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# The Cooldown Aspects of the TMI-2 Accident

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Prepared for U.S. Nuclear Regulatory Commission

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## The Cooldown Aspects of the TMI-2 Accident

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## ABSTRACT

The cooldown of the TMI-2 reactor vessel due to high pressure injection that occurred at 200 minutes into the accident is re-examined. Flow regimes and condensation heat transfer in the cold legs and downcomer are considered. The presence of noncondensibles (hydrogen) and a mechanism leading to its accumulation around the condensation interfaces lead to conclusions that are materially different from those of a previous study that did not consider these effects.

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## NOMENCLATURE

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с	concentration
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$c_p$	heat capacity
C	solubility
d	diameter
D	mass diffusivity (molecular)
h	heat transfer coefficient, or enthalpy
K	Henry's constant
l	width of liquid layer
$\dot{m}$	mass flow rate
M	molecular weight
Р	pressure
q	heat flux
$\dot{Q}$	heat flow rate
Q	volumetric flow rate
R	gas constant
S	surface area
t	time
T	temperature
$T^*$	saturation temperature
U	velocity
V	volume
w	mass fraction
x	mole fraction
z	axial coordinate

## Subscripts

с	critical
e	exit condition
fg	liqui <b>d</b> -to-vapor

HPI	high pressure injection
i	inlet condition or hydrogen
,. 0	reference state
R	ratio (of molecular weights)
S	steam
t	turbulent
Т	total
1/2	defines a constant-see Appendix B
<sup>∞</sup>	condition in the reactor vessel

## Superscripts

c condensation

## Greek

α.	thermal diffusivity or flow fraction entering core, Eq $(26)$		
λ	length scale of turbulence	•	
$\mu$	viscosity	· .	
ho	density		
u	kinematic viscosity		

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### 1. INTRODUCTION

Previous studies on the Thermal-Hydraulics of the TMI-2 accident have focused on phenomena leading to core uncovery, associated heatup and eventual degradation. With the recent interest in the Pressurized Thermal Shock (PTS) issue [1] it would be interesting to consider, also, the cooldown behavior, particularly that associated with the actuation of the High Pressure safety Injection (HPI) system [2].

In fact, the B&W Owner's Group (OG) task force on PTS considered this problem soon after the accident [3]. The phenomenology invoked in this assessment is schematically illustrated in Figure 1. The basic idea is that the vent valves allowed steam from the rest of the primary system to flow towards the HPI stream and to condense on it, thus heating it from the rather low initial temperature of  $T_i \sim 10^{\circ}$ C (50°F) to a considerably higher level before it reached the cold leg exit into the downcomer. Based on condensation rates alone this exit temperature,  $T_e$ , was estimated at 207°C (406° F). On the other hand, neglecting (conservatively) the quantity of steam already stored within the voided portions of the primary system, an estimate based on available decay heat (i.e., steaming rate) of 141°C (286°F) was given. The fracture mechanics analysis for the TMI-2 vessel was carried out at 141°C (286°F).

In fact within the assumptions of the OG analysis the limit based on condensation rates is conservative, while that based on the decay heat is irrelevant. Each part of this statement is clarified, in turn, below.

The condensation rates were based on the stratified regime (Figure 1), and did not consider the jetting/splashing phenomena (Figure 2) at the point of injection. We have estimated that just over the jet length a heatup to 115°C (249°F) can be expected [4]. Considering the increased interfacial area (wall films, entrainment of drops into the vapor space) generated as the jet splashes against the opposite wall, most of the heatup should have occurred prior to the flow's entering the stratified regime. It does not appear, therefore, that condensation rates could have imposed any limitations on achieving near saturation temperatures at the cold leg exit (i.e., 284°C or 543°F at 1,000 psia).



Fig. 1. Owner's Group evaluation model.



Fig. 2. Flow regime at the injection point.

