UNITED STATES
NUCLEAR REGULATORY COMMISSION
ADVISORY COMMITTEE ON REACTOR SAFEGUARDS
WASHINGTON, D. C. 20555

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MEMORANDUM FOR: P. G. Shewmon, Chairman
ACRS Members

FROM: H. Etherington, ACRS Member

SUBJECT: FLOW BLOCKAGE BY STEAM DURING NATURAL CIRCULATION IN PWRs

At the June 1982 ACRS meeting, there was a brief discussion of this subject. Questions have been asked by the Union of Concerned Sceintists, Dr. Henry Myers, and the ASLAB for Rancho Seco. The purpose of this memorandum is to explore and quantify some fundamentals of the problem.

1. In the absence of a heat sink, steam cannot be condensed, in any amount, by repressurization. When steam (or any other vapor) is compressed, it becomes superheated. For example, the Mollier chart shows that isentropic compression of saturated steam from 1000 psia to 1500 psia results in superheat of about 40°F; irreversible adiabatic compression results in greater superheat.

2. Condensation of a steam pocket is not a simple reversal of the steam formation process, i.e., it should not be assumed that steam formed during a pressure transient can be quickly condensed by restoring the original pressure.

   Steam separates by gravity and accumulates at high spots in the system, but the steam may be a product of flashing over a substantial part of the liquid system. The reverse process, steam condensation, proceeds by heat transfer processes that have no relation to the mass separation process.

3. Simple classical modes of heat transmission are inadequate for rapid condensation of a large steam void. When the system is repressurized, the steam quickly loses its slight superheat by contact with the steel boundary and surface water. Thereafter, the steam and the water surface remain at the saturation temperature corresponding to the new pressure. As steam condenses by contact with water, the latent heat of condensation is transmitted downwards by the very slow process of conduction into stratified water -- the temperature gradient is in the wrong direction for convection.

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Other modes of heat transfer are also investigated in an illustrative calculation which shows that a layer of steam, three feet high, might be condensed in the following times, each mode being treated separately.

1.	Conduction into water	65 hr.
2.	Conduction along a low-alloy steel pipe	18 hr.
3.	Conduction through low-alloy steel pipe from the steam space to just below the surface of the water	3 hr. (?)
4.	Heat loss through pipe to atmosphere	15 hr. (?)

(The last two items are based on unsupported hypotheses -- both are calculable, and item 3 might be worth developing.)

5. At the assumed conditions, steam is a significant heat radiator, and if much of the radiant heat from the steam or pipe could pass through the thin layer of heated water, it would be possible to condense about 1/2 ft/hr of steam by this mode. However, whereas water is transparent to radiation in the visible part of the spectrum, it is relatively opaque to low-temperature heat radiation. Also, any internal radiation "from hot water to cold water" is presumably included in the experimentally determined conductivity. It appears unlikely that radiation could contribute importantly to condensation.

4. <u>Other modes of heat transfer may predict greater rates of steam condensation</u>, but these would have to be justified either generically or on a case-by-case basis.

   It might, for example, be demonstrated that the system is not sufficiently quiescent to sustain a fully stratified thin layer of heated water at the surface; or that alternately raising and lowering the level, by varying the system pressure or by surges, will permit effective heat transfer by alternately heating and cooling the steel pipe.

   On the other hand, sustained interruption of circulation could lead to intrusion of hotter water and even more steam into the hot leg pipes.

5. <u>A high repressurization pressure is strongly favorable to steam condensation</u>.

   The driving force for all modes of heat transfer, except heat loss to atmosphere, is the temperature difference between the

steam and the water, i.e., the difference between the saturation temperatures corresponding to the repressurized pressure and the depressurized pressure.

The numbers given in Section 3 are for depressurization to 980 psia and repressurization to 2000 psia. If the system were repressurized to 1200 psia, the times for modes 1, 2, 3, and 4 would be increased to 1180, 324, 3.6, and 17.5 hr., respectively.

In the event of a small-break loss-of-coolant accident (SBLOCA), the attainable repressurization pressure may be limited by conflicting procedural requirements, or inadequacy of HPI pump head or capacity. With no repressurization and no natural recirculation, heat transfer ceases by all modes discussed in Sections 3 and 4 except by heat-loss through the pipe to the sourroundings.

6. Feasibility of developing a large steam void. It is not the purpose of this memorandum to discuss how steam voids may form in a system during natural circulation, but it is pertinent to inquire whether the illustrative example is reasonable.

