gpm)	2750
Injection Locations	Cold Leg

Low Pressure Sagfety Injection	
Number of Pumps	2
Design Pressure (Psia)	250
Design Temperature (^{o}F)	100
Design Flow Rate/Pump (gpm)	5500
Injection Locations	Upper Plenum
Pump Flow at Reduced Pressure (30 Psia), (gpm)	5900

a blowdown calculation and consider both the flow into or out of the plenums and the variable pressure differences across the core. It is likely too that two-dimensional effects in the core are important also, as there is quite a range of temperatures found in the core late in blowdown. In a word, a complete loop code must be used along with some two- or three-dimensional modeling in the core.

Chapter II

Heat Transfer to the Emergency Core Coolant (ECC) Before Reaching the Core

The LPSI water enters the upper plenum through four vertically positioned four-inch pipes which extend about 3-1/2 feet (an assumed value used in Appendix A) below the upper support plate, and the injection nozzles are positioned about 6 feet above the upper core plate (see Figure 2). During a hypothetical LOCA, only one low-pressure pump is, conservatively, assumed to pump water, at a design rate of 5500 gallons per minute, into the upper plenum through these four separate nozzles. The velocity of the water jet entering the upper plenum is calculated to be 35 ft/sec in vertically downward direction.

During the hypothetical core cooling process, both the upper portion of the core deluge tubes and the four jets of water serve as a heat sink for condensing steam. The amount of steam passing through the steam generator and the resulting pressure drop in the steam generator will both be reduced. Any steam binding problem will also be reduced and the refill and the reflood will proceed more rapidly. In any case, a rapid flow of steam through the core will help cool it.

Condensation on the Upper Portion of the Core Deluge Tube

The physical model under consideration is a vertical tube exposed in a saturated steam environment with ECC flowing down inside the tube as shown in Figure 3.

For a vertical tube exposed to a convection environment on its inner and outer surfaces, the overall heat transfer by combined conduction and convection can be expressed by the relation

$$q = U_i A_i \Delta T_{overall} \tag{1}$$

where the overall heat transfer U_i based on the inside surface of the tube A_i is

$$U_i = \left[\frac{1}{h_i} + \frac{A_i \ln\left(\frac{r_o}{r_i}\right)}{2\pi K_s L} + \frac{A_i}{A_o h_o} \right]^{-1} \tag{2}$$

During the emergency core cooling process, the core deluge coolant vessel through the tube is on the order of 35 ft/sec, a value which justifies the assumption of turbulent flow. For forced-convection, turbulent flow in tubes, the heat transfer coefficient, h_i, using McAdams correlation [17], is

$$\frac{h_i d_i}{K_b} = 0.023\left(\frac{Gd_i}{\mu}\right)_b^{0.8} \left(\frac{\mu C_p}{K}\right)_b^{0.4} \tag{3}$$

Here the subscript b refers to the bulk fluid temeperature at which the particular fluid property is evaluated.

The analysis of the heat transfer rate associated with film condensation on a vertical surface was reported by Rohsenow [23].

Rohsenow suggested that for a vertical tube with Pr > 0.5 and $c\Delta T/h_{fg} \leq 1.0$, the average heat transfer coefficient on the outer wall, h_o, takes the following form

$$h_o = 0.943\left[\frac{g\rho_f(\rho_f - \rho_g)k^3\,(h_{fg} + 0.68\;C_p\Delta T)}{L\,\mu\,(T_{sat} - T_w)}\right]^{\frac{1}{4}} \tag{4}$$

where the fluid properties k, and c are evaluated at the following reference temperature

$$T_{ref} = T_w + 0.31(T_{sat} - T_w) \tag{5}$$

From Equations (2), (3), (4) and (5), the overall heat transfer can be evaluated as

$$q = U_i(2\pi r_i L)(T_{sat} - T_b) \tag{6}$$

The average steam condensation rate, $\dot{m}_{tube}$, on a core deluge tube of length L is

$$\dot{m}_{tube} = \frac{q}{h_{fg}} = \frac{U_i(2\,\pi\,r_i L)(T_{sat} - T_b)}{h_{fg}} \tag{7}$$

An example calculation for the steam condensation rate on a core deluge tube is presented in Appendix A.

Direct Contact Condensation of Steam on Water Jets

Once the ECC flows out of the core deluge nozzles in the upper plenum, the coolant is immediately exposed to a saturated or superheated vapor environment. It is possible to condense the steam on the jets of water on its way to the top of the core. The condensation rate is determined by the rate of transporting the heat from the surface of the jet to the center of the jet. The transport of heat from the condensation surface involves both molecular and turbulent diffusion. The theoretical and experimental investigations on the heat transfer from the steam to the liquid jet may be found in [1-9]. In Figure 4, the average values of heat transfer coefficient for a jet length of 800 mm are plotted as a function of the velocity of flow according to the formulas of G. Abramovich (curve 1), S.S. Kutaladze (curve 2), and experimental data of N. Zinger (curves 3, 4) [2]. As seen from Figure 4, the values of heat transfer coefficient calculated using the method proposed by S.S. Kutaladze represent a lower limit. Consequently, the amount of steam being condensed on the water jet can be estimated conservatively using this formula.

The following problem (Fig. 5) is examined. A continuous cylindrical free-flowing jet in a motionless vapor with an initial vessel V_1 flowing from a nozzle of radius R_1. The equation of motion for the jet takes the following form

$$\rho g - \frac{dP}{dy} = \rho V_y \frac{\partial V_y}{\partial y} + \frac{V_y - V_v}{\pi R_y^2 g} \cdot \frac{d\Gamma}{dy} \tag{8}$$

The equation of continuity of the jet is given by

$$\pi R_y^2 \rho V_y = \pi R_1^2 \rho V_1 + \int_0^y d\Gamma \tag{9}$$

where

Γ is the rate of condensation

The equation for the distribution of heat in cylindrical coordinates assuming the radial temperature gradient is much larger than the axial gradient is

$$V_y \frac{\partial T}{\partial y} = \frac{K + K_{turb}}{C_p \rho} \left(\frac{\partial^2 T}{\partial R^2} + \frac{1}{R}\frac{\partial T}{\partial R}\right) \tag{10}$$

The instantaneous coefficient of heat transfer to a jet is defined as

$$h_y = -\left(\frac{K + K_{turb}}{T_v - \bar{T}_x}\right)\left(\frac{\partial T}{\partial R}\right)_{R_y} \tag{11}$$

S.S. Kutaladze solved the above heat transfer problem and obtained the following expression for the heat transfer coefficient

$$h_y = \left[\frac{K}{R_1\sqrt{\phi}}\left(1 + \frac{2\phi^2 gy}{V_1^2}\right)^{\frac{1}{4}} + \varepsilon_* C_p \rho \sqrt{\left(\frac{V_1}{\phi}\right)^2 + 2gy}\ \right] \cdot \tag{12}$$

$$\cdot \frac{\sum_{i=1}^{\infty} e^{-\beta_i^2} f\left(\frac{y}{R_1}\right)}{\sum_{i=1}^{\infty} \frac{2}{\beta_i^2} e^{-\beta_i^2} f\left(\frac{y}{R_1}\right)}$$

where

$$f(\frac{y}{R_1}) = \frac{a}{V_1 R_1^2}\, y + \frac{2\varepsilon_* V_1^2}{5\phi^{5/2} g R_1}\left[\left(1 + \frac{2\phi^2 gy}{V_1^2}\right)^{5/4} - 1\right] \tag{13}$$

$$a = \frac{K}{\rho C_p} \tag{14}$$

ϕ = contraction coefficient of the jet at the nozzle (see Appendix B). For $f(y/R_1) > 0.05$, Kutaladze showed that the heat transfer coefficient can be confined to the first term of the series only namely,

$$\frac{\sum_{i=1}^{\infty} e^{-\beta_i^2 f(\frac{y}{R_1})}}{\sum_{i=1}^{\infty} \frac{2}{\beta_i^2} e^{-\beta_i^2 f(\frac{y}{R_1})}} \simeq \frac{2}{\frac{1}{2}} = 2.892 \tag{15}$$

Adopting Kutaladze's values of ε_* equal to 0.0 and 5×10^{-4} for a laminar flow, and a turbulent jet, respectively, a simplified equation for the heat transfer coefficient with turbulent flow of the water jet is

$$h_y = (2.892)(5.0\text{x}10^{-4})\, C_p \rho \sqrt{\left(\frac{V_1}{\phi}\right)^2 + 2gy} \tag{16}$$

As shown above, the heat transfer coefficient is directly proportional to the initial velocity and the square root of the length of the jet stream.

The average heat transfer coefficient along the jet length, $\overline{h}$, can be calculated as follows:

$$\overline{h} = \frac{1}{L} \int_0^L h_y d_y = (4.82 \times 10^{-4})(\frac{C_p \rho}{gL})\{ [(\frac{V_1}{\phi})^2 + 2gL]^{3/2} - (\frac{V_1}{\phi})^3 \} \qquad (17)$$

The steam condensation rate by direct contact method, $\dot{m}_{direct}$, is then

$$\dot{m}_{direct} = \frac{\overline{h}(2\pi R_1 L)\ (T_{sat} - T_b)}{h_{fg}} \qquad (18)$$

An example calculation on direct contact condensation of steam on the water jet is presented in Appendix B. An upper bound for the direct contact condensation of steam on the water jets can be estimated by assuming the water is all heated to the saturation temperature as soon as it enters the upper plenum. In terms of the following energy balance:

$$\dot{m}_{upper\ condensation} = \frac{\dot{m}_{ECC}\overline{C}_p(T_{sat} - T_{inlet})}{(h_{fg})_{T_{sat}}} \qquad (19)$$

$$= \frac{\overline{\rho}_{ECC}\overline{C}_p(T_{sat} - T_{inlet})}{(h_{fg})_{T_{sat}}} \dot{Q}_{ECC}$$

An example calculation of the condensation of steam on the upper plenum is presented in Appendix C, and the graphical results are shown in Fig. 6.

