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DISCUSSION OF NEED FOR HOT WALLS
IN CORE BYPASS ECC EXPERIMENTS

Battelle

Pacific Northwest Laboratories
Richland, Washington 99352

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DISCUSSION OF NEED FOR
HOT WALLS IN CORE BYPASS
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R. T. ALLEMANN

July 1973

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INTRODUCTION

An experiment is being planned to study the delivery of Emergency Core Cooling (ECC) water to the lower plenum of a pressurized water reactor during the course of a cold leg break loss-of-coolant accident. The experiment is being designed to simulate in as large a physical scale as possible, the countercurrent flow of the injected coolant and the reverse core steam flow in the annulus. Because of limitations in available steam supplies the tests cannot be run at full design conditions of a prototype reactor. The scaling parameter which is being used is the momentum flux of the countercurrent flows in the annulus. The question of whether the ECC water will reach the lower plenum is similar to that in countercurrent flooding in a wetted wall or packed column-for which momentum scaling will correlate the data. The experiment will duplicate the momentum flux of a prototype for the major portion of the injection flow by operating at a lower pressure than the prototype.

It has been suggested that perhaps the walls of such an experiment should be heated to simulate the hot walls of a reactor system at the time of blowdown and ECC injection. Such heating is not planned for most of the experiments and the basis for this is explained in the following discussion.

CONCLUSIONS

The heat transfer from the hot walls will not require that the walls of a simulation experiment be thick to adequately simulate it. The actual effect of the heat transfer is expected to be small, either because the wall is very hot giving film boiling and hence a relatively low heat flux, or because the blowdown cooling has already established a temperature gradient in the wall which also results in negligible heat transfer effects. For

those places on the annulus wall that occur with an unusual condition having both high heat content and high heat transfer, the cool down would occur very rapidly and the main effect would be on the pressure drop required to flow past these places. In the annulus of a vessel, at which the Battelle experiment is mainly directed, such pressure drops would probably have a salutary effect in promoting water flow to the lower plenum with counterflow of steam to the broken pipe leg, because the breakthrough of water at one place would be maintained.

Although the effects of the heated wall are probably small in a large scale test, it cannot be concluded that there is no possibility of the small effects causing changes or triggering changes in flow pattern which then allow accentuated primary effects to take place. Furthermore, it cannot be stated unequivocally at this time that the lack of heated walls which might otherwise create the above effects in an experimental test would result in conservative or unconservative data from the test. To answer this question requires some data concerning the flow pattern that would occur in the test and the prototype. It is, therefore, worthwhile that some tests be run without heated walls in order to obtain some of this information. If the tests show that down flow of water occurs as a wall film - the only regime on which heat transfer from walls might have much influence - rather than bubbly or slug flow then further tests can be run (probably quasi-steady state runs with preheated vessel walls) to see if heated walls are important to reflood at large scale. It is not out of the question that a separate, more detailed study should be done of the flow patterns of emergency coolant water entering an annulus. This experiment would have hot walls and injection water flowing countercurrent to steam.

DISCUSSION

The tests which are proposed are to study the action of the injected coolant in the annulus during periods of counterflow of steam. The dominant mechanisms of the interactions of the two flows is expected to be due to momentum exchange and direct condensation of the steam on cool injection water. The system pressure in the experiment is to be run lower than expected for the prototype in order that the momentum flux of the steam upflow will be, as far as possible, equal to that in the prototype.

Will the results be invalid if wall heat flux is not included? The question of whether to include wall heat transfer as a necessary part of the modeling in such an experiment is difficult to answer. It depends on the relative size of the different mechanisms and upon the actual flow conditions that are experienced. Some estimates can be made of the magnitude of the effects, and of the probable flow regimes. Having made some of these estimates for a prototype, one then considers whether the effects would be the same for the momentum scaled experiment. If they are not the same, then the inclusion of the wall heat transfer may cloud the results and require corrections, thus giving little advantage over applying corrections to the unheated case. It would be well, however, to test the effects experimentally to some extent to exemplify the validity of the wall heat transfer assessment. This is the plan for the small scale quasi-steady tests.

Little note is generally taken of the difficulty of providing the large heat fluxes suggested in a large scale steady state experiment.

Estimates of the power required for a heated wall steady state experiment, range from 4 to 13 MW. This amount of power is essentially precluded by the cost and complexity involved.

A summary of many of the ideas and calculations made for the preliminary assessment of the wall heat transfer effects is presented in Table I. The table shows examples of wall heat flux conditions and the projected effects on the ECC water in a prototype and in a momentum-scaled low pressure experiment. For some items, the table refers to page numbers in the discussion below.

The following is a review of some data and calculations made concerning the possibility of the heat contained in the vessel walls to influence the ECC injection hydraulics in a prototype or experimental simulation of the annulus of a PWR.

Recently, R. W. Shumway sent a letter (Attachment 1) which discussed the belief that the "fluxes could be high and the walls were only marginally Conduction Limited". In the letter he also showed the results of some scoping calculations for a prototype reactor based on a heat transfer coefficient of $1000 \text{ Btu/hr-ft}^2 \text{ }^\circ\text{F}$ (Figure 1, Attachment 1). Finally, he showed some data (Figure 1 and 2, Attachment 1) that were taken during a reflood test in the LOFT semiscale vessel.

A check was made using a heat transfer coefficient of $1000 \text{ Btu/hr-ft}^2 \text{ }^\circ\text{F}$ (Appendix C) of the wall temperatures of the LOFT reflood test by using the exact solution for a semi-infinite slab as given in Ref. 1. The results for both a stainless steel and a carbon steel wall are shown along with the experimental data in Figure 1.

(1) F. Kreith, "Principles of Heat Transfer", 2nd Ed., Int. Textbook Co., Scranton, Pa. (1965), p. 160.