The calculation is based on depressurization to 980 psia of a system whose saturation pressure is 1000 psia (a negative temperature margin of 2.44°F). Such a pressure loss might be associated with an open valve in the pressurizer or actuation of a pressurizer spray. For the assumed conditions, a three-foot high steam layer forms at the high points of a loop -- more exactly, three cubic feet of steam in the U-bend region for each square foot of pipe cross section. For two 3 ft. diameter loops, the total volume of steam (at actual conditions) is 42.4 cu. ft. This quantity of steam is associated with expulsion of water to the pressurizer which causes a 12 in. increase in pressurizer level. Since the calibrated height of the pressurizer is 400 in., it appears that much larger steam voids could form without generating strong self-limiting tendencies.

7. An important one-step reduction in steam volume. Repressurization, if permissible, raises the temperature of the steam above that of the pipe or vessel, and the latter becomes a heat sink, causing fairly rapid partial condensation of steam. (This rapid one-step partial condensation is distinct from the slow continuing condensation described in Section 3).

The fraction of steam condensed depends on the initial water temperature, the final pressure, and the rate of repressurization. An illustrative calculation, improbably favorable in these

respects, shows that 45% of the steam could be condensed by this mechanism.

8. Steam voids can be quickly dispersed by forced circulation. Problems arise only when conditions or procedures require shutdown of the reactor coolant pumps. Natural circulation is assumed in following discussions unless otherwise stated.

9. Blockage of an inverted U-bend ("candy cane") in a B&W system.

   (1) If the void does not completely block flow, it will not stop natural circulation.

   (2) If the void completely blocks flow, the heat sink (steam generator) is isolated and the temperature of a subcooled system rises until saturation is reached; cooling then proceeds in the boiler-condenser mode.

   The boiler-condenser mode requires that the steam void extend over several feet of the riser pipe, over the entire U-bend and the upper plenum ("channel") of the steam generator, and down into the steam generator tubes far enough to provide sufficient heat transfer surface to condense the steam. Complete blockage implies that the water level in the riser pipe, allowing for static and dynamic effects of steam bubbles, is low enough to prevent two-phase flow or slug flow over the bend.

10. It appears possible that there is no direct recovery to single-phase natural circulation from the boiler-condenser mode. Replenishment of inventory by make-up pumps will compress the steam void, raising the level of water in the tubes, and probably into the plenum, thereby decreasing the heat transfer surface or isolating the heat sink. There will be some steam condensation by mechanisms described in Sections 3 and 7, but the temperature of the water will slowly rise until it reaches saturation at the increased pressure, boiling will start again and lower the level of the water in the steam generator plenum and tubes until the boiler-condenser mode is re-established at the new pressure. It appears that the recirculation pumps must be started for re-establishment of single-phase recirculation. (The basis for the B&W admonition to "bump the pump"?)

    Note: A LANL draft report "Small-Break LOCA Recovery in B&W Plants" was distributed with a memorandum dated July 19, 1982, T. M. Novak to ASLAB for Rancho Seco. This report is based on a TRAC analysis and concludes that natural circulation can be reestablished by restoring the inventory. The analysis shows

an intermediate condition of slug flow, apparently associated with oscillation of water level. I don't know whether this condition can be demonstrated and quantified in a real system.

11. Non-condensible gas in a void strongly inhibits all modes of steam condensation.

(1) There is no effective mechanism for absorption of a non-condensible gas in a non-flowing sytem. Most of the gas originates, like steam, from regions remote from the surface. Return of gas to the water proceeds by slow mass transfer analogous to heat transfer by conduction, but in the case of a gas, there are no alternative faster modes of mass transfer.

(2) The presence of gas reduces the partial pressure of the steam, and therefore the saturation temperature which provides the driving force for steam condensation. When the saturation temperature of the steam is reduced to that of the water, the water and steel are no longer available as heat sinks.

(3) Hydrogen, in normal concentrations, is not likely to cause a problem. At 1000 psi, the volume of added hydrogen in the system is only 2 to 3 cubic feet. This is small compared with the volume of a steam void that could cause trouble.

(4) Hydrogen from a metal-water reaction or nitrogen from core flooding accumulators could lead to large quantities of non-condensible gases in the steam voids.

12. Water must be supplied to fill a void! If water is not supplied, the system cannot be repressurized except by objectionable increase of bulk water temperature and additional boiling.

Water may be supplied by the pressurizer, by transfer from another voided region (the vessel head), by makeup pumps, or by starting the circulating pumps to disperse the void throughout the system. In a very small break LOCA, these processes may suffice to eliminate a void, possibly becoming effective only after partial depressurization. With a slightly larger break, voids may persist at least until the low pressure emergency cooling system can function.