The decay heat is available to generate steam for only so long as the core remains covered. The fact that a voided cold leg condition is being addressed implies a core already uncovered or well on its way to being so. Thus a portion of the decay heat, equal (roughly) to the portion of the core that still is in contact with liquid coolant should be utilized. In fact, at TMI-2 at the time the HPI was turned on, essentially the whole core was uncovered and severely damaged. Any steaming estimates must, therefore, be based upon fuel quenching and water availability to this process, rather than decay power levels. On the other hand, the steam content of the various voided parts of the primary system could be significant but was neglected in the OG analysis. Even the liquid still present in the lower elevations could have been activated into steaming if the already existing steam inventory continued to deplete and system pressure decreased. This flashing process could further be aided by energy stored in structures still wetted.

There is another aspect to this problem, however, that dominates the heat-up process to such an extent that any uncertainities associated with the above mentioned complex phenomena affecting steam availability are negligible by comparison. This aspect arises because of the presence of noncondensibles, that is, hydrogen and fission product gases, in mixture with the steam. Noncondensibles are notorious in degrading the performance of process condensers. They impose an additional, diffusional, resistance to the condensation process, and because they tend to collect in the immediate vicinity of the condensing interfaces, they can be detrimental even in trace quantities. Even more importantly, in the present case, we will demonstrate a mechanism which leads to continuing accumulation of noncondensibles within the cold leg and eventually to shutting off the condensation process. The rate of accumulation is proportional to the heatup of the HPI (i.e., proportional to the condensation rate), yielding a self-regulating behavior whereby condensation shut-off is rapidly obtained independently of the details used in the calculation.

The purpose of this paper is to reassess the TMI-2 vessel wall temperature transient, due to HPI, in the light of the phenomena introduced above. The assessment proceeds in two steps. In the first, a simple analytical model is used to quantify the condensation transient, as affected by the accumulation of noncondensibles, and hence to arrive at the temperature transient of the HPI water entering the downcomer. In the second step, these results are used in conjunction with methods adapted from past PTS thermal mixing analyses [5] to determine the vessel wall temperature.

#### 2. THE CONDENSATION TRANSIENT

#### Formulation of the Analytical Model

Consider the cold leg/HPI-stream configuration of Figure 3. The cold leg is horizontal, its length is L, its volume is V, and it is initially filled with saturated steam at a pressure  $P_{\infty}$ . The HPI water enters at time t = 0 with a mass flow rate  $\dot{m}_{HPI}$  and a temperature  $T_i$  which is well below the saturation temperature corresponding to the system pressure,  $T^*(P_{\infty})$ . It forms a stream along the bottom of the cold leg and exits at the other end with a temperature  $T_e$ . The cold leg is assumed to comunicate with an *inexhaustible* supply of a steam/hydrogen mixture such that any mass loss from volume V, due to condensation, is perfectly compensated to maintain a constant pressure level  $P_{\infty}$ , throughout the system. The mass flow rate of this supply is denoted by  $\dot{m}_T$  and its hydrogen content is expressed by the mass fraction  $w_{i\infty}$ . The condensation rate over the whole of the liquid/vapor interface (area  $S = \ell \times L$ ) is  $\dot{m}_s^c$ . The removal of hydrogen, from volume V, by absorption into the water stream may be assumed to be neglibible (see Appendix A). As a consequence the hydrogen density,  $\rho_i$ , within the cold leg will increase with time, yielding corresponding transients on  $\dot{m}_T$ ,  $\dot{m}_s^c$  and  $T_e$ . Our task is to relate these quantities through the relevant flow and condensation processes.

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As the liquid heats up from  $T_i$  to  $T_e$ , along its flow path, the potential for condensation decreases. Condensation rates are, therefore, functions of both time and position, z, and as a consequence we would expect the composition within the gaseous space to vary as well. We believe that the essence of the physical behavior may be captured without having to consider this detail. That is, the whole steam/hydrogen space within the cold leg is assumed to be well mixed, and  $\rho_i$  is taken to be a function of time only.



Fig. 3. Transient Condensation Model.