TABLE 1. WALL HEAT TRANSFER IN ECC ANNULUS FLOW

POSTULATED WALL TEMPERATURE CONDITION					
FILM COUNTERFLOW DISPROPORTION COUNTERFLOW		HOT WALL, UNQUENCHED SURFACE	HOT WALL, QUENCHED SURFACE	HOT WALL, GRADIENT ESTABLISHED	COLD OR NEUTRAL WALL

1. Experimental evidence of the condition and comment	LOFT flooding test preheated vessel. (See Fig. 1) Discus. p 7) Probable condition if wall clear of liquid for > 10 sec..	Maximum case would need forced quenching. Can be considered as a transition condition from + to -	CSE blowdown tests. See Fig 3-2. Discussion p. Probable condition if ECC comes <10 sec after liquid recedes	Early in blowdown CSE	
2. Typical heat flux expected from prototypic conditions during ECC injection	10^4 Btu/hr ft ² (Discussion p. 7)	3×10^7 Btu/hr ft ² (Impossibly high) Discussion p. 9	10^4 Btu/hr ft ² Discussion p 8	$< 10^3$ Btu/hr ft ²	
3. Time duration of condition if present in prototype	30 seconds	0.25 seconds	0 - 200 seconds	0 - 5 seconds (large break)	
4. Heat flux effect on quality of ECC injected (in annulus prototype)	Heat fluid toward saturation or increase quality by .003 (Discussion p. 7)	Heats fluid toward sat. or increases quality (Discussion p. 7)	Heats fluid toward saturation or increases quality by .005 (Discussion p. 7)	Negligible	
5. Projected effect of heat flux on flow pattern of coolant in prototype	No major change in flow pattern or direction	Possible sputtering off and pressure drop increases	No major change in flow pattern or direction	Momentum exchange and heat exchange effects on flow pattern not influenced by wall heat flux factor.	
6. Projected mechanism of effect in item 5.	Film of steam found close to wall gives little wall friction allows fluid to move easily in response to other forces.	Rapid formation of steam causes disruption of flow next to wall and/or change in volume. This drives steam away or accelerates flow giving low ΔP with steam higher. ΔP with homogeneous flow and small diameter duct.	Steam formed by nucleate boiling. Fluid flows as two phase mixture.	Little or no steam formed or condensed. Relatively high wall friction because of liquid film.	
7. Effect of wall condition in annulus on momentum exchange effects between ECC injected and reverse core steam flow (upflow in annulus) steam	Less wall drag, fluid carried more easily upward by steam in counterflow or allowed to descend more easily if disproportionate flow conditions are established.	Pressure zone may divert steam flow somewhat but efficient momentum exchange will cause liquid to be carried up more. Less flow area might be available for disproportionate flow.	Two phase wall friction slightly higher than single phase but break up of surface would promote higher momentum exchange with rising steam.	Upward momentum of steam is sufficient to give rising film flow if phases uniformly distributed around the annulus but disproportionate counterflow has maximum opportunity of occurrence.	
8. Effect on cold water injection	Some heating by wall of the water would reduce condensation and momentum transfer (at point of contact) with rising steam.	Larger amount of heating of water than	Same as	Momentum exchange is high and concentrated due to rapid steam condensation until water temperature is saturated	
9. Effects in momentum scaled (low pressure) tests (ECCS two-phase mixing tests)	Large volume change with vaporization, thicker vapor film. wider temperature range for this phenomenon to occur. Increased effects due to breakup such as Item 7. p (19)	Less likely to occur. Heat transfer coefficient is less. Exaggerated effects such as 7.	More violent boiling, larger void fraction created, than prototype. Heat transfer coefficient is less. Increased effects such as item 7.	Little effect. Same as 6 & 7.	

Note: Projected wall thicknesses of ECC tests are sufficient to give heat fluxes for a transient (blowdown) test but in steady state tests the heat flux from hot walls would not be prototypic without heating the walls continuously during the test. The ability to have preheated walls for the quasi-steady tests depends on the initial conditions and will probably require a dry vessel to start.

FIGURE 1. LOFT REFLOOD TEMPERATURES

It can be seen that the relatively slow decrease in the experimental wall temperature relative to the calculated results indicates that the experimental heat transfer coefficient is much less than 1000. Using the data at 10 seconds and assuming the semi-infinite slab analysis applies at this time it was found that the effective heat transfer coefficient for the experiment is only about $90 \text{ Btu/hr-ft}^2 \text{ } ^\circ\text{F}$ which would give a flux of $3.5 \times 10^4 \text{ Btu/hr ft}^2$. Obviously this relatively low flux is due to the film boiling effects. At this flux approximately $.008 \text{ lb/sec ft}^2$ of steam would be produced. If a typical injection rate of some 5000 lb/sec were uniformly distributed over all the annulus surface (including a thermal shield) it would give more than 2.3 lb/sec of injected water per square foot of surface area. Hence the water if already saturated, would be vaporized at the surfaces to an average quality of .003. The countercurrent reverse core flow steam is present at some 10-20 lbs/sec across each lineal foot of surface. The void fraction, though not negligible at low pressures, would be initially formed at pressures above 300 psi and would be on the order of 0.2. The void would be formed at the walls so as to give a rather low wall friction. The void fraction is probably high enough to result in the film boiling regime and low heat transfer. Hence, during this flow regime the fluid would be somewhat less influenced by the wall friction than otherwise, and would tend to flow more easily and extensively in accordance with its inertia and the other forces that exist. If the flow is not uniform, then the quality increase due to vaporization at the wall would be even less at those places where water flow is larger than average. Hence, since the quality change is relatively small on the average, and low relative to the countercurrent steam flow, one would not expect that a wall at a high temperature, heated beyond transition boiling regime would have much effect on ECC water flow during the conditions of ECC injection.

Figure 2 shows some ECCS predictions made by a vendor for a large PWR. Note that the accumulator injection begins before the end of blowdown and lasts about 30 seconds for the large break case. This shows the basic time frame at which important effects must operate. There is evidence

FIGURE 2. Double Ended Cold Leg Break

that the blowdown process itself cools the wall of the vessel within such a time frame. This process is accomplished with a relatively high wall heat transfer coefficient as evidenced by the very small temperature difference between the flashing blowdown water and the wall itself.

Figure 3, 5, and 6 show the temperature time histories of several thermocouples in the wall and at the inner surface of a 3-1/2" thick stainless steel clad vessel which were taken during the course of the Containment Systems Experiment. Figure 4 is a more detailed look at conditions of Run B19 B but using a separate set of sensors. Figure 7 is also a more detailed view of this type of data in which, for increased assurance, the surface thermocouple was made intrinsic with the metal wall rather than just bonded on as in the case of the other surface thermocouples. The close correspondence of the fluid temperature and the surface temperature can be seen. This suggests that during the relief of pressure a nucleate boiling regime exists. These figures show that during a time span of about 30 seconds a thermal gradient is built into the metal wall that leaves thermocouples at a depth of one inch or so essentially unchanged. Therefore, for experiments of this time span, thicker walls need not be used. If one calculates the heat flux which is producing the average cool down rate of the surface slab (see Figure 5) it is about $3-6 \times 10^4$ Btu/hr-ft². This rate is also small so that again little additional quality is produced but in this case the slowness is due to a thermal gradient in the metal. This gradient was established during the blowdown and the heat transferred then was of little import to the large quantities of water flowing.