Chapter III

Flow and Heat Transfer Through the Core

Jet Spreading and Pooling Above the Upper Core Support Plate

When the ECC water hits the upper core support plate several things might happen. The steam flow up through the core may be so large that the water cannot flow back through the core at all. There is a possibility that the steam velocity up will be low enough so that some of the some of the water can dribble down, i.e. we are on the flooding line. Possibly the steam velocity will be so low that the water simply pours through the core. Which one of these occurs depends on how the rest of the system is behaving and the decision as to what is actually happening can only be made on the basis on a complete system calculation. In this section we shall look at the phenomena that can occur and suggest in which these phenomena can be modeled.

The first item to look at is what happens to the water when it hits the top of the core. Referring to Figure (2) it is evident that the entire upper plenum is filled with a jungle of core support columns and guide tubes. The flow jets down these and then can do several things when it gets to the top of the core. It can shoot straight through the core. If the top of the bundle is clean enough, hydrodynamically, this will happen. This is very unlikely however as the upper plenum internals were not designed with this in mind. These are mixing vanes in the way.

This possibility does provide a minimum jet area however so is useful for scoping calculations. Assuming this is the case, however, the water's velocity would be unaltered as the core was approached and just enough ECC jet spreading would occur so that the hole area for the water entering was equal to the jet area just above the core. To be consistent with this idea, the minimum area for the down flow region at the top of the core would just equal the cross-sectional area of the jet immediately above it in the upper plenum.

More likely, however, the jet will hit the top of the core and spread. Because the upper plenum is so complex, it is proposed that one should calculate the area over which the jet will spread as one might calculate flow between a series of leaky egg crates. Let me describe this calculation in greater detail.

Imagine an egg crate with holes in the separators and bottom. If water were dumped into one of the separators, it would pool up to a depth so that just as much water was leaving as entering. The water leaving out the bottom, for a first approximation would leave as a stream with the velocity $\sqrt{2gh}$, while the jet area would equal the hole area times a discharge coefficient. That is

$$\dot{m} = \rho AV = \rho_f A C_d \sqrt{2gh} \tag{20}$$

C_d would depend on the geomtery, but would probably be about .6. A similar expression would be appropriate for the holes at the sides of the box. The flow leaving one box would enter the next one and would either run down into the core or pool up and go into the adjoining box. In this way once could calculate how the pool would spread before it ran down through the core.

A very similar process occurs when one fills an ice tray under the tap. The section under the jet has the highest level, level limited by flow out the bottom and top. When it gets full enough, the water leaks out the bottom or top into the adjoining cube and so forth. Except for the absence of holes on the sides of the separators in the ice tray, the process is exactly the same.

The hydraulic properties of a complex geometry such as the upper plenum of a reactor can only be estimated. If the details of jet spreading turn out to be important, the only real answer is the one obtained from the experiment. One would guess, however, that if the pool ever became deep enough so the flow area for cross flow due to slots, gaps between core support columns and control rod guide tubes and so forth became comparable to the projected flow area of the core, the pool on top would be in approximately hydrostatic equilibrium. This is another easily calculated limit. This is quite likely to happen if the flow up through the core support plate exceeds the flooding velocity (a quantity to be described in the next section).

Flooding and Drainage Through the Upper Core Support Plate

There are really two quantities that determine how rapidly the water drains through the core, the pool depth and the steam velocity up. When the pool depth is greater than the hole diameter and there is upflow of steam through the holes, the pool depth does not affect the flow through the holes. The steam velocity then limits the back flow through the core. In this section we shall show the down flow rate as a function of these two variables.

Flooding has been studied for many years. Much of what we know is summarized in Reference 14 where a flooding criterion is given for round tubes. The relevant equation is

$$j_f^{*\frac{1}{2}} + j_g^{*\frac{1}{2}} = C \tag{21}$$

where

$$.7 < C < 1.$$

The constant C depends on geomtery so that one would expect the value to be slightly different for slots than holes, for instance. One also finds that multiple holes tend to have higher flooding velocities than single holes so that the flooding equation

$$j_f^{*\frac{1}{2}} + j_g^{*\frac{1}{2}} = 1 \tag{22}$$

has been chosen as the best estimate of the flooding limit. This equation has been plotted on the right-hand side of Figure (7) for the conditions shown in Appendix D. These conditions are typical of reflood.

As can be seen on Figure (7), there is no downflow of liquid through the core for vapor velocities greater than about 12 ft/sec. This has some important consequences.

The rate at which ECC is dumped into the upper plenum is well above that needed to sustain a steam flow of 7.5 ft/sec maximum (the flooding limit Fig. 7) through the core if all the subcooling in that water is removed. The assumption that all the subcooling is removed is probably a good one because direct contact heat transfer in the upper plenum is so good. I believe the water will pool up in the upper plenum and the excellent heat transfer between steam and cold water will insure that the high steam flow up through the core will persist until the pressure in the system drops below that in the containment. At that time air will flow through the break into the upper plenum and the condensation rate will be drastically reduced. It is not clear to me whether the upper plenum will fill before this occurs or not.

If the upper plenum does fill, the ECC water will either be forced down through the core or out the break or into intact loops. Only a complete system code will be able to predict what will happen in this case. No matter what happens though, vigorous steam flow up through the core or ECC flowing down through the core are going to provide good cooling. This cooling should be accounted for.

Returning now to the calculation of the flow through the core. A recent work, Reference [19] describes the flow through holes as a function of pool depth. (The expressions in this paper are appropriate for a constant pressure environment.) The drainage can be divided into two parts, drainage from a shallow pool and drainage from a deep one. For a shallow pool

$H/D \leq 0.4$ and

$$V_f = 2.66 \sqrt{gD_h}\,(H/D)^{1.5} \tag{23}$$

For draiange from a deep pool

$0.4 \leq H/D \leq 3$

$$V_f = 1.6 \sqrt{gD_h}\,(H/D)^2 \tag{24}$$

The results of these calculations are shown on the left side of Figure (7) which is the axis where the steam velocity is zero. A series of curves are faired in connecting these answers to the ones from the flooding calculation. The details are shown in Appendix D.

Before passing on to other matters, there should be some discussion of the dimensions appearing in the flooding and drainage equations. Flooding can occur anywhere. It is most likely to occur at the smallest flow area, however, as that is where the vapor velocity up is likely to be largest. For a well-designed core and upper plenum, this area is either in the core or at the upper core plate (Fig. 2). Therefore, the hydraulic diamter in the upper core plate or the core itself, whichever is smaller, is the dimension to use in the flooding equation. A hydraulic dimater is not a sufficient specification of the geometry to predict flooding or drainage accurately, an experiment is needed too. If one needs an answer for the flooding velocity which has to be closer than a factor of 2, it is necessary to run an experiment to determine it. This experiment should be run in the correct geometry with steam and water conditions appropriate to the conditions of interest. The flooding correlations are only good to a factor (2) in untested geometries.

Similar doubt exists about the dimensions for the drainage equations shown earlier. They are valid for round hoes. We are interested in a very complex slot and hole geometries, however. Using a hydraulic diameter is again only a guess. It is obvious that if a precise answer is necessary, then experiments must be run. In the example worked out in Appendix D, the hydraulic diameter used in the drainage equation is that for the bundle which is the only one we had. Actually, the one for the upper core plate is probably more appropriate but one that very likely differs only slightly from the one used.

Downflow Heat Transfer in the Core

Once the ECC makes into the core, the heat transfer from the fuel rods to the ECC results in tremendous amounts of vapor generation. In order to find out whether the amount of steam being generated in the hot core is sufficient to cause the flow to flood and expel the ECC, a downflow heat transfer model is needed.

Paul Robershotte's theoretical and experimental work on downflow post-critical heat flux transfer of low-pressure water is recommended. In this case it should be used with the assumption that ECC is in saturated state [11]. By the time it drains into the core, it probably is.

The physical model under consideration is based on a unit cell defined as the open space among four adjacent rods. The units cell is characterized by a circular tube of hydraulic diamter, D_h (see Figure 8). The post-critical heat flox can be analyzed after evaluating an effective vapor Reynolds number

$$Re_v = \frac{XGD_h}{\mu_v} \tag{26}$$

where the viscosity is evaluated at saturation pressure and temperature. For laminar downflow with $Re_v < 2000$

$$Nu_\infty = \frac{h_\infty D_h}{K_v} = 3.66 \tag{27}$$

the heat flux can be calculated as

$$q/A = h_\infty(T_w - T_{sat}) = (3.66)(\frac{K_v}{D_h})(T_w - T_{sat}) \tag{28}$$

the amount of vapor generated, G_v, as a function of the distance from the top of the core is derived as follows

$$Re_v = (XG)\left(\frac{D_h}{\mu_v}\right) = \frac{\dot{m}_v}{\left(\frac{\pi D_h^2}{4}\right)} \left(\frac{D_h}{\mu_v}\right) \tag{29}$$

but

$$\dot{m}_v = \frac{(q/A)(\pi D_h y)}{h_{fg}} \tag{30}$$

therefore

$$Re_v = \frac{4(q/A)D_h}{h_{fg}\mu_v}\left(\frac{y}{D_h}\right) \tag{31}$$

$$G_v = XG = Re_v\left(\frac{\mu_v}{D_h}\right) = \frac{4(q/A)}{h_{fg}}\left(\frac{y}{D_h}\right) \tag{32}$$

If $Re_v > 3000$, the flow is turbulent

$$(q/A)_{turb} = (q/A)_v + (q/A)_d \tag{33}$$

where $(q/A)_v$ is the wall to vapor heat flux and $(q/A)_d$ is the wall to droplet term

$$(q/A)_v = h_{DB}(T_w - T_{sat}) \tag{34}$$

$$h_{DB} = (0.023)\left(\frac{K_v}{D_h}\right)(Re_v)^{0.8}(Pr_v)^{0.4} \tag{35}$$

$$(q/A)_d = (1 - \alpha)\,\rho_f h_{fg} V_1 \varepsilon \tag{36}$$

where α is the void fraction; ρ_f, the density of the liquid; h_{fg}, the heat of evaporation; V_1, the perpendicular of deposition velocity and ε, the effectiveness or percentage of the drop evaporating. There are two restrictions on Equation (36). First, there must be sufficient vapor to ensure a turbulent, dispersed flow and second, Equation (36) is only applied when void is in excess of 98%. Since the major portion of void fraction along the flow path is less than 98%, it is therefore reasonable to neglect the $(q/A)_d$ term.