A hydrogen mass balance, over the gaseous volume of the cold leg (without loss of generality taken as V), yields:

$$\frac{d}{dt}(V\rho_i) = \dot{m}_T w_{i\infty} \tag{1}$$

It is convenient to work with mole fractions,  $x_i$ , instead of densities, and steam mass flow rates,  $\dot{m}_s$ , instead of total flow rates. The transformations can be carried out as follows: We use

$$\rho_i = M_i \ c_i = M_i \ c_T \ x_i \tag{2}$$

and

$$c_T = c_i + c_s = \frac{P_i}{RT} + \frac{P_s}{RT} = \frac{P_i + P_s}{RT} = \frac{P_\infty}{RT} = const.$$
 (3)

to express the left hand side of Equation (1), as:

$$\frac{d}{d}(V\rho_i) = V \ M_i c_T \frac{dx_i}{dt} \tag{4}$$

For the right hand side of Equation (1) we use

$$w_i = \frac{\rho_i}{\rho_T}, \quad \rho_i = \frac{P_i M_i}{RT}, \quad \rho_s = \frac{P_s M_s}{RT}, \quad P_i + P_s = P_\infty$$
(5)

$$x_i = \frac{P_i}{P_{\infty}}, \quad x_s = \frac{P_s}{P_{\infty}}, \quad M_R \equiv \frac{M_s}{M_i}, \quad \text{and} \ x_s + x_i = 1$$
 (6)

to obtain

$$w_{i\infty} = \frac{x_{i\infty}}{M_R + (1 - M_R)x_{i\infty}} \tag{7}$$

Clearly,

$$\dot{m}_T = \dot{m}_i + \dot{m}_s \tag{8}$$

and since steam and hydrogen enter the cold leg with the same velocity:

$$\frac{\dot{m}_i}{\dot{m}_s} = \frac{\rho_i}{\rho_s} = \frac{c_i M_i}{c_s M_s} = \frac{x_{i\infty}}{x_s M_R} = \frac{x_{i\infty}}{(1 - x_{i\infty})M_R} \tag{9}$$

therefore,

$$\dot{m}_T = \dot{m}_s \left[ 1 + \frac{\dot{m}_i}{\dot{m}_s} \right] = \dot{m}_s \frac{M_R + (1 - M_R) x_{i\infty}}{(1 - x_{i\infty}) M_R}$$
(10)

Thus, Equation (1) becomes:

$$\frac{dx_i}{dt} = \frac{\dot{m}_s}{V \ M_s c_T} \left\{ \frac{x_{i\infty}}{1 - x_{i\infty}} \right\}$$
(11)

The steam flow rate entering the cold leg,  $\dot{m}_s$ , together with the rate of mass depletion of steam within the cold leg, must constitute the rate of condensation,  $\dot{m}_s^c$ . That is:

$$\dot{m}_s - \frac{dx_s}{dt} c_T \ M_s \ V = \dot{m}_s^c \tag{12}$$

But  $dx_i = -dx_s$  and together with Equation (11), Equation (12) becomes

$$\dot{m}_s = \dot{m}_s^c (1 - x_{i\infty}) \tag{13}$$

This result is used in Equation (11) to obtain the final form:

$$\frac{dx_i}{dt} = \frac{\dot{m}_s^c}{V M_s c_T} x_{i\infty} \tag{14}$$

Next we must relate  $\dot{m}_s^c$  to the appropriate temperatures and heat transfer coefficients. The role of hydrogen in the condensation process is illustrated in Figure 4. The steam is assumed to be in thermodynamic equilibrium with the liquid at the interface. If there had been no diffusional resistance on the vapor side of the interface the concentration distribution would have been uniform, at  $x_i$ ; the interface temperature would have been

$$T_i = T^*[(1 - x_i)P_{\infty}]$$
(15)

and the condensation rate would have been controlled by the turbulence in the liquid and a corresponding heat transfer coefficient,  $h_o$ , i.e.,

$$q^{0} = h_{o} \{ T^{*}[(1 - x_{i})P_{\infty}] - T_{HPI}(z, t) \}$$
(16)

In fact, diffusional resistance of steam against the hydrogen molecules is extremely important. It gives rise to a buildup of noncondensibles at the interface and a corresponding decrease of the interfacial temperature, as shown in Figure 4. That is,

$$T_i = T^*[(1 - x_{ii})P_{\infty}]$$
(17)



Fig. 4. Mechanism of condensation in the presence of noncondensibles.