13. A one-inch vent line at the top of a U-bend could easily eliminate a steam void in a subcooled system as fast as makeup could be supplied.

But venting a steam space in a saturated system without makeup could be an exercise in futility.

14. The main concern is a B&W System with a SBLOCA that is too small to depressurize the system sufficiently for early operation of the low pressure safety systems, yet too large to permit the inventory to be maintained by the high pressure pumps. Loss of inventory leads to formation of steam voids at high spots, and the possibility of "degraded" modes[1] of heat transfer as discussed in Sections 9 and 10.

15. Other Possible Concerns. Other conditions that could cause concern are possible but not likely.

(1) Steam blockage of the U-bends in a B&W system as a result of an operating transient is conceivable. In this case the reactor coolant pumps (RCP) would probably be available to disperse the steam. If the RCPs were not available, it might be possible by repressurization to reduce the steam volume sufficiently to permit passage of water; or it might be possible to depressurize the system and sufficiently reduce the inventory to permit cooling in the boiler-condenser mode. If these procedures cannot be relied on, it may be necessary to review the adequacy of heat transfer modes discussed early in this memorandum.

(2) In U-tube steam generators (W and CE), the level of the secondary system water is normally above the U-bend, and a steam pocket could not form so long as the temperature of the secondary system is below that of the primary system. Departure from the normal condition could lead to conditions similar to those described for a B&W system.

(3) Blockage by non-condensible gas remains as a low-probability occurrence.

[1]The NRC Staff considers single-phase natural circulation and boiler-condenser heat transfer both acceptable.

16. Feed-and-bleed offers an alternative mode of heat removal. This requires use of non-safety-grade components and is not an NRC requirement. Licensing Boards, however, appear to give some weight to this capability.

17. Conclusion. The Committee may want to review the final disposition of this problem, and to be assured that the various possibilities are reflected in sufficiently flexible and understandable operating procedures.

ATTACHMENT A

ILLUSTRATIVE CALCULATION

Initial hot-leg water temperature: 544.6°F (corresponding to saturation at 1000 psia), and at some satisfactory overpressure.

Depressurization, caused by pressurizer malfunction (e.g., open PORV or spray actuation) to 980 psia at high point in the system (saturation temperature 542.2°F).

Calculated void fraction (cu. ft. of steam, at actual temperature and pressure, per cu. ft. of water): 0.097 at high point, decreasing linearly to zero 62 ft. below the final surface of the water.

Water expelled to pressurizer: 87% of steam volume.

Steam formation and condensation in pipe over 62 ft. high.[1]

Height of steam void	3.0 ft.
Associated heat of condensation per sq. ft. of water surface	3690 Btu/ft^2
Difference between saturation temperatures	93.6°F
Condensation time by conduction to water	65 hr.
Condensation time by conduction to carbon steel:	18 hr.

[1] The quantity of steam formed is greater if the pipe extends less than 62 ft. above the vessel outlet, because boiling then also occurs in the much larger volume of the reactor vessel; but much of the extra steam will collect in the vessel head.

Effect of pressure.

Difference between saturation temperatures, t

2000 psia/980 psia	93.6°F
1200 psia/980 psia	25.0°F

Heat of condensation of steam

2000 psia	561 Btu/lb.
1200 psia	641 Btu/lb.

Condensation time by conduction to water or steel is greater at 1200 psia by the factor:

$$(641/561)^2(93.6/25.0)^2 = 18.3$$

Heat capacity of water. If a layer of water could be heated uniformly from the depressurized temperature to the repressurized temperature, condensation of 3 ft. of steam would heat a layer 0.72 ft. thick at 2000 psia, 2.9 ft. at 1200 psia, and 20 ft. at 1000 psia.

ATTACHMENT B

CALCULATION IN SUPPORT OF MEMORANDUM

Steam Table Data

		spec. vol, v		enthalpy		density, 1/v	
psia	°F	liq	vap	liq	vap	liq	vap
980	542.17	.0215	.4557	539.3	1192.6	46.51	2.194
1000	544.61	.0216	.4456	542.4	1191.8	46.30	2.244
			Average	540.85		46.405	

α = steam fraction formed by volume
f = volume fraction of water expelled at average density and enthalpy

Material Balance per cu. ft. of initial liquid:

$$(1-\alpha)46.51 + \alpha 2.194 + f46.405 = 46.30$$

Heat Balance per cu.ft.

$$(1-\alpha)46.51 \times 539.3 + \alpha 2.194 \times 1192.6 + f46.405 \times 540.85 = 46.30 \times 542.4$$