Combining Equations (31), (34) and (35), the turbulent heat flux takes the following form

$$(q/A)^{0.2}_{turb} = (0.023)\left(\frac{K_v}{D_h}\right)(Pr_v)^{0.4}(T_w - T_{sat})\left(\frac{4D_h}{h_{fg}\ {}_v}\right)^{0.8}\left(\frac{y}{D_h}\right)^{0.8} \tag{37}$$

the vapor generation rate, G_v, is

$$G_v = XG = \frac{4(q/A)_{turb}}{h_{fg}}\left(\frac{y}{D_h}\right) \tag{38}$$

We now have a method of calculating the downflow post-critical heat flux heat transfer in the reactor core. A sample calculation is presented in Appendix E and the results of heat flux vs. axial length and vapor generation rate vs. axial length are plotted in Figures 9 and 10, respectively.

Flow Through the Core

There are really three possibilities for flow through the core, namely up flow, counter flow, and down flow.

Up flow occurs when the flow of vapor (and perhaps entrained liquid) is up at a high enough velocity so that liquid in the upper plenum cannot drain back through the core. The flooding velocity at either the upper core support plate or in the upper part of the core is exceeded by the vapor.

Counterflow occurs for vapor velocities which are between the maximum flooding limit and zero. Liquid under these conditions liquid runs down from above and evaporates, sputters or falls through. Normally, because the core is hot, enough vapor will form so that the gross flow through the core will be appreciably altered. Except, in a one-dimensional way, there is no tool to handle this. There is an excellent chance, however, that the time spent by a PWR system in the region where the core vapor velocity is between the flooding velocity and 0 is so short that the effect on the course of the accident primarily the peak clad temperature, is not large. As it is not essential for a PWR to assume that there is counterflow in order to get water through the core I suggest that, for a first approximation, no counterflow be assumed to occur in the core and flow not start down until the vapor velocity out the top of the core drops below zero. We would then have downflow which is much easier to handle. It is also, probably, a conservative assumption.

Downflow will probably result in a large amount of liquid dumping through the core from above, giving excellent heat transfer an rapid quenching from both the top and the bottom.

The tools to calculate top quenching do not exist at this time. There is no reason to assume that they cannot be assembled, but the appropriate heat transfer coefficient matrix such as the one that is the heart of the REFLUX code, for instance, Reference [12] has not yet been assembled far down flow. The appropriate heat transfer coefficient relations and heat transfer regime transition criteria have not been developed or assembled either.

Even though we may choose to ignore counterflow in the core in a local sense, one can imagine counterflow in a global sense. This regime is likely to be important and should be considered. One could have a flow of liquid and vapor down through the cold regions of the core and vapor and perhaps entrained liquid up through the hot regions of the core. The drainage of the pool on the top of the core would cause a very rapid reflooding and quenching of the core. I think we should know how to calculate this. At this time it is only possible to make some recommendations on how to do it. This is a problem beyond most of the currently published codes.

At the very least we must divide the core into two radial and probably three axial nodes. Each node would have about equal area for flow and would be about the same length. Cross flows should be put in. Though I don't think mixing is very significant over the radial distances we're interested in, crossflow probably is. To calculate it, I'd assume the crossflow resistance was what one would calculate in going from the average radius of one radial node to the average radius of the other. The flow would first start down in the node that had an average velocity at the top that dropped below zero. The vapor formed would go where ever the plenum pressures drove it. I would guess though that there would be a big rush of steam out both ends of the core as a result of liquid dumping through it. The course of the transient would be calculated using local values of the mass velocity, quality and pressure to calculate the heat transfer coefficient much like that done by Bjornard in Reference [10].

This is a truly two dimensional problem. An attempt at a simplified one dimensional analysis failed because so much vapor was formed, with the assumed temperatures and flow rates, that the assumption of small radial flow was not even approximately correct. A two dimensional code and model is essential if one is to be successful in calculating the global counter flow that might occur under these circumstances. It is also necessary to do these calculations with the loops included as the direction the vapor flows depends strongly on the plenum pressures. These pressures depend in turn on what is happening in the loops. For the worst double ended cold leg break, the shortest path out the upper plenum, where the ECC enters, to the containment where it goes is down through the core so I would expect that is the path chosen by most of the flow.

The calculation outlined is probably conservative for the following reason. There are a few bundles at the outside of the core that are much colder than the rest. The semi-scale experiments show that these bundles, with half or less of the maximum bundle heat fluxes on them never get very hot. They stay wet all through blowdown and increase in temperature only very slowly after dryout. One would expect the vapor velocity out the top of these bundles to be quite low under any circumstances so that liquid would have little difficulty draining back through them. If even a very small area was draining down it would provide rapid drainage of any pool in the upper plenum. Recall how quickly a bathtub drains with a drain hole that is probably less than 0.001 of the tub bottom area.

With the recommended 50-50 (on area) radial nodalization scheme the few really cold bundles will be averaged with a large number of much hotter ones. As a result the calculated rapid drainage down will occur much later than the actual drainage. Of course the more detail one is able to put into the core description the more faith one would have in the answer. We are only describing a minimum calculation.

Summary of Events Leading to Water Appearing in the Lower Plenum

1) Water enters upper plenum through core deluge tubes and condensation occurs on them and on the bare jet itself.

2) Water hits the top of the core and either pools up, dribbles through or pours through. It pools up if the vapor velocity up through the core is greater than the maximum flooding velocity. It dribbles through if the vapor velocity up is between zero and the maximum flooding velocity. It pours through if the flow is down. The minimum area for downflow would be given by a piece of the core which had a flow area equal to the jet area with one of these areas for each jet. The maximum area for downflow would be the whole core. The maximum at this time seems most likely.

3) Once the velocity through the core is down, liquid will drain through the core at essentially its free fall velocity. Heat transfer will occur in a mode governed primarily by the local fuel rod temperature. Quenching will ultimately occur both by bottom flooding and top flooding quite rapidly for these flooding rates corresponding to the core deluge system flow rate. As we can now only calculate bottom flooding the progress of the quench will be more rapid than calculated following these recommendations.

Chapter IV

Reflood Heat Transfer

The reflood stage begins when the ECC injected from the upper plenum and intact cold leg reaches the bottom of the reactor core. As the water level rises from the bottom of the core, the energy stored in the fuel is removed and the cladding is quenched to the local coolant temperature.

Because reflood heat transfer plays an important role in the LOCA, there have been extensive investigations of reflood heat transfer both experimentally and theoretically. Experimentally, the NRC-sponsored PWR-FLECHT (Full Length Emergency Core Heat Transfer) represents the most extensive study of the bottom reflooding for PWR's [15]. In addition, Kirchner [12] developed a computer code, REFLUX, to predict the temperature-time histories of rod bundles undergoing a flooding process, based on physical description of the process involved. The temperature-time histories predicted by REFLUX agree fairly well with the FLECHT experiments for the high pressure (60 psia) and high flooding rate (2-6 in) cases. For a more detailed description of REFLUX and the predictions of temperature-time histopries for selected FLECHT reflood simulation experiments, the reader is referred to Reference [12].

The core of the reactor under study is two feet shorter than the rod bundle used in the W FLECHT program. The twelve foot bundle used in FLECHT is the maximum for any core now in use. In view of this fact, two REFLUX calculations were run, one with an axial length of ten feet and the other twelve feet, to determine the effect on the reflood heat transfer of a reduction of the core length to a somewhat smaller value.

The input conditions for the two runs are presented in Table 2. This set of input conditions is chosen be cause the temperature-time history predicted by REFLUX agrees fairly well with the FLECHT experimental result simulated under the identical set of input conditions.

The results of the temperature-time histories corresponding to the midplane elevation for both ten feet and twelve feet axial lengths are plotted in Fig. 11. A comparison of peak clad temperatures and quenching times is listed below:

Table 3

Axial length	Peak Temperature (^oF)	@ time after break (sec)	quenching time (sec)
10 ft. rod bundle	1792	20	108
12 ft. rod bundle	1821	25	128

From Fig. 11 it is interesting to note that both temperature curves fall in a parallel fashion after reaching the peak clad temperature; consequently, it is conceivable that a simple "cut off" of the W FLECHT core will be sufficient to predict the reflood heat transfer.

The results shown in Table 3 are not surprising. For a given similar temperature distribution one would expect a 20% shorter core to quench in about 20% less time for the same flooding rates.

Table 2

REFLUX Input Conditions

Initial clad temperature	1604°F
Flooding rate	3.9 in/sec
Peak power	1.24 Kw/ft
Decay curve	ANS Decay + 20%
Pressure	58 psia
Cladding material	stainless steel
Axial temperature initilization	truncated sine curve
Axial nodes	100
Radial nodes	8
Time step size	1.0 sec

Chapter V

Conclusions

The sequence of events occuring from when ECC is injected into the upper plenum and makes its way to the lower plenum has been examined. The phenomena occurring have been delineated and recommendations made for calculating both conservative and best estimate values. The calculations have been illustrated in a series of examples given in the appendices A through E. To do an integrated calculation a loop code with some capability of modeling the two dimensional effects in the core is essnetial.