FIGURE 3. Temperature Histories in 3 1/2" Vessel Wall

FIGURE 4. Temperature Histories in Metal Plug

FIGURE 5. Temperature Histories in 3 1/2" Vessel Wall

FIGURE 6. Temperature Histories in 3 1/2" Vessel Wall

FIGURE 7: Temperature Histories in 3 1/2" Vessel Wall

Therefore, we have seen that either a very hot wall or a previously cooled wall will produce little steam relative to the flow rates of the ECC coolant. One may ask, "Will there be an intermediate condition in which the wall surface has been quenched but the gradient has not yet been established such that extremely high heat fluxes will occur?" In the time frame that is being considered, it does not seem likely that such would be the case. When a gradient has been established in the wall during blowdown due to liquid water cooling there is a time delay of about 20-40 seconds for the surface to heat up again due to thermal radiation and conduction to the wall after the fluid level uncovers it. This is shown in Figure 3 by the surface thermocouple which warms up again in about 40 seconds. There is possibly a certain timing of events which could result in the situation described; in which the wall surface was at a temperature below transition and the wall interior near the surface was still at a high temperature with little gradient. Since the fluid level in the reactor vessel can be postulated to drop at a certain rate depending on the break size, it has been suggested that there may be a level in the vessel that has reheated just the right amount for this condition to occur. It should be emphasized that the above data and reasoning point to just a location (not necessarily a uniform level) on the vessel wall that may have these characteristics at the time of coolant injection. It would not be a general wall condition throughout the annulus of the vessel and the piping. Also, it should be remembered that liquid and vapor flow up the annulus from the plenum during a cold leg LOCA. To bound what might happen if ECC water were to advance to a wall region that

was quenched but still very hot, a calculation (Appendix A) was made using this assumption, i.e., a pool boiling heat transfer coefficient and a uniformly hot wall. The results in Figure 8 show that the hot wall would cool very rapidly and produce about $.5 \text{ lb/ft}^2$ of steam in .24 seconds. The wall heat flux that results under these assumptions is so high ($\sim 36 \times 10^6 \text{ Btu/hr ft}^2$, which is about twice the energy flux from the surface of the sun) that such conditions could not exist in reality. Therefore, a high temperature wall would have to be cooled by film boiling until it reached quenching. The flux would have dropped to fairly low values by this time also. Hence, it is shown that the case of the hot wall with a quenched surface is not probable, and that if it did occur the surface would cool very quickly to give heat transfer fluxes controlled by the thermal gradient in the metal. What normally will happen with a hot wall that does not have a cool surface is that cooling will be by the relatively slow film boiling process, requiring on the order of 30-40 seconds to quench.

The effect of the sudden production of steam (with quenching) on the pressure drop was calculated in a rough way (Appendix C & D) on the basis of separation of the steam in which the steam moves out ahead of the liquid, and for the case in which the steam does not separate but mixes with the water and is pushed out as a homogeneous mixture. The pressure drop is associated with the acceleration of the fluid as it gains specific volume in passing the vaporization zone. The pressure drop calculation assumes that the supply of the fluid is constant and is not changed by the possible change in pressure drop. This assumption precludes non-steady effects such as chugging and flow redistribution. The results of the calculations

FIGURE 8. Computed Temperature for Quenched Surface Hot Wall

are shown in Table D-II. Some rather large pressure drops in pipes are associated with homogeneous flow although by the nature of the calculation it is implied that the flow will get past the hot spot. In an annulus of six inches for a large reactor, injecting at 5000 lb the pressure drop would be 1 psi to pass the homogeneous steam water, downward. The velocity head of saturated injection water could perhaps be on the order of 300 psi in the annulus hence should easily overcome backpressure effects of its acceleration and, therefore, the water coolant would break through a hot spot barrier quite quickly and proceed to cool the wall as shown in Figure 8. The heat flux from the wall will fall off accordingly and the acceleration backpressure will drop off. This action results in an improvement in the downward flow in the region of first break through and suggests that even for the unlikely high heat flux condition the system might itself set up relatively stable counter flow paths (disproportionation) for the cooling water down the annulus and the steam flow from the core up the annulus to the break pipe.

Other Heat Transfer

There are other heat effects which can be larger than the wall heating. If the water is cold, steam formed at the metal wall will condense extremely rapidly until the water is heated. The water will also be heated by the condensation of countercurrent steam flow.

The condensation of steam in colder water occurs at rate equivalent to a heat flux of 5×10^7 Btu/hr ft². This has been estimated from steam jet condensation experiments reported by Binford, Stanford & Webster.*

*ORNL NS R&D Program Bimonthly Report ORNL-TM-2283 (July 30, 1968)
p. 75-90.

The velocity of approach of condensing steam under low pressure conditions is 360 ft/sec, essentially the molecular speed. Given some contact area in the annulus of the same order as the metal surface area the condensation heat transfer will completely dominate the heat up due to wall heat transfer (at $q/A = 3 \times 10^4$ Btu/hr ft²), until saturation temperature is reached. If the condensation occurs when the steam is flowing up the annulus, the transfer of momentum flux to the water will also be efficient. The momentum will be transferred by direct mass convection to the liquid phase rather than by the shear stress. For this reason, it may be possible to vary the momentum flux transfer in an experiment by changing the temperature of the water injected.

Momentum Flux Effects of Heat

The thin film of steam that may be found on the walls due to wall heat transfer would have little primary effect on the direction of the ECC water. The projected momentum fluxes are so high that the flow regime ultimately would be dispersed droplets in the steam. If the water were running down the walls, the momentum flux of the upward steam is again so high that climbing film flow would result. For example, the calculated reverse core flow of 1000 lb_m/sec at 200-400 psi (see Figure 2) results in an upward momentum flux in the annulus of about 4400 lb_m/ft-sec² in a 3 or 6" annulus. Climbing film experiments^(2, 3) with air-water for 1 inch annuli show that the film of water will climb with air momentum fluxes greater than 284 lb_m/ft sec².