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Now the heat flux is

$$q = h_o \{ T^*[(1 - x_{ii})P_\infty] - T_{HPI}(z, t) \}$$
(18)

and a significant degradation in heat transfer is seen if  $x_{ii}$  is significantly higher than  $x_i$ . The quantities  $x_{ii}$  and  $T_i$  can be determined by coupling the mass diffusion process on the vapor side to the heat transfer process on the liquid side of the interface. For laminar flow this can be done exactly [6], but for turbulent flow the process remains poorly understood. The experimental data are scarce and highly non-prototypic (steamair system, low pressures, film condensation on cooled walls). Our approach is to express the degradation in heat transfer as an exponential function of the hydrogen mole fraction as follows:

$$\frac{q}{q^0} = \frac{T^*[(1-x_{ii})P_\infty] - T_{HPI}(z,t)}{T^*[(1-x_i)P_\infty] - T_{HPI}(z,t)} = \exp\left\{-\frac{x_i}{x_{1/2}}\right\}$$
(19)

And in combination with Equation (18) we have

$$q = h_o \exp\left\{-\frac{x_i}{x_{1/2}}\right\} \left\{T^*[(1-x_i)P] - T_{HPI}(z,t)\right\}$$
(20)

The basis for Equation (19) is given in Appendix B.

The heatup of the HPI stream can be obtained from an energy balance over a differential element, dz, as shown in Figure 3. That is,

$$\dot{m}_{HPI} c_p \frac{dT_{HPI}}{dz} = \ell h_o \exp\left\{-\frac{x_i}{x_{1/2}}\right\} \left\{T^*[(1-x_i)P_\infty] - T_{HPI}\right\}$$
(21)

which upon integration yields,

$$T_{e} - T_{i} = \{T^{*}[(1 - x_{i})P_{\infty}] - T_{i}\} \left\{ 1 - \exp\left[-\frac{h_{o}S \exp\left\{-\frac{x_{i}}{x_{1/2}}\right\}}{\dot{m}_{HPI}c_{p}}\right] \right\}$$
(22)

This heatup may also be related to the latent heat of condensation by:

$$\dot{m}_{HPI}c_p(T_e - T_i) = h_{fg}\dot{m}_s^c \tag{23}$$

Thus the condensation rate may be written as,

$$\dot{m}_{s}^{c} = \frac{\dot{m}_{HPI}c_{p}}{h_{fg}} \left\{ T^{*}[(1-x_{i})P_{\infty}] - T_{i} \right\} \left\{ 1 - \exp\left[ -\frac{h_{o}S \exp\left\{ -\frac{x_{i}}{x_{1/2}} \right\}}{\dot{m}_{HPI}c_{p}} \right] \right\}$$
(24)

and the exit temperature as,

$$T_e = T_i + \frac{h_{fg} \dot{m}_s^c}{c_p \dot{m}_{HPI}}$$
(25)

The solution,  $x_i(t)$ , is obtained by integrating Equation (14) with  $x_i(o) = 0$ , in conjunction with Equation (24). This ignores some initial accumulation due to condensation already occurring on the makeup flow but does not affect the conclusions of the study. The quantities  $\dot{m}_s^c(t)$  and  $T_e(t)$  can then be readily calculated from the last two equations.

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## The Choice of Parameters for the TMI-2 Case

The relevant geometric dimensions were taken from the design information from Oconee, which is a B&W reactor similar to TMI-2. From the cold leg diameter of 0.7m and its length of 8.15m, between the reactor vessel and the pump, a volume of  $3.1m^3$  is obtained. The volume of the upper 1m of the downcomer corresponding to each one of the four cold legs is  $\sim 1m^3$ . Considering that a good fraction of the inclined portion of the cold leg volume may not participate in the mixing process a value of  $V = 3.5m^3$  was chosen for the calculations. The sensivity to this parameter was examined by also considering a value of  $V = 2.5m^3$ .

The high pressure injection in TMI is believed to have occurred with ~ 15kg/s [7]. We bracket this value by choosing  $\dot{m}_{HPI} = 10kg/s$  for the base case and  $\dot{m}_{HPI} = 20kg/s$  for a sensitivity case. The inlet water temperature is taken at  $T_i = 10^{\circ}C(50^{\circ}F)$ . Assuming critical open channel flow at the cold leg exit, the water stream is estimated to attain a depth of 8cm, a width  $\ell \sim 20cm$ , and a velocity of  $u \sim 1m/s$  (for the 15 kg/s injection rate). The minimum value of the vapor/liquid interfacial area is thus estimated at  $S \sim 1m^2$ .