One cannot look at the parameters that characterize this sysstem without being impressed with the immense flow rates that the ECCS is capable of delivering. We find it very difficult to imagine that the system will not promptly quench the core.

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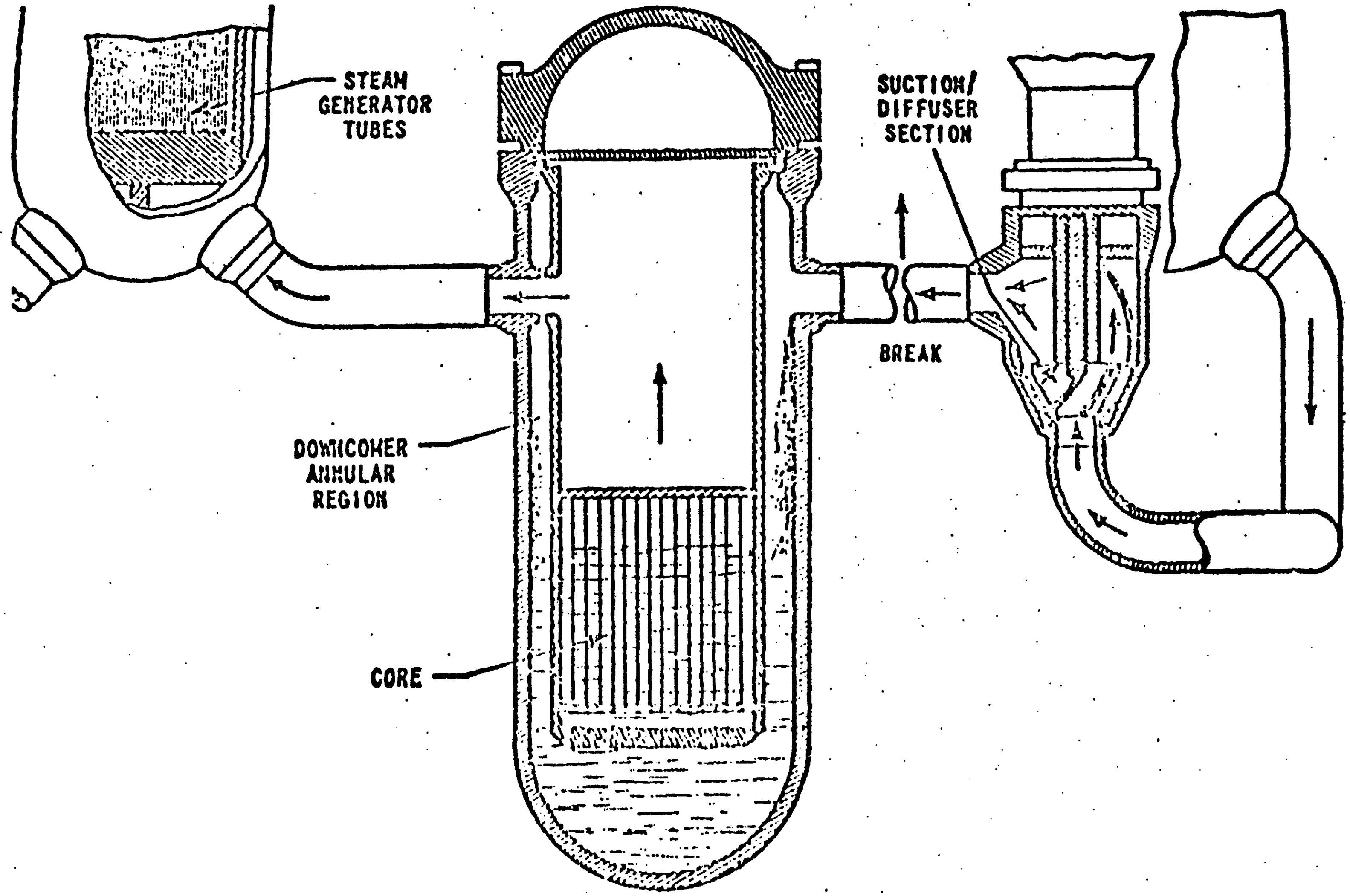
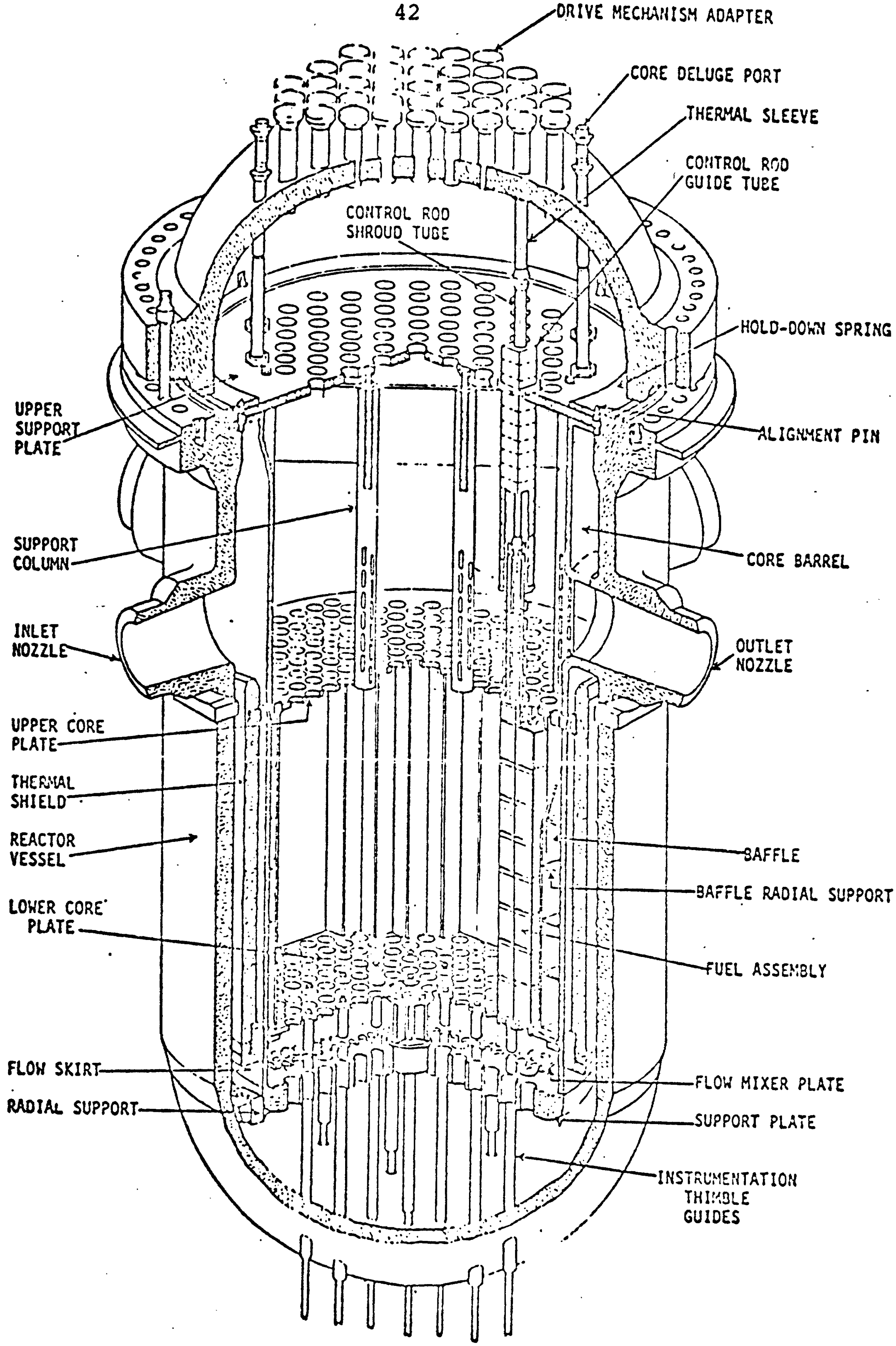


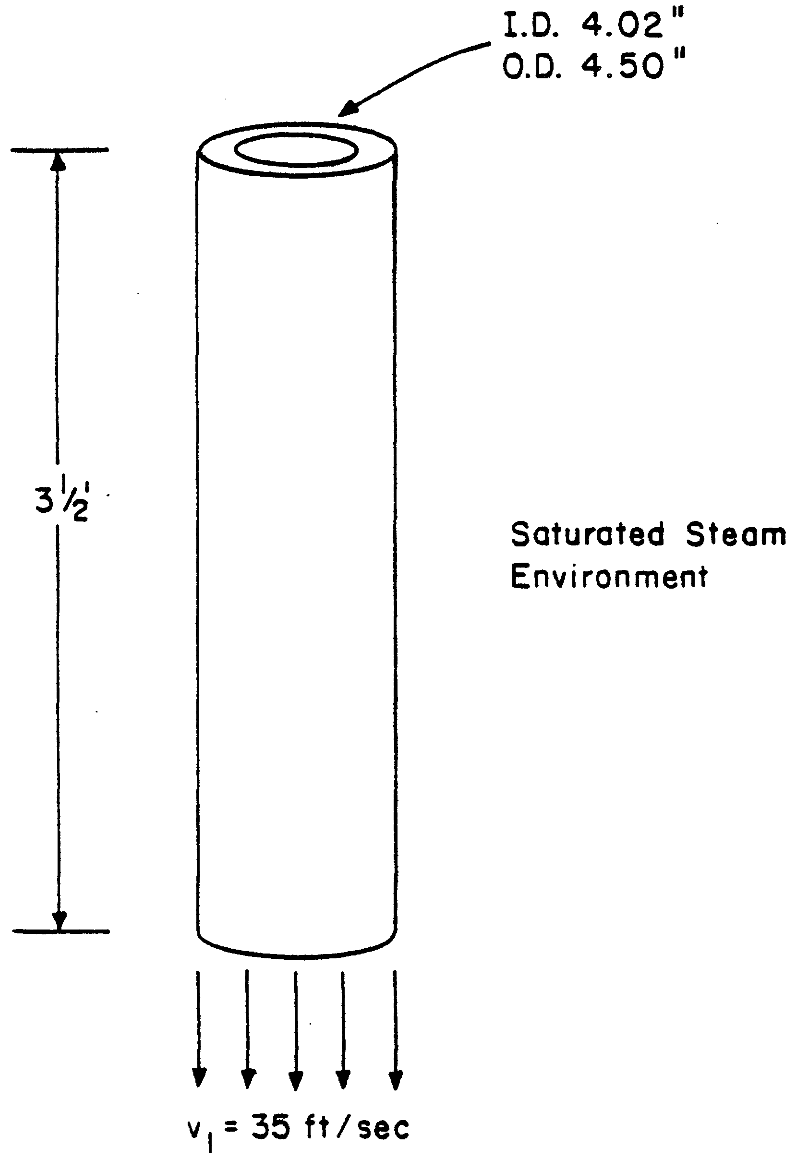
FIGURE 1 - Schematic of PWR with Double Ended, Cold Leg Break [12]



REACTOR VESSEL AND INTERNALS

FIGURE 2.