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- (2) A. M. Sutey, J. G. Knudsen, "Mass Transfer at the Solid Liquid Interface for Climbing Film Flow in an Annulus Duct." AICHE J. 15, 5 p.719.
 - (3) Wallis & Collier, "Two Phase Flow and Heat Transfer." Dartmouth College. Hanover, N. H. (1968).

The acceleration pressure drop from heat transfer from the walls of expected rates will not cause large pressure effects. Calculations (Appendix D) indicate that acceleration pressure drop effects for driven ECC flow would be on the order of 1 psi in the annulus.

The unknown effects of the wall heat transfer have to do with the break up of the wall film (if one exists) causing it to be swept out more easily by the upward steam. This is sort of a triggering action because the steam flow is rather overwhelming as shown above. If such a mechanism exists in the prototype and not in the experiment, the results obtained with a cold wall experiment would indicate more water than prototypical - would reach the plenum. Using a slightly colder water into which more rising steam condenses with attendant high momentum transfer would allow less water to reach the plenum and give conservative results. The momentum flux is scaled to the prototype in the Battelle Plenum Filling Experiment (PFE) but the mass flux is not. This could cause problems if energy exchange in the form of heat were a large factor in the prototype. This is because vaporization or condensation causes relatively large changes in the volume of the steam phase flowing and a relatively efficient and short spaced transfer of the steam momentum to the liquid phase. This increased or changed pattern of transfer of momentum flux would tend to force the injection water and its added condensed steam upward in the annulus and give less deposit of ECC water in the lower plenum. Condensation provides a transverse driving force causing the vapor to move toward the condensing surface which accounts somewhat for its high efficiency. Wall heating of the ECC water by the hot wall would therefore reduce the amount of heating

that the water would get by condensation of the rising steam and hence reduce the direct momentum interchange by mass transfer. This effect would offset some of the increased exchange due to the break up of the liquid film by the vaporization at the wall.

In momentum scaled experiments without heated walls, the possible difference in momentum transfer due to boiling could be matched by using different temperatures for the injection water. Some tests will need to be run before a complete answer can be given, because of the need to know the flow regime.

APPENDIX A

HEAT TRANSFER PHENOMENA IN THE PIPE AND ANNULUS

Some reactors have ECC injection in the cold leg piping and some inject into the annulus. The purpose of the mixing experiment is to determine what happens to the injected water. The main aspects of mixing and separation effects of a two-phase should be scaled according to the momentum of the phases. This is because the forces of interaction are the result of momentum exchanges. If the flow effects are dominated by heat exchange then the temperature differences or enthalpy differences should be modeled.

It has been suggested that heat transfer from the hot pipe walls may play a role in the coolant injection phase of the LOCA. Some calculations shown below indicate that the pipe wall heat transfer occurs in such a manner that the metal heat conductivity quickly limits the wall heat transfer and the temperature of the wall drops to below the boiling point. The steam created by the quick flashing is a small amount but could create a significant back pressure in a pipe if it did not separate from the unflashed water. The back pressure is higher for smaller pipes and suggests that the tests be done at large scale to minimize exaggeration of this effect.

In an annulus, which has a relatively small hydraulic diameter and lower mass flux at the same mass flow rate, the back pressure is reduced from that in a pipe because in a typical reactor annulus the lower mass flux dominates. However, the situation in the annulus is complicated because the flow is not one-dimensional and the flow may be disproportionate due to back pressure causing the flow to divert for a time to another path.

The calculations have shown, however, that even if the quenching time of the reactor wall is very short* (footnote: This calculation did not use film blanketed coefficients which would make quenching time much longer hence reduce pressure effects) (even if the wall had not cooled due to blowdown flashing) compared to particle transit times, it would probably play a minor role during the long term reflood phenomenon.

Heat Transfer from Heated Pipes and Reactor Walls

- a. CSE blowdowns showed temperatures at the surface very close to that of water and a gradient of 250° to a 1" depth when the temperature was dropping the fastest. See Figure 7.
- b. Calculation of wall temperatures based on free convection boiling.

$$h = .64 (q'')^{.75} \left(\frac{P}{P_s}\right)^{.4} \quad (1) \quad A-1$$

$$P_s = 14.7 \text{ psia}$$

$$\text{where } h = \text{heat transfer coefficient, } \frac{\text{BTU}}{\text{ft}^2 \text{ hr}}$$

$$q'' = \text{heat flux, } \frac{\text{BTU}}{\text{ft}^2 \text{ hr}}$$

$$p = \text{pressure, psia}$$

On saturation line:

$$\frac{p}{103} \approx \left(\frac{T}{330}\right)^{4.481} \quad A-2$$

$$\text{Where: } T = ^\circ\text{F}$$

$$P = \text{psia}$$

$$\text{Then: } P = 103 (T/330)^{4.481}$$

(1) Jakob & Hawkins. Elements of Heat Transfer, 3rd Ed. John Wiley 1957.

This surface heat transfer coefficient was programmed into a solution for a 1 1/2" slab of steel coated with a 1/4" layer of stainless steel according to the difference equations shown below and flooding with 70° water.

For general node

$$\rho c A \Delta x \frac{\partial T_i}{\partial \theta} = k A \frac{\partial T}{\partial x_i} - k A \frac{\partial T}{\partial x}$$

$$\frac{\partial T_i}{\partial \theta} = \frac{k}{\rho c \Delta x} (T_{i+1} - T_i) - \frac{k}{\rho c \Delta x} (T_i - T_{i-1})$$

$$\frac{\partial T_i}{\partial \theta} = \frac{k}{\rho c \Delta x^2} (T_{i+1} + T_{i-1} - 2 T_i) \quad A-3$$