As seen in Equation (24) the interfacial area, S, appears only in product with the heat transfer coefficient,  $h_o$ . The effect of higher areas, likely to exist due to splashing and entrainment as shown in Figure 2, was covered by considering a generous range in the sensitivity analysis for  $h_o$ . As discussed in Appendix C a realistic estimate for this quantity is  $h_o \sim 1,300$  BTU/hr ft<sup>2o</sup>F. Accounting for a moderate increase in interfacial area due to splashing and associated enhancement of  $h_o$  (jet condensation, etc.) the low end of the range for  $h_o$  considered in the calculation was set at 5,000 BTU/hr ft<sup>2o</sup>F (i.e., nearly a factor of four higher). The base case value was taken nearly as one order of magnitude higher at  $h_o \sim 10,000$  BTU/hr ft<sup>2</sup>°F. The upper end, at  $h_o \sim 50,000$  BTU/hr ft<sup>2</sup>°F represents an enhancement by a factor of x37 and is judged to adequately cover the most extreme behavior.

For the base case the liquid heat capacity was taken at  $c_p \sim 1.3$  BTU/lb°F. Because of the wide temperature range present the sensitivity to this parameter was examined by considering also the case of  $c_p \sim 1$  BTU/lb°F. The saturation temperatures and latent heat of condensation  $(h_{fg})$  were obtained from the steam tables. The system pressure,  $P_{\infty}$ , was taken as 1,000 psia. Sensitivity to this parameter was examined by considering the case of  $P_{\infty} \sim 2,000$  psia. The molecular weight of steam is  $M_s = 18$  g/g-mole.

As discussed in Appendix B for  $x_{1/2}$  a value of 0.05 is deemed appropriate. To generously cover for uncertainities and not to overestimate the degradation in heat transfer, a base case value of  $x_{1/2} \sim 0.1$  was selected. The upper end of the range was set at  $x_{1/2} \sim 0.2$ .

Finally, the choice of  $x_{i\infty}$ , must be made on the basis of core steaming and oxidation rates at the time of high pressure injection (around 200 minutes after the start of the accident). It is now known that by this time the TMI-2 core had already been severely oxidized and partially molten. There seems to be some indication that the molten material relocated in the lower portion of the core where it froze forming substanital blockages [8]. Eventually some 20 to 30 tons of core material made it into the lower plenum where it was found in a solidified debris form. At 174 minutes the "B" loop primary coolant pumps were actuated for a short period of time. As a result some loop seal water was displaced into the reactor vessel yielding a vigorous steaming process as verified by the rapid pressurization to 2,000 psia seen around that time. The details of this process will remain highly uncertain; however, there should be little doubt that the renewed steam availability reactivated the oxidation processes (hydrogen production). On the other hand, at 142 minutes the pressurizer block valves were closed, they were opened briefly (~5 minutes) at 192 minutes and remained closed until 220 minutes. All indications are that the pressurizer remained more than 3/4 full throughout the accident; it is unlikely, therefore, that any venting of steam (and non-condensible gases) occurred during this brief opening of the pressurizer block valve. Thus at the time of HPI operation (200 to 217 minutes) the major fraction of hydrogen already produced and of the fission products released from the disrupted fuel rods, were still present within the primary system volume. The total amounts of gases released in the accident are estimated at [9]: Kr ~ 3.5 kg,  $Xe \sim 40.6kg$  and  $H_2 \sim 510kg$ .\* The total primary system volume is ~10,000 ft<sup>3</sup>. Assuming that loop seals (~ 2,700 ft<sup>3</sup>) and lower plenum (~  $685ft^3$ ) were still full with water the total steam mass, saturated at 2,000 psia, is estimated at 18,976 kg. A low bound of noncondensibles mass fraction may be obtained by assuming a homogenous distribution throughout the steam volume. For Kr, Xe, and  $H_2$  we obtain 0.018%, 0.214%, and 2.68% respectively. Clearly, only the hydrogen is significant. Its concentration may be obtained on a mole fraction basis as  $x_{i\infty} \sim 29.7\%$ . On this basis we chose the values  $x_{i\infty} = 0.05, 0.1, 0.2, 0.3$  and 0.4 to parametrically cover a broad range around the above estimated value. For the base case we chose  $x_{i\infty} = 0.2$ .