SCHEMATIC OF A PWR VESSEL AND THE RESPECTIVE LOCATION OF THE CORE DELUGE TUBES



Sketch of a Core Deluge Tube in the Upper Plenum

Figure 3

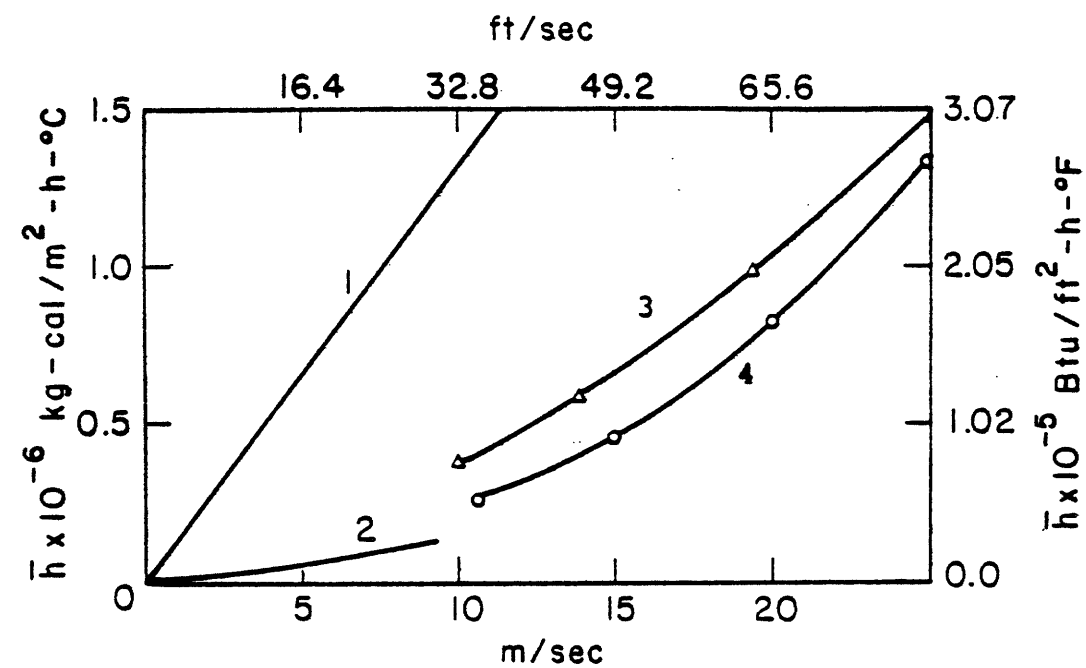
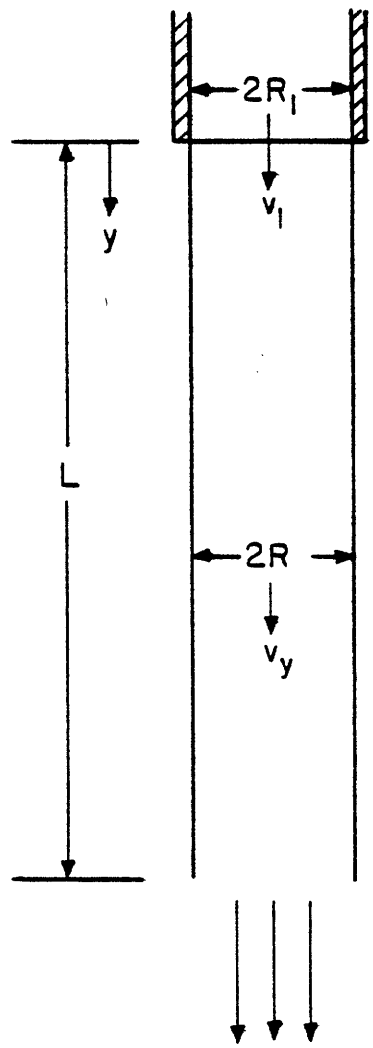


Figure 4 Variation of the Coefficient of Heat Transfer with the Velocity of Flow (L= 800 mm) [2]

1. – G.N. Abramovich [3]

2. – S. S. Kutateladze [1]

3.4.– N.M. Zinger [2]



Sketch of a Free-Falling Jet

Figure 5

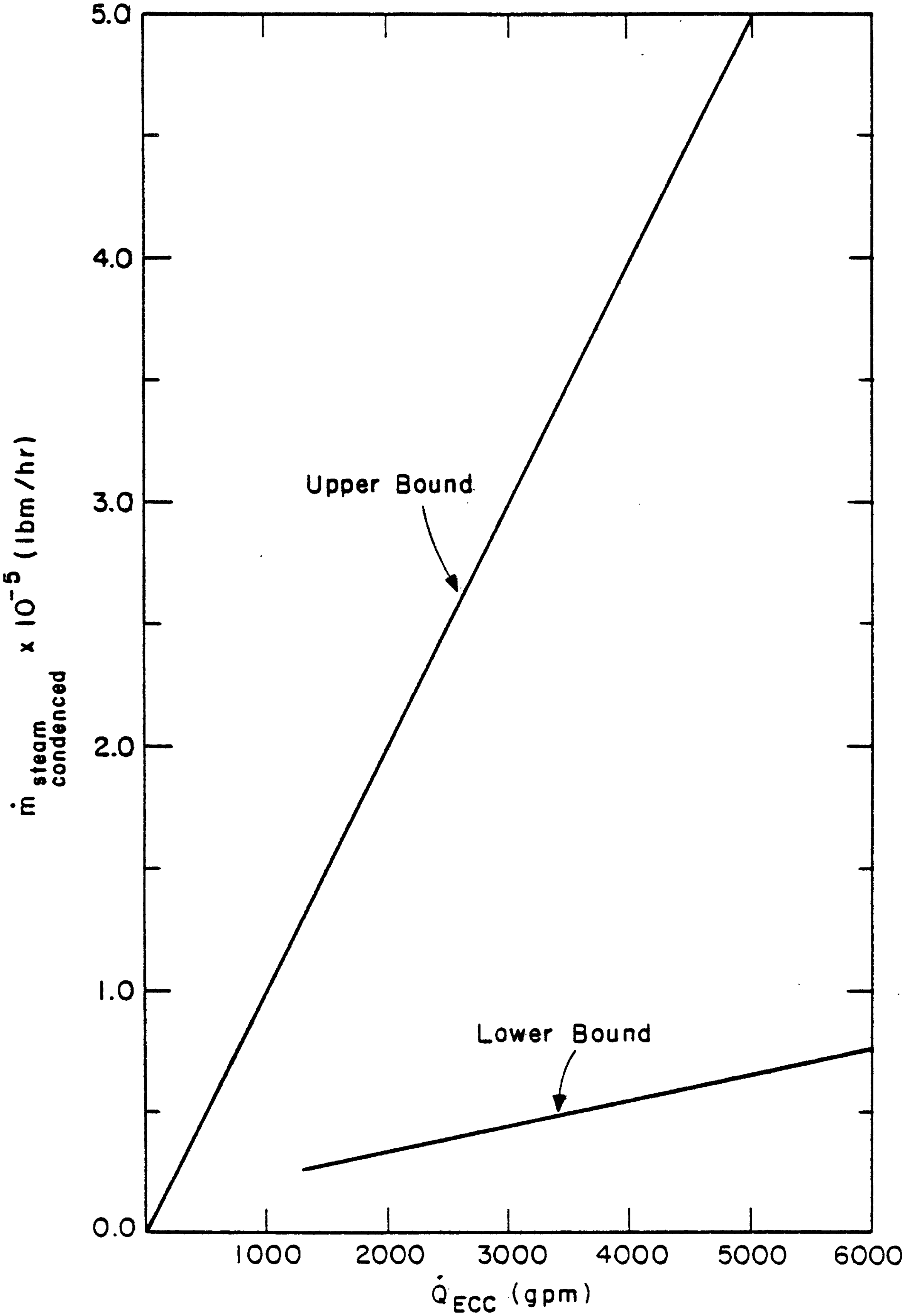


Figure 6 Upper and Lower Bounds on Steam Condensation Rate vs. ECC Injection Rate