For surface node

$$\rho c A \Delta x \frac{\partial T_o}{\partial \theta} = k A \frac{\partial T}{\partial x} - h A (T_o - T_\infty)$$

$$\frac{\partial T_o}{\partial \theta} = \frac{k}{\rho c \Delta x^2} (T_i - T_o) + \frac{h}{\rho c \Delta x} (T_\infty - T_o) \quad A-4$$

where h is defined:

$$h = .64 (q'')^{.75} \left[\frac{103}{14.7} \left(\frac{T_o}{330} \right)^{4.481} \right]^{.4} \quad A-5$$

$$\text{but } q'' = h (T_o - T_\infty) \quad A-6$$

$$h = .64 h^{.75} (T_\infty - T_o)^{.75} \left[\frac{103}{14.7} \left(\frac{T_o}{330} \right)^{4.481} \right]^{.4}$$

$$h^{.25} = .64 (T_\infty - T_o)^{.75} \left[\frac{103}{14.7} \left(\frac{T_o}{330} \right)^{4.481} \right]^{.4}$$

$$h = .1678 (T_o - T_\infty)^3 \left[\frac{103}{14.7} \left(\frac{T_o}{330} \right)^{4.481} \right]^{1.6} \quad A-7$$

For inner node

$$\rho c A \Delta x \frac{\partial T_6}{\partial \theta} = k A \left[\left(\frac{T_b + T_6}{2} \right) - T_6 \right] - k A (T_6 - T_5)$$

$$\frac{\partial T_6}{\partial \theta} = \frac{k}{\rho c \Delta x^2} \left[\frac{T_b}{2} + T_5 - 1.5 T_6 \right]$$

A-8

The results are shown in Figure 8. They show that a 600° SS clad slab 1-3/4" thick would have evaporated .53157 lb/ft² of water in 0.24 sec. However, temperature of clad has dropped to 212° by 2.5 x 10⁻³ seconds so that this incredibly high heat flux rate no longer applies. The flux would be less than assumed here.

Check: Volume x ρ x c_p x $\Delta T \div h_{fg}$ = lb evaporated

where ρ = density

c_p = heat capacity

ΔT = temperature change

h_{fg} = latent heat of evap.

$$1/4 \times 1/12 \times \frac{.11 \times (600-212) \times 488}{1054} = .411 \text{ lb/ft}^2$$

which compares with .41 given by code from integrated flux.

So in 2.5 milliseconds after contacting 70° water, the hot surface is quenched and heat flux goes down as does the rate of steam production.

If the water flow rate in the pipe is

$$\frac{G}{\rho} = \frac{1.5 \times 10^6}{62} \frac{\text{lbm}}{\text{ft}^2/\text{hr}} = 24 \text{ ft/sec}$$

where G = mass flux

ρ = density of coolant.

Hence water front moves $24 \times 2.5 \times 10^{-3} = .06 \text{ ft} = .7 \text{ inches}$ by the time the metal is quenched behind.

In the more realistic case the quenching takes 30-40 seconds. In this case the fluid can travel the entire length of flow duct (at the lengths involved in power reactor systems) and steam will still be produced at the inlet. That is, the quenching length is very long. Steam would be produced continuously during ECCS flooding. This heat addition, even after heating the water to saturation involves very little generation of steam, hence low acceleration pressure drop. In the annulus the increase in quality for downflow of the coolant is .003 (p4). Rate of steam production is

$$q/A\lambda = \frac{3 \times 10^4 \text{ Btu/hr-ft}^2}{1054 \text{ Btu/lb} \cdot .3600 \text{ sec/hr}} = .008 \text{ lb/ft}^2 \text{ sec (unquenched)}$$

APPENDIX B

DISCUSSION OF EFFECT OF STEAM PRODUCTION BY WALL HEAT TRANSFER ON PRESSURE DROP IN ANNULUS

If the injected water flows downward in the annulus is intended to fill the lower plenum, and if the countercurrent flow of steam from the core is neglected, one can conveniently calculate the pressure drop that might be caused by the generation of steam due to wall heat transfer.

If the annulus flow area A_f is completely filled by the liquid injected and if steam is generated over the entire annulus surface area A_s , then it produces steam at a rate

$$\dot{m} = q/A \cdot \frac{A_s}{\lambda} \quad \text{B-1}$$

where q/A is the wall heat flux

A_s = annulus surface area forming steam

λ = heat of vaporization

At steady state the mass flow is constant so $(u/v)_{in} = (u/v)_{out}$

$$\text{or } u_{in}/v_f = u_{out}/v_{out} \quad \text{B-2}$$

where u = velocity

v = specific volume

v_f = specific volume liquid

$$v_{out} = xv_g + (1 - x)v_f = v_f + v_{fg}$$

B-3

where x is quality out.

The quality out is

$$x = \frac{\dot{m}}{\dot{M}}$$

B-4

where $\dot{M}$ is mass flow rate down annulus.

The pressure drop due to acceleration is

$$\Delta P = \frac{\rho u \, du}{g_c} = \frac{G du}{g_c}$$

B-5

$$G = \dot{M}/A_f$$

B-6

where A_f = flow area

$$du = u_{out} - u_{in} = u_{in} \left(\frac{v_{out}}{v_{in}} - 1 \right)$$

B-7

but $v_{in} = v_f$, so,

$$du = u_{in} \left(\frac{v_f + x v_{fg}}{v_f} - 1 \right) = x u_{in} \frac{v_{fg}}{v_f}$$

B-9

Substituting:

$$\Delta P = \frac{\dot{M}}{g_c A_f} \times u_{in} \frac{v_{fg}}{v_f} = \frac{\dot{M}^2}{g_c A_f^2} v_{fg} \times$$

B-9

$$\text{since } u_{in} = \frac{\dot{M} v_f}{A_f}$$

Then,

$$\Delta P = \frac{\dot{M} \dot{m}}{g_c A_f^2} v_{fg} = \frac{\dot{M} q/A \cdot A_s}{(144) g_c A_f^2 \lambda} v_{fg} \quad \text{B-10}$$

For a prototype

$$\dot{M} = 5000 \text{ lb/sec}$$

$$q/A = 3 \times 10^4 / 3600 \quad \frac{\text{BTU}}{\text{ft}^2/\text{sec}}$$

$$A_s = 4 \times \pi \times D \times z$$

$$z = \text{height} \sim 14 \text{ ft}, D \approx 13 \text{ ft}$$

$$v_{fg} = 27 \text{ ft}^3 \text{ at } 1 \text{ atm}$$

$$\lambda = 1054 \text{ BTU/lb}$$

$$A_f = \pi D T$$

$$T = \text{thickness of annulus} = 0.5 \text{ ft}$$

Substituting

$$\Delta P = \frac{5000 \times 8.333 \times 4 \times 14 \times 27}{144 \times 32.16 \times \pi \times 14 \times (.5)^2 \times 1054} = 1.2 \text{ psi}$$

This is practically a negligible pressure drop for the conditions during reflood.