$f = 0.086$ $\alpha = 0.097$

Rough Check

$$46.3(542.4 - 539.3) = \alpha(1192.6 - 539.3) \times 2.194$$

$$\alpha = 0.100$$

Depth corresponding to 20 psi

$$d = 20 \times 144/46.51 = 62 \text{ ft.}$$

Height of steam void

$$62 \times 1/2 \times 0.097 = 3.0 \text{ ft.}$$

Condensation Times

(calculation based on 1 sq ft of water surface)

1. Pressure, psi	980	1000	1200	2020	
2. Saturation temperature t_s, °F	542.17	544.61	567.2	635.8	✓
3. Water enthalpy, saturated, Btu/lb	539.3	542.4	571.7	671.7	✓
4. Steam " , superheated, "		1194	1213	1233	*
5. Density of water, lb/ft³	46.51	46.30	44.84	38.41	✓
6. " " steam, "	2.194	2.244	2.763	5.325	✓
7. Heat of condensation, Btu/lb, (#4 − #3)		655	641	561	✓
8. Enthalpy increase of water, from sat. at 980 psi to t_s, (#3 − 539.3)		3.9	32.4	132.4	✓
9. Heat of condensation per ft of steam height referred to 980 psi sat, (#7 × 2.194)		1436	1406	1231	✓
10. Depth of water, hypothetically heated to saturation, to condense 1 ft of steam, (#9 ÷ #5 ÷ #8)		7.92	0.968	0.239	✓
11. Time T to condense 1 ft of steam by conduction into water (see below), hr		14200	131	7.07	✓
12. Ditto by conduction into steel pipe		3920	36	1.98	✓

Time (#11 and #12) varies as square of steam height

Heat Conduction Properties

	Cond. k	Sp. Ht c_p	Density w	Diffus. $a = k/c_p w$	$\alpha = k/\sqrt{a\pi}$	
Water, 550°F (sat.)	.3246	1.31	45.9	.00540	2.49	Use 2.46 ✓
" 600°F "	.2919	1.51	42.4	.00456	2.44	✓
Low-alloy Steel, 575°F	25	0.14	490	.364	23.4	✓

Steel Cross Section per Sq Ft of Water Surface, Ratio R

Design pressure 2500 psi, design stress 25,000 psi, thickness b, radius r

$$b = 2500\,r / 25{,}000 = 0.1\,r$$

$$R = 2\pi r b / \pi r^2 = \underline{0.2}$$

For semi-infinite medium at uniform initial temp. t_0 raised to constant surface temp. t_1

(* - Estimate, detailed steam tables not available) HE telecon 11/8/82
RSS

Heat flux per sq ft of water surface at time T:

$$h = k(t_1 - t_0)/\sqrt{\pi a T} = \alpha(t_1 - t_0)/\sqrt{T}$$

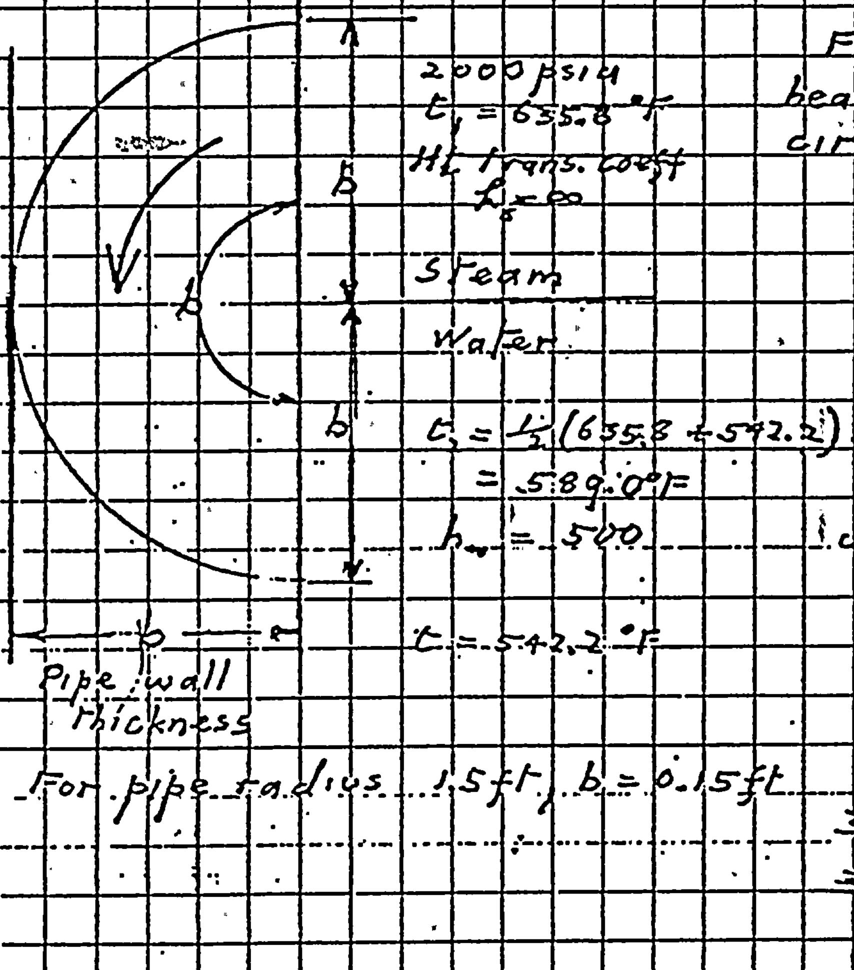
Total heat flow per sq ft to time T:

$$H = \int_0^T h\,dT = 2\alpha(t_1 - t_0)\sqrt{T}$$

$$T_w = H^2/[2\alpha(t_1 - t_0)]^2 = (\#9)^2/[4.92(\#2 - 542.17)]^2$$

For steel: $T_s = H^2/[0.2\alpha_s(t_1 - t_0)]^2 = 0.276\,T_w$ ✓

Hypothetical rate of heat flow from steam to water through steel



For mean path length b, heat flow per ft of pipe circumference

$$= \frac{t_1 - t_2}{b/kb + 1/500b}$$

$$= 46.8/(0.04 + 0.013)$$

$$= 883 \text{ Btu/hr}$$

Heat flow per sq ft of water surface

$$= 883 \times \frac{2\pi r}{\pi r^2}$$

$$= 1177 \text{ Btu/ft}^2\text{-hr}$$

For pipe radius 1.5 ft, $b = 0.15$ ft

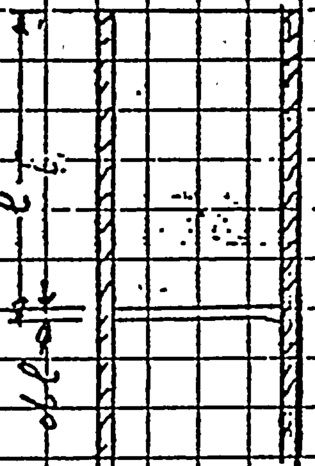
Time to condense 3 ft of steam

$$= 1231 \times 3/1177 = 3.14 \text{ hr}$$

Steam Condensation by Heat Capacity of Pipe

Based on 1 sq ft of water surface

t = steam saturation temperature
C = heat capacity of pipe per ft of length per °F
$C = 0.2 \times 490 \times 0.141 = 13.8$ Btu/ft-°F
w = density of steam, lb/ft³



Assume very slow repressurization, no axial heat flow

Weight of steam $W(t) = w\ell$
Steam condensed $dW = -w\,d\ell + \ell\,dw$

Heat balance:

$$(w\,d\ell + \ell\,dw)\,h_{fg} + C\ell\,dt = 0$$

$$-\frac{d\ell}{\ell} = \frac{C}{w h_{fg}}\,dt - \frac{dw}{w}$$

More clearly: $h_{fg}\,d(w\ell) + C\ell\,dt = 0$

$$\frac{d(w\ell)}{w\ell} = -\frac{C}{w h_{fg}}\,dt$$

$$\ln(\ell w) = -\int \frac{C}{w h_{fg}}\,dt$$

$$\frac{\ell_2 w_2}{\ell_1 w_1} = \exp\left[-\int_{t_1}^{t_2} \frac{C}{w h_{fg}}\,dt\right]$$

Fraction of steam condensed $= \dfrac{\ell_1 w_1 - \ell_2 w_2}{\ell_1 w_1}$

$$= 1 - \exp\left[-C\int_{t_1}^{t_2} \frac{1}{w h_{fg}}\,dt\right]$$

Integrate by Simpson's Rule

psia	t	h_{fg}	$v = 1/w$		$1/w h_{fg} = v/h_{fg}$
980	542.2	653.3	0.4557		.000698
(1420.5)	589.0	682	0.2959	.000434	.001735
2000	635.8	561	0.1878		.000335
				S =	.002768

$$\int = \frac{1}{6} \times .002768\,(635.8 - 542.2) = 0.04327$$

Fraction of steam condensed $= 1 - e^{-13.8 \times 0.04327} = 0.449$

HE 8-1-82