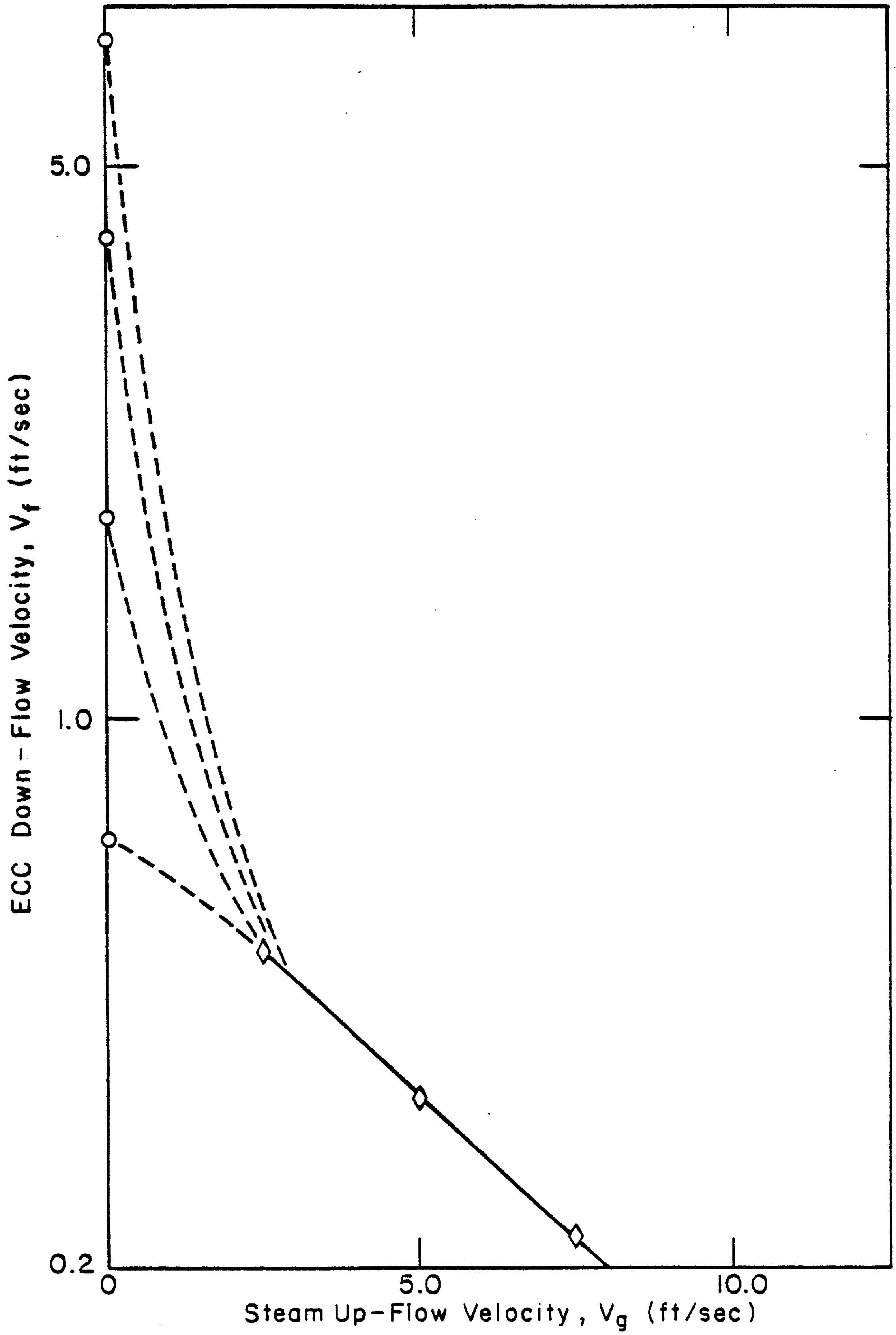
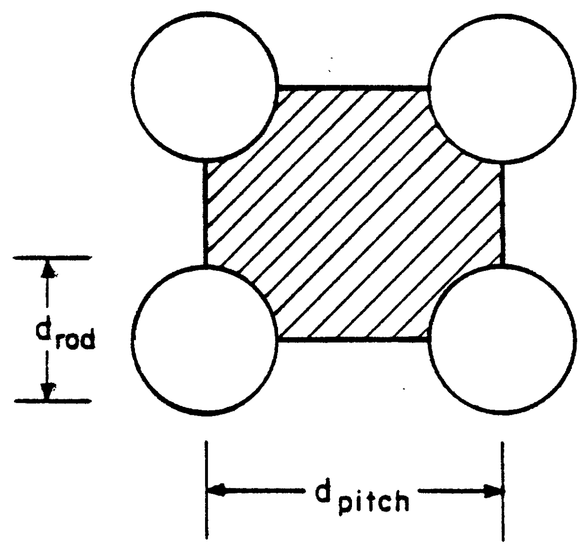