APPENDIX C

DISCUSSION OF LOFT DATA OF ATTACHMENT 1

Several examples show that the surface temperature is very close to that of the fluid for temperature changes in the 0-40 second time span.

LOFT data for flooding does not agree because the surface is so hot perhaps that it is flashing the water and forming a non-conductive film. A calculation performed below on the bases of semi-infinite slab shows that the heat transfer coefficient of the wall would be about 90 BTU/hr °F ft².

The flux, can be obtained by evaluation the interface temperature.

Solve for T_i to determine the flux for the LOFT data at 10 seconds.

$$T_i = \frac{T_w \frac{k}{\Delta x} + T_s h}{h + \frac{k}{\Delta x}} = \frac{570.128.16 + 90.220}{90 + 128.16} \quad \text{C-1}$$

$$T_i = 555.26$$

$$\text{LOFT Flux } \dot{q}/A = 90(555.26 - 220) = 30173 = 3 \times 10^4 \text{ BTU/hr ft}^2$$

In Kreith* p. 160 are some exact solutions to the transient heat flux problem in a semi-infinite slab. One can use these results,

*F. Kreith, "Heat Transfer", International Text Book Co., 2nd Ed. (1958).

1) to predict the cooling based on the given positions of the thermocouple in the LOFT quenching experiment or 2) one can use the data and the exact solutions to estimate the effective heat transfer coefficient.

Method 1. for:

$$\bar{h} = 1000 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{F}$$

$$\rho = .284 \text{ lb/in}^3$$

$$c = .11 \text{ BTU/lb } ^\circ\text{F}$$

$$k = 360 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{F/in (carbon steel)}$$

$$x = 1/16''$$

$$\alpha = \frac{k}{\rho c} = \frac{360 \times 1/12}{.284 \times 1728 \times .11} = .556 \text{ ft}^2/\text{hr}$$

The Biot modulus is:

$$Bi = \frac{\bar{h}x}{k} = \frac{1000 \cdot 1/16 \cdot 1/12}{360 \cdot 1/12} = .174$$

The Fourier modulus is:

$$\theta = \left(\frac{\bar{h}}{k}\right)^2 \alpha t = \left(\frac{1000}{360 \cdot 1/12}\right)^2 \cdot .174 \cdot \frac{t_{\text{sec}}}{3600} = .0537 t_{\text{sec}}$$

From Kreith, p. 160, Figures 4-16, determine temperature ratio

$$T_R = \frac{T - T_\infty}{T_0 - T_\infty}$$

where:

T = local temp.

T_∞ = fluid temp

T_0 = initial uniform temp.

TABLE C-I

<u>Time</u>	<u>Carbon Steel</u>			<u>Stainless Steel</u>		
	<u>θ</u>	<u>T_R</u>	<u>T</u>	<u>θ</u>	<u>T_R</u>	<u>T</u>
	$\theta \text{ Bi} = .174$			$\theta \text{ Bi} = .488$		
10	.537	.61	452	4.44	.37	360
20	1.074	.49	406.2	8.88	.27	322
30	1.61	.44	387.2	13.33	.24	311
40	2.15	.39	368	17.76	.19	292
50	2.685	.36	357	22.21	.185	290

Based $T_{\infty} = 220$

$T_0 = 600$

$$T_R = \frac{T - T_{\infty}}{T_0 - T_{\infty}}$$

$$T = T_R(T_0 - T_{\infty}) + T_{\infty}$$

$$T = T_R(380) + 220$$

The results of Table C-I (the semi-infinite solution) are shown in Figure 1 by offsetting the time zero by about 4 seconds to allow for the time it took the surrounding temperature to drop to the new low temperature. The solution compared to the data shows that the initial heat transfer coefficient is much less than 1000 BTU/hr ft² °F but that subsequent to quenching, it may be more or the limited heat capacity of the wall is beginning to show an effect.

If the slab is stainless steel $\alpha = \frac{128 \times 1/12}{.282 \times 1728 \times .114} = .182$

$$Bi = \frac{\bar{h}x}{k} = \frac{1000 \cdot 1/16 \cdot 1/12}{128 \cdot 1/12} = .488$$

The Fourier Modulus is:

$$\theta = \left(\frac{\bar{h}}{k}\right)^2 \alpha t = \left(\frac{1000}{128 \times 1/12}\right)^2 \frac{.182}{3600} t_{\text{sec}} = .444 t_{\text{sec}}$$

It is quite obvious from Figure 1, that a constant heat transfer coefficient solution will not match the experimental data. The experimental data must be the result of a varying heat transfer coefficient. Method 2 below is an attempt to find an initial value of this coefficient based on infinite slab analysis.

Method 2.

For S.S. at 10 sec $T = 560$

$$T_R = \frac{560 - 220}{600 - 220} = .8947$$

Look up θ vs Bi in Kreith.

$$\bar{h} = Bi \cdot \frac{k}{x} = \frac{128 \times 1/12}{1/16 \cdot 1/12} = Bi \times 2048$$

$$\bar{h} = \sqrt{\frac{\theta}{\alpha t}} k = \sqrt{\theta} \sqrt{\frac{(128)^2}{.182 \cdot 10/3600}} = 474.4 \sqrt{\theta}$$

Solve for $\bar{h}$.

$$T_R = .8947$$

<u>Bi</u>	<u>$\bar{h}_{Bi}$</u>	<u>θ</u>	<u>$\bar{h}_\theta$</u>
.2	409	.075	129.9
.1	205	.055	111.1
.07	143	.04	91.8
.01	20.48	.03	82.16

<90

This table results in an effective heat transfer coefficient of only 90 BTU/hr ft² °F for the early phases of the LOFT data.

Hence, early flux is $90 \times 350 \approx 3 \times 10^4$

Using estimate of Figures 4-8, p. 149, Kreith.