#### Numerical Results and Discussion

The solution for the base case is shown in Figures 5 to 7. A rapid buildup of hydrogen in the cold leg and a concomitant decrease in steam condensation rate are predicted. As a result, the HPI water heat up is drastically reduced with cold leg exit temperatures approaching inlet values within a matter of 1 to 2 minutes. All parameter sensitivity results lead to the same conclusion. The  $x_{i\infty}$  and  $x_{1/2}$  parameterics are summarized in Figures 8 and 9. The results of the  $x_{i\infty} = 0.4$  parametric calculation are given in Appendix B. All other parametric results are collected in Appendix D.

<sup>\*</sup> A quantity of only 25.6 kg of hydrogen would be sufficient to obtain a mole fraction of 0.4 in all four cold legs.



Fig. 5. Predicted hydrogen concentration transient for base case.





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Fig. 9. HPI temperature transient at cold leg exit for a variety of condensation degradation assumptions  $(x_{1/2} \text{ parametric})$ .

## 3. THE DOWNCOMER COOLDOWN TRANSIENT

From the above analysis we expect that cold HPI water, nearly at the injection temperature of  $\sim 50$  F, entered the TMI-2 downcomer for the major portion of the injection period. Our task here is to determine the resulting vessel wall cooldown.

We will attempt to look at the problem from two complementary perspectives. First, we examine whether any portion of the HPI stream entering the downcomer could have come in contact directly with the reactor vessel wall. Second, we consider the downcomer fluid temperature transient as it fills with the HPI.

#### Direct Contact Mechanisms

The first critical consideration is whether the HPI cold stream comes directly into contact with the vessel wall. An experimental run at the UCSB 1/2-Scale integral thermal mixing facility with an injection flow rate of 1.2 kg/s revealed the flow regime indicated in Figure 10. That is, the stream impinged upon the core barrel side of the downcomer and fell along it as a well-defined attached film. It is our judgment that this flow pattern should also have prevailed at full scale under the 15 kg/s flow rate. If there were no obstacles along the path of this film the flow would have continued smoothly until it reached the water level within the downcomer. Unfortunately, such obstacles did exist and very likely caused an abrupt deflection, of the downwards flow, laterally, towards the reactor vessel wall as illustrated in Figure 11.



Fig. 10. Flow regime at 1/2-scale and 1.2 kg/s.



Fig. 11. Mechanism for contact with vessel wall.

The detailed geometry of those obstacles is illustrated in Figure 12. The obstacles are called clips. Twenty of them are attached to the core barrel to support the upper end of the thermal shield. From the design information provided [10], their positions relative to the cold legs could be determined as illustrated in Figure 13. Clearly, nearly one-half of the flow must have impinged upon the upper surfaces of those clips causing it to splash upon the reactor vessel wall. The other half must have continued undisturbed and been confined between the core barrel and the thermal shield space.

This behavior was actually demonstrated by means of a simple model experiment. The open air arrangement involved a horizontal acrylic tube positioned opposite a vertical wooden wall fitted with a model clip as illustrated in Figure 14. The relative positions, dimensions (except for the tube diameter), and flow rates of tap water were at 1:1 (full) scale. The flow regime is illustrated, from several perspectives in Figure 14. The vigorous splashing and flow diversion away from the core barrel observed in this experiment leave no doubt that the areas of the TMI-2 vessel under the cold leg nozzles were exposed directly to near 50°F water.



Fig. 12. Representation of clip position and geometry (all dimensions in cm).





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Fig. 14. Full scale demonstration of the off clip splashing.

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#### A REMIX-Type Cooldown Analysis

In addition to the direct contact of cold water discussed above the TMI-2 vessel wall was gradually exposed to the fluid filling up the downcomer. The situation is illustrated in Figure 15. Except for the thin thermal plumes (created as the falling films enter the fluid volume) which mix quickly as they descend, the rest of the fluid volume would be well mixed. This situation is very similar to that of a portion of the REMIX procedure utilized in the usual HPI thermal-mixing analyses dealing with a liquid-full cold leg situation [5].