Figure 7 ECC Down-Flow Velocity vs. Steam Up-Flow Velocity



Sketch of a Unit Cell

Figure 8

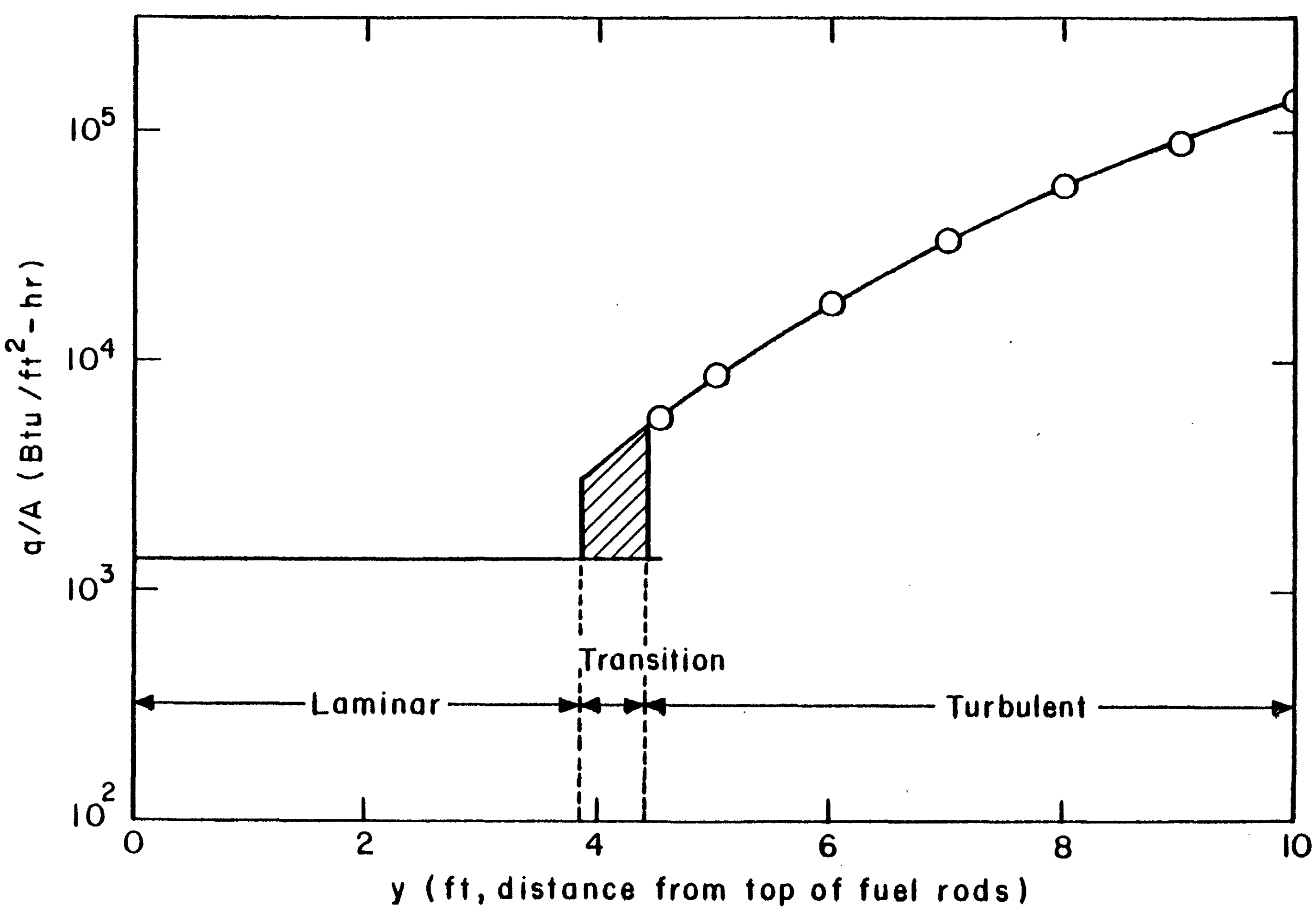


Figure 9 Down-Flow Heat Transfer vs. Axial Length

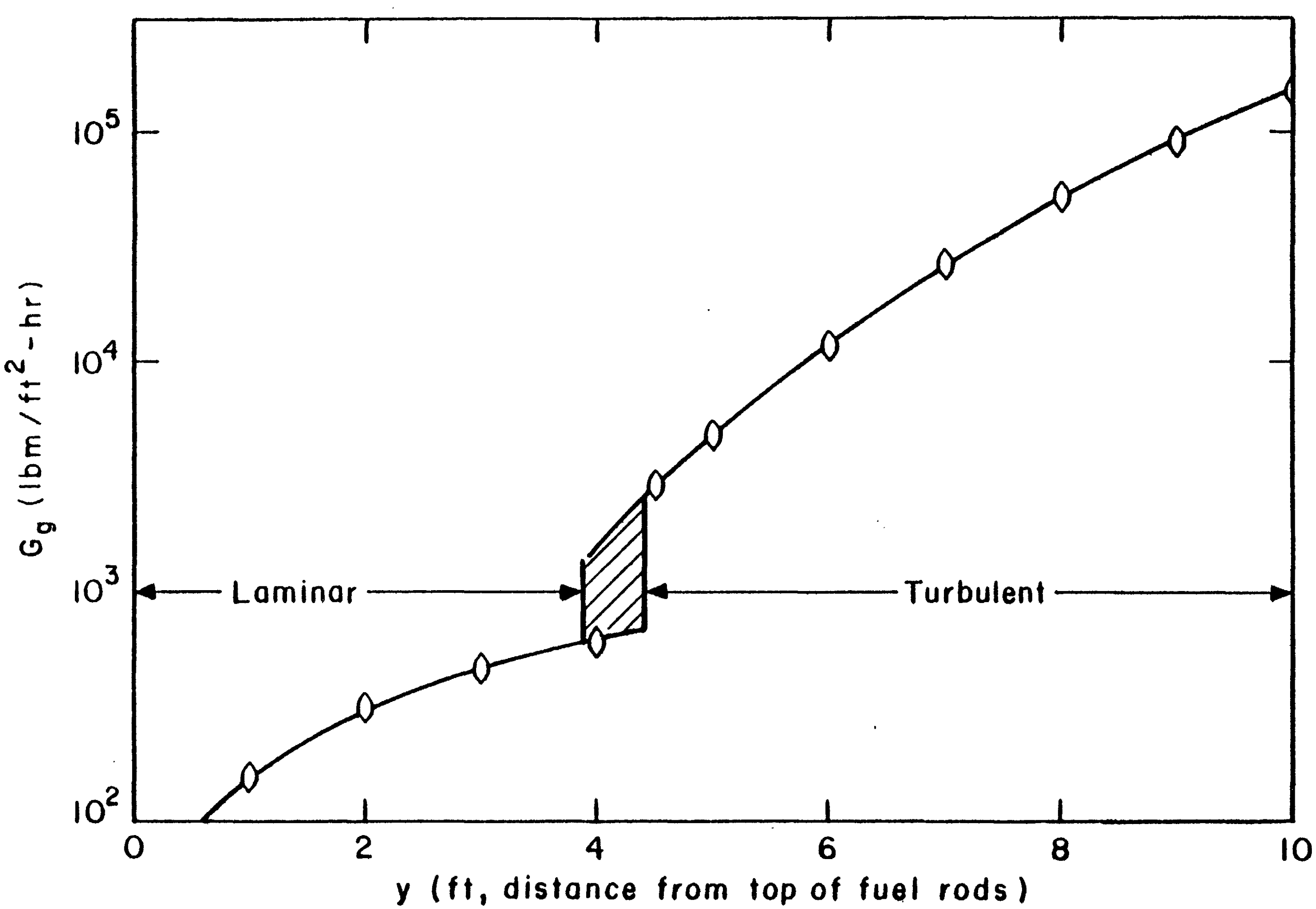


Figure 10 Vapor Mass Velocity, G_v vs. Axial Length

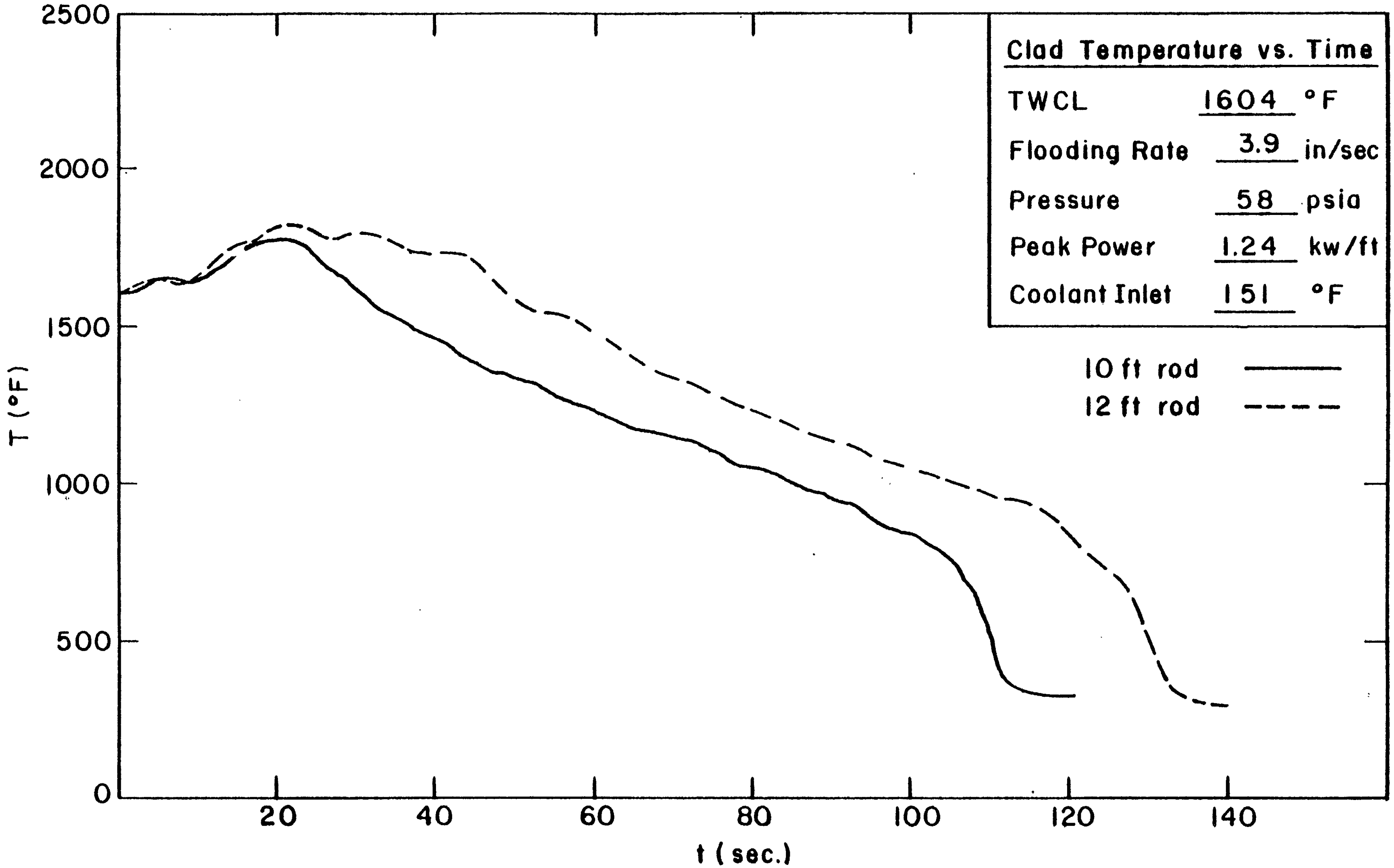


Figure 11 Temperature - Time History (10' vs. 12' Core Length)

Appendix A

Steam Condensation Rate on the Upper Portion of a Core Deluge Tube

(See Fig. 3)

Given:

Vertical Condensable Length, 3.5 ft.

Temperature of Saturate Steam @ 45 psia, $T_{sat} = 274.46^{o}F$

$T_b = 100^{o}F$

$r_o = d_o/2 = 0.188$ ft.

$r_i = d_i/2 = 0.168$ ft.

$k_b = 0.362$ Btu/hr-ft $-^{o}R$

$\mu_b = 0.677$ lbm/hr-ft $-^{o}R$

$Cp_b = 0.998$ Btu/lbm$-^{o}F$

$G_f = 7.76 \times 10^6$ lbm/ft^2-hr

h_{fg} (45 psia) = 928.8 Btu/lbm

$k_{ref} = 0.394$ Btu/hr-ft$-^{o}R$

$\mu_{ref} = 0.309$ lbm/hr-ft

$Cp_{ref} = 1.01$ Btu/lbm^{o}F

ρ_f@ 45 psia = 58.11 lbm/ft^3

ρ_g@ 45 psia = 0.106 lbm/ft^3

Assume $T_{sat} - T_w = 50^{o}F$

From equation (3), the heat transfer coefficient for forced-convection, turbulent flow in tubes

$$h_i = (0.023) \cdot \left(\frac{G_f d_i}{\mu_b}\right)^{0.8} \cdot \left(\frac{\mu C_p}{K}\right)_b^{0.4} \cdot \left(\frac{K_b}{d_i}\right)$$

$$= (0.023) \cdot \left[\frac{(7.76 \times 10^6 \text{ lbm/ft}^2\text{-hr})(0.336 \text{ ft})}{(0.677 \text{ lbm/hr-ft})}\right]^{0.8} \cdot$$

$$\cdot \left[\frac{(0.667 \text{ lbm/hr-ft})(0.998 \text{ Btu/lbm-}^{o}\text{F})}{(0.362 \text{ Btu/hr-ft-}^{o}\text{R})}\right]^{0.4} \cdot$$

$$\cdot \left[\frac{(0.362 \text{ Btu/hr-ft-}^{o}\text{R})}{(0.336 \text{ ft})}\right]$$

$$h_i = 5902 \text{ Btu/hr-ft}^2\text{-}^{o}\text{R}$$

From equation (5), the fluid properties are evaluated at the reference temperature.

$$T_{ref} = T_w + 0.31\ (T_{sat} - T_w)$$

$= (224.46^\circ F) + 0.31(50^\circ F)$

$= 240^\circ F$

From equation (4), the heat transfer coefficient associated with film condensation on a vertical surface

$$h_o = 0.943 \left[\frac{g\rho_f(\rho_f - \rho_g)K^3(h_{fg} + 0.68\, C_p \Delta T)}{L\,\mu(T_{sat} - T_w)} \right]^{1/4}$$

$$= 0.943 \left\{ \frac{(4.173 \times 10^8 \text{ ft/hr})(58.0 \text{ lbm/ft}^3)}{(3.5 \text{ ft})\ (0.31 \text{ lbm/ft-hr})} \cdot \frac{(0.394 \text{ Btu/hr-ft-}^\circ R)^3(963 \text{ Btu/lbm})}{(50^\circ R)} \right\}^{\frac{1}{4}}$$