For .7" wall $\frac{k_s}{hL}$

at 10 seconds

$$F_o = \frac{\alpha \theta}{L^2} = \frac{10}{3600} \cdot \frac{(144)}{(.7)^2} \cdot \alpha$$

$$\alpha = .556 \text{ ft}^2/\text{sec}$$

$$F_o = \frac{1440}{3600} \cdot \frac{.556}{.49} = .453$$

at 10 sec

$$\frac{x}{L} = 1/16.7 = .0892$$

$$T_R = \frac{560-220}{600-220} = .8947$$

From Figure 4-8.

$$\frac{1}{B_c} = \frac{k_s}{\bar{h}L} = 5$$

thickness = $2L$

$$\bar{h} = \frac{k_s}{5L} = 100$$

Agrees with other estimate.

APPENDIX D

Discussion of Effect of Quenching Steam Production on Pressure Drop in Pipes

What effect do the heat transfer rate and wall cooling have on the reflood phenomena?

Consider a case of a hot pipe which is being reflooded with a slug of cool water. If the steam production front is short such as $\tau = 2.5$ milliseconds in duration as for the hot-quenched wall from the heat transfer calculation (Figure 8); and if the slug velocity is u then the distance traveled during major steam production is τu , and

$$u = \frac{G}{\rho} = \frac{\dot{M}}{\rho A_f}, \quad \text{D-1}$$

where A_f is pipe flow area

G = mass flux

$\dot{M}$ = mass flow rate

Because of the steam production at the quenching front, the average density goes down so the velocity of the steam or steam water mixture that is pushed out ahead of the front goes up.

Case 1

If the steam is pushed out ahead, that is, the steam formed separates completely and leaves the solid slug of water behind, then there is a pressure drop associated with acceleration of only the flashed steam to the new higher velocity.

$$\Delta P = \frac{F}{A_f} = \frac{M}{g_c A_f} \frac{du}{dt} \quad D-2$$

where M = amount of steam flashed in a certain region or pipe length,

u = velocity

t = time

F = axial force

A_f = area of pipe.

A mass balance for the flashed steam across the moving flashing zone is,

$$X \frac{u}{V_f} A_f = \frac{1}{V_g} (u + du) A_f, \quad D-3$$

where: X = fraction of total flow which gets vaporized

v_f & v_g = specific volumes of fluid and vapor.

$$\text{Solving: } du = u \left(\frac{X}{v_f} - \frac{1}{v_g} \right) v_g \quad D-4$$

From heat conduction analysis of Appendix A, the average flux of steam vaporized from a hot wall is 0.4 lb of water per ft^2 per 2.5×10^{-3} sec (for the hot quenched surface)

$$\text{evaporation rate} = \frac{.4}{2.5 \times 10^{-3}} = 160 \frac{\text{lb}_m}{\text{ft}^2 \text{ sec}}$$

If the mass considered is that in the zone of major steam production, the length is dz and the area is

$$A_s = \pi D dz \quad \text{for a round pipe}$$

where A_s = steam production surface area

The mass of steam formed on the steam production area

$$m = \phi_s \cdot A_s \cdot \tau$$

where τ = quenching time

$$\text{and } x = \frac{m}{\rho_f A_f \tau u} = \frac{\phi_s \pi D u \tau^2}{\rho_f \frac{\pi D^2}{4} u \tau} = \frac{4 \phi_s \tau}{\rho_f D} = \frac{4 \phi_s \tau v_f}{D} \quad \text{D-5}$$

$$\text{then } \Delta P = \frac{m}{g_c A_f} \frac{du}{dt} = \frac{\phi_s \pi D u \tau^2}{g_c \pi/4 D^2} \frac{u \left(\frac{x}{v_f} - \frac{1}{v_g} \right) v_g}{\tau} \quad \text{D-6}$$

or,

$$\Delta P = \frac{4 \phi_s u^2 \tau}{g_c D} \left[\frac{4 \phi_s \tau v_g}{D} - 1 \right]$$

since $u = G v_f = \dot{M} v_f / A_f$

$$\Delta P = \frac{16 \phi_s \dot{M}^2 v_f^2 \tau}{\pi^2 g_c D^5} \left[\frac{4 \phi_s \tau v_g}{D} - 1 \right] \quad \text{D-7}$$

Check for some typical pipe sizes considering an injection rate of 1600 lb_m/sec in a pipe:

$$\phi_s = 160 \text{ lb}_m/\text{ft}^2\text{-sec}$$

$$v_f = .0167 \text{ ft}^3/\text{lb}$$

$$v_g = 27 \text{ ft}^3/\text{lb at 1 atm}$$

$$\tau = 2.5 \times 10^{-3} \text{ sec}$$

$$D = .5, 1, 2 \text{ ft}$$

$$\dot{M} = 1600 \text{ lb}_m/\text{sec}$$

$$\Delta P = \frac{64 \times 160 \times (1600)^2 \times (.0167)^2 \times 2.5 \times 10^{-3}}{144 \times \pi^2 \times 32.16 \times D^5} \left[\frac{4 \times 160 \times 27 \times 2.5 \times 10^{-3}}{D} - 1 \right]$$

TABLE D-I

<u>D ft</u>	<u>ΔP, psi</u>
.5	1090
1	16.8
2	.257

Case 2

If the steam does not separate from the water but pushes a two-phase mixture out ahead, then the advancing mixture will move faster than the slug front and will increase the quenching length because the slug flow will be preceded by a mixture that will have some cooling effect. However, the flashing of some of the water to steam which then accelerates the entire mass rather than just the steam alone will give a higher pressure drop.

Assuming that the quenching length is known, the mass balance across the quenching length would be

$$\text{mass in} = \text{mass out}$$

$$\frac{u}{v_f} A = \left(\frac{u+du}{v_m} \right) A$$

where v_m = specific volume of mixture

$$v_m = x v_g + (1-x) v_f = x v_g + v_f$$

D-8

u = velocity

x = the fraction flashed

$$\text{solving: } du = u \left(\frac{1}{v_f} - \frac{1}{v_m} \right) v_m$$

D-9

$$= u \left(\frac{v_m}{v_f} - 1 \right)$$

$$= u \left(\frac{x v_{fg} + v_f}{v_f} - 1 \right)$$

$$= u \left(\frac{x v_{fg}}{v_f} \right)$$

$$du = xu \frac{v_{fg}}{v_f}$$

D-10

The mass being accelerated in this case is approximately all the mass in the quenching zone.

$$M = \rho_f u \tau A_f = \frac{u \tau A_f}{v_f}$$

D-11

$$\text{then } \Delta P = \frac{M}{g_c A_f} \frac{du}{dt} = \frac{u \tau}{g_c v_f} \frac{x u v_{fg}}{v_f \tau}$$

D-12

and with eq. D-5

$$\Delta P = \frac{u^2 v_{fg}}{g_c v_f^2} \times = \frac{4 u^2 \phi_s \tau v_{fg}}{g_c D v_f} \quad \text{D-13}$$

$$\Delta P = \frac{4 G^2 v_f \phi_s \tau}{g_c D} v_{fg} \quad \text{D-14}$$

$$G = \dot{M}/A_f = 4 \dot{M}/\pi D^2 \quad \text{D-15}$$

$$\Delta P = \frac{64 \dot{M}^2 \phi_s \tau}{144 \pi^2 D^5} v_f v_{fg} \quad \text{D-16}$$

The result for a typical pipe size at injection rate of 1600 lb_m/sec

$$\dot{M} = 1600$$

$$\phi_s = 160$$

$$\tau = 2.5 \times 10^{-3} \text{ sec}$$

$$v_f = .0167$$

$$v_{fg} = 27$$

$$D = 2$$

$$g_c = 32.16$$

$$\Delta P = 649 \text{ psi}$$

TABLE D-II

PRESSURE DROP DUE TO
EXTREME QUENCHING IN PIPES (eq. D-16)

Injection Rate → Pipe Diam, ft.	Pressure Drop, psi	
	Separated Steam	Homogeneous Steam-Water
	1600 lb _m /sec	1600 lb _m /sec
.5	1090	--
1	16.8	--
2	.257	649

The actual case even if heat flux were this high would be somewhere between these two cases because the steam formed would tend to slip ahead of the fluid and the result would be a more extended front eventually approaching the Case 1 condition.

The front shape is idealized also, and we have considered an extreme case of hot walls with no credit taken for blowdown cooling.

It may be seen that depending on the flow pattern, the pressure drop will be influenced inversely by the pipe size. Hence, proper scaling may be important to eliminate difficulties with experimental data from this effect, or relatively cool walls should be used for small scale tests.

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ENERGY TRANSFER FROM VESSEL WALLS - RWS-3-73

At a December 6, 1972 meeting in Chicago on countercurrent flow with representatives from RDT, REG, PNL, ECL, CE, and ANC, Mr. J. Griffith of AEC-RDT ask me to submit calculations on vessel wall heat fluxes to CE and PNL. That is the purpose of this letter.

At the meeting I expressed the belief that the negligible wall heat flux resulting from relatively thin annulus walls at low temperatures planned in the CE and BNWL tests may yield optimistic results relative to ECC delivery for LOFT and large pressurized water reactors. Without a large wall heat flux, steam generation by the walls will be small and steam condensing by the injected water will be maximized. I mentioned that my estimates of the wall heat fluxes show fluxes could be high and that the walls were only marginally "conduction limited".

Two experiments which relate to the problem have been run since I made the calculations and I am including these results with the calculations.

Calculated Reactor Energy Transfer

Figure 1 shows results of scoping calculations on wall heat flux versus time for a reactor vessel and a thermal shield during blowdown. Both calculations used a constant heat transfer coefficient of $1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ and a saturation reference temperature predicted from a double-ended break depressurization. These values were input to a heat conduction computer program with 35 and 25 radial nodes, respectively, for the vessel wall and the thermal shield. In the case of the vessel wall 11 of these nodes were placed in the 1/8 in. stainless steel vessel liner.

The flux from a PWR fuel rod operating with a power level of 19 kW/ft is also shown on Figure 1 for reference purposes.

The fact that the fluxes out of the vessel wall are higher than those from the thermal shield is attributed to the vessel being thicker and its thin stainless steel liner being backed up with carbon steel. Carbon steel has a conductivity about 2.5 times higher than that of stainless steel.

The fluxes from the core barrel would be close to those from the thermal shield since it has about the same dimensions and is also built from stainless steel.

Experimental Blowdown Energy Transfers

If the stored energy in the wall were to transfer out prior to the time of ECC injection, wall fluxes would not potentially influence ECC delivery to the lower plenum. In a blowdown experiment at ANC using a small vessel with 0.7 in. thick carbon steel walls, the wall surface flux was as shown in Figure 1. As can be seen, the fluxes from this experiment are about the same magnitude as those calculated for a reactor. The initial pressure was 2200 psia and the temperature was 615°F. After 35 seconds the blowdown vessel wall had lost 75% of its energy while the reactor vessel wall has lost only 3-1/2% and still has the potential to maintain a high flux as is evident from Figure 1.

Experimental Top Flooding Energy Transfers

As part of the semiscale program ANC is performing countercurrent flow tests in the semiscale vessel. A preliminary test involved dumping 25 gpm of 50°F water into the heated vessel with no reverse steam flow from the core. The vessel steam pressure for this test was approximately 14 psia. Figure 2 shows an axial midplane fluid and metal temperature versus time trace. The fluid temperature is in the annulus gap 1/16 in. from the vessel wall and the vessel wall thermocouples 1/16 in. from the surface. The fluid temperature indicates that steam was being generated for about 50 seconds. The vessel wall temperature versus time information was used in a conduction computer program to calculate the surface flux shown in Figure 1. The relatively slow increase of the flux in Figure 1 is another indication of partial wall steam blanketing early in time.

Summary

During the time period of 15 to 30 seconds after blowdown when countercurrent flow in the downcomer may present an ECC delivery problem, energy stored in the vessel walls may not be patently ignored. However, I know of no method except experimental which will decisively show that heated walls are not necessary in your planned countercurrent flow tests. The problem is, as usual, that the coupled thermal-hydraulic process can not be accurately analyzed. Although wall heat fluxes as high as 100,000 Btu/hr-ft² are anticipated, proportioning this energy to decreasing the liquid subcooling, vaporizing water or superheating steam is a formidable task.

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We will be happy to provide data to you on the effect of wall heat flux in the semiscale countercurrent flow experiments. This data should give an upper bound on the problem since the semiscale surface area to downcomer volume is larger than will exist in your planned tests.

R. W. Shumway, Section Leader
Experiment Planning & Eval. Section
Experimental Projects Branch

lk

cc: G. McGovern, AEC-ID
J. Griffith, AEC-DRDT

Figure 1 Wall Heat Flux vs Time

Figure 2 Midplane Temperature