$= 1049 \text{ Btu/hr-ft}^2\text{-}^\circ R$

the overall heat transfer coefficient, U_i, is calculated from equation (2)

$$U_i = \left[\frac{1}{h_i} + \frac{A_i \ln\left(\frac{r_o}{r_i}\right)}{2\pi K_s L} + \frac{A_i}{A_o h_o} \right]^{-1}$$

$$= \left[(5902 \text{ Btu/hr-ft}^2)^{-1} + \right.$$

$$\left. + \frac{(0.168 \text{ ft}) \ln\left(\frac{0.188}{0.168}\right)}{(9.82 \text{ Btu/ft-hr-}^\circ\text{R})} + \frac{(0.168 \text{ ft})}{(0.188 \text{ ft})(1049 \text{ Btu/hr-ft}^2\text{-}^\circ\text{R})} \right]^{-1}$$

$$= 340 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

Check if $T_s - T_w \simeq 50^\circ F$ is valid.

$$T_{sat} - T_w = \frac{U_i A_i (T_{sat} - T_b)}{h_o A_o}$$

$$= \frac{(340 \text{ Btu-ft}^2\text{-}^\circ\text{R})(0.168 \text{ ft})(175^\circ\text{R})}{(1049 \text{ Btu/hr-ft}^2\text{-}^\circ\text{R})(0.188 \text{ ft})} = 51^\circ\text{F}$$

The steam condensation rate on each core deluge tube is calculated from equation (7)

$$\dot{m}_{tube} = \frac{q}{h_{fg}} = \frac{U_i (2\pi r_i L)(T_{sat} - T_b)}{h_{fg}}$$

$$= \frac{(340 \text{ Btu/hr-ft}^2\text{-}^\circ\text{R})(2\pi)(0.168 \text{ ft})(3.5 \text{ ft})(175^\circ\text{F})}{(929 \text{ Btu/lnm})}$$

$$\dot{m}_{tube} = 237 \text{ lbm/hr.}$$

Appendix B

Direct Contact Condensation of Steam on a Water Jet (see Figure 5)

The average heat transfer coefficient along the jet length by means of direct contact condensation is calculated using equation (17).

$$\bar{h} = (4.82 \times 10^{-4})\left(\frac{C_p \rho_f}{g_L}\right)\left\{\left[\left(\frac{V_1}{\phi}\right)^2 + 2gL\right]^{3/2} - \left(\frac{V_1}{\phi}\right)^3\right\}$$

where:

1) $C_p(140^oF) = 9.98 \times 10^{-1}$ Btu/lbm-^oR
2) $\rho_f(140^oF) = 61.4$ lbm/ft^3
3) $g = 32.2$ ft/sec^2
4) L (length of the liquid jet) = 6 feet
5) $V_1 = 35.0$ ft/sec
6) $\phi = 0.9$
7) $h_{fg}(60\ \text{psia}) = 916$ Btu/lbm
8) $A = 2\pi R_1 L = 6.33$ ft^2

$$\bar{h} = (4.82 \times 10^{-4})\left[\frac{(0.998\ \text{Btu/lbm-}^oR)(61.4\ \text{lbm/ft}^3)}{(32.2\ \text{ft/sec}^2)\ .\ (6\ \text{ft})}\right].$$

$$\left\{\left[\left(\frac{35.0\ \text{ft/sec}}{0.9}\right)^2 + 2(32.2\ \text{ft/sec}^2)(6\ \text{ft})\right]^{3/2} - \left(\frac{35.0\text{ft/sec}}{0.9}\right)^3\right\}$$

$= 3.66$ Btu/ft^2-sec-^{o}R

or

$h = 13166$ Btu/ft^2-hr-^{o}R

The steam condensation rate by direct contact method, $\dot{m}_{direct}$, is then calculated from equation (18).

$$\dot{m}_{direct} = \frac{\bar{h}A(T_{sat} - T_b)}{h_{fg}}$$

$$= \frac{(13166 \text{ Btu/ft}^2\text{-hr-}^o\text{R})(6.33 \text{ ft}^2)(150^o\text{R})}{(916 \text{ Btu/lbm})}$$

$$\dot{m}_{direct} = 1.37 \times 10^4 \text{ lbm/hr.}$$

For four identical water jets, the local $\dot{m}_{direct\ total}$ is

$$\dot{m}_{direct\ total} = 4\ \dot{m}_{direct} = 5.48 \times 10^4 \text{ lbm/hr.}$$

Appendix C

Upper and Lower Bounds of m_{direct} Condensation as a Function of G_v

Based on the following assumptions:

1) Temperature of saturated steam at 60 psia, $T_s = 290^oF$
2) Vertical condensable lengh. L = 6 feet
3) Inner radius of the core deluge tube, r_i = 2.013 inches
4) Average density of ECC, $\bar{\rho}_{ecc} = 59.8$ lbm/ft^2
5) Temperature of ECC at inlet, $T_{ECC\ inlet} = 100^oF$
6) h_{fg} (p = 60 psia) = 917.5 Btu/lbm
7) $Cp_{ECC} = 0.977$ Btu/lbm^{o}F

The upper bound, using equation (19), can be simplified

$$\dot{m}_{upper} = 99.64\ \dot{Q}_{ECC}$$

where $\dot{Q}_{ECC}$ is in (gpm)

The lower bound, using equations (17) and (18), can be reduced to

$$\dot{m}_{lower}[\text{lbm/hr}] = (2.912)\{[5.49 \times 10^{-5}\ Q^2_{ECC} + 386.4]^{1.5} - 4.065 \times 10^{-7}\ Q^3_{ECC}\ \}$$

where $\dot{Q}_{ECC}$ is in (gpm)

The results of $\dot{m}_{upper}$ and $\dot{m}_{lower}$ are plotted against $\dot{Q}_{ECC}$ and are shown in Figure 6.

Appendix D

ECC Down-Flow Velocity vs. Steam Up-Flow Velocity Through the Core

For high steam up-flow velocity through the core, the penetration of the ECC into the core is calculated based on Wallis' correlation for flooding in vertical tubes [14].

$$j_g^{*1/2} + j_f^{*1/2} = 1 \qquad \text{(D1)}$$

Equation (D1) can be rewritten as follows:

$$V_g^{1/2} \left[\frac{\rho_g}{\rho_f g D_h} \right]^{\frac{1}{4}} + V_f^{\frac{1}{2}} \left[\frac{1}{g D_h} \right]^{\frac{1}{4}} = 1 \qquad \text{(D2)}$$

Using the following physical assumptions

1) Hydraulic diameter of the fuel rods, $D_h = 3.92 \times 10^{-2}$ft
2) Density of steam at 60 psia = 0.139 lbm/ft^3
3) Density of liquid at 100^{o}F = 62.0 lbm/ft^3

Equation (D1) can be further reduced to

$$0.205\ V_g^{\frac{1}{2}} + 0.943\ V_f^{\frac{1}{2}} = 1 \qquad \text{(D3)}$$

when the pressure differential between the upper plenum and lower plenum reduces to a small value, the liquid downflow rate is found to be independent of vessel pressure or gas flow rate. The maximum attainable steady-state liquid superficial velocity V_f is a function of H/D alone, where H is the liquid height and D is the diameter of the drain [19]. For $H/D_h \leq 0.4$ liquid flow is self-venting and the liquid superficial velocity, V_f is

$$V_f \leq 2.36 \sqrt{gD_h}\,(H/D_h)^{1.5} \tag{D4}$$

after reduction,

$$V_f \leq 2.66\ (H/D_h)^{1.5} \tag{D5}$$

For $0.4 \leq H/D_h \leq 3$, McDuffie suggested the following relationship:

$$V_f \leq 1.6 \quad gD_h \quad (H/D)^2$$

or

$$V_f \leq 1.80(H/D_h)^2$$

The above analyses are combined and plotted in Figure 7.

Appendix E

Downflow Post CHF Heat Transfer Example Calculations

In this example we want to calculate the heat flux and vapor mass velocity as functions of axial length, which is measured downwind from the top of the reactor core.

The following physical dimensions and thermal properties are assumed:

1) $d_{pitch} = 0.506$ in
2) $d_{rod} = 0.382$ in
3) $\alpha = 1$
4) $T_w = 1200^oF$
5) T_{sat} @ 60 psia = 293^oF
6) $K_v(293^oF) = 0.0163$ Btu/hr-ft-^oF
7) $h_{fg}(293^oF) = 915$ Btu/lbm
8) $u_v\ (293^oF) = 0.0336$ Btu/lbm-ft-hr
9) $(pr)_v = 0.975$
10) $(c_p)_v(293^oF) = 0.445$ Btu/lbm-^oF

The data can be reduced into the following two regions:

a. Laminar Flow Region ($Re_v < 2000$)

The vapor Reynolds number is evaluated using equations (28) and (31)

$Re_v = 179.5y$

The heat flux can be calculated by combining equations (27) and (28)

$(q/A) = h\ (T_w - T_{sat}) = 1380$ Btu/hr-ft^2

The vapor mass velocity is obtained from equation (32).

$$G_v = XG = \frac{4(q/A)}{h_{fg}} \left(\frac{y}{D_h}\right) = 153.9y \; [\,lbm/ft^2\text{-}hr\,]$$

b. Turbulent Flow Region ($Re_v > 3000$)

From equation (31), the Reynolds vapor number is reduced to

$$(Re)_v = (5.31 \times 10^{-3})(q/A)(y/D_h)$$

the heat flux as a function of axial length is obtained from equation (37).

$$(q/A)^{0.2} = 0.127(y/D_h)^{0.8}$$

The vapor mass velocity for the turbulent flow region using equation (38) is:

$$G_v = (0.112)(q/A)(y)$$

In Figures (9) and (10), the heat flux and vapor mass velocity are plotted against the axial length.