Let  $V_f(t)$  be the volume of the fluid within the lower plenum and the downcomer corresponding to each of the four cold legs as shown in Figure 15. The density of this fluid is denoted by  $\rho_m$ . The cold stream enters at a volumetric flow rate  $Q_e$  and a density  $\rho_e$ . A fraction,  $\alpha$ , of the flow,  $Q_o$ , which would have to exit if the control volume was fixed in time is assumed to enter the core, while the remaining,  $(1 - \alpha)Q_o$ , is utilized to increase the fluid volume within the downcomer. Based on available flow areas within the core and the downcomer, the maximum value of  $\alpha$  is 0.6. For a fully obstructed core  $\alpha = 0$ . In the presence of vent valves any steam binding would be insignificant. Since the degree of lower core blockage in TMI is not known, the quantity  $\alpha$  is treated as a parameter within the above range.

An overall mass balance may be written as

$$\frac{d}{dt}(V_f \rho_m) = Q_e \rho_e - \alpha Q_o \rho_m \tag{26}$$

The rate of change of the control volume is given by

$$\frac{dV_f}{dt} \doteq (1 - \alpha)Q_o \tag{27}$$

The energy balance accounting, for the thermal energy conducted out of the metal structural components, as they become submerged,  $\dot{Q}_w$ , is written as

$$\frac{d}{dt}(V_f \rho_m h_m) = Q_e \rho_e h_e - \alpha Q_o \rho_m h_m + \dot{Q}_w$$
<sup>(28)</sup>




Together with the equations of state

$$\rho = f(h)$$
 and  $T = g(h)$  (29)

we have a closed system of five equations in the five unknowns, namely,  $\rho_m, h_m, T_m, Q_o$ and  $V_f$ .

For an approximate estimation of  $Q_w$  the donwcomer was discretized axially into four equal segments. When the level of fluid volume  $V_f$  reached the lower end of each axial segment, the conduction calculation for that segment was initiated and continued to the end of the calculation. The total heating was obtained by summing up the contribution of all the thus-activated segments.

Solutions were obtained by marching out, numerically, in time for specified initial conditions.

### Choice of Parameters and Numerical Results

With the exception of the initial fluid volume,  $V_f(o)$ , and of the initial temperature of the non-submerged protion of the structures, the choice of parameters in this rather basic analytical model is straightforward. The relevant geometric quantities were taken to be those of the Oconee reactor (see Table 1). The cold water was taken to enter at  $T_e = 50^{\circ}$ F with a mass flow rate of  $Q_e \rho_e = 15 kg/s$ . Parametric calculations with  $T_e = 60^{\circ}$ F and 70°F also were performed. As mentioned earlier the value of  $\alpha$  was considered parametrically within the 0.0 to 0.6 range.

The possibility that additional condensation upon the core barrel film which would increase  $T_e$  prior to contact with the downcomer water level should also be mentioned here. From the thermal shield length (Table 1) and an assumed lateral spread of ~ 1 m we obtained an additional condensation surface area of ~ 5.3  $m^2$ . The downcomer volume is ~ 6.3  $m^3$ . Employing the results of the condensation parametrics we see that the additional area (a factor x5.3 from base case) is amply covered by the  $h_o \sim 50,000$ BTU/hr ft<sup>2°</sup> F parametric caculation (factor of x37). Also as we can see from Equation (14) the increased volume (a factor of x1.8 from the base case) may be thought of as compensated by the  $x_{i\infty} \sim 0.4$  parametric (i.e., a factor of x2 from the base case). Thus, again, unless an unforseen phenomenon strongly mitigates the condensation inhibiting effect of noncondensibles the values of  $T_e$  utilized in this analysis would not be materially altered.

## Table 1 - GEOMETRIC CONFIGURATION OF OCONEE

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	Cold Leg	Vessel/ Downcomer	Lower Plenum	Pump	Loop Seal	Core Barrel	Thermal Shield
Inner Diameter/ Width (ft)	2.33	14.22	14.22				
Length (ft)	24.5	18.6				18.6	16.0
Base Metal Wall Thickness (ft)	0.21*	s 0.703	0.36*			0.19*	0.167
Clad Thickness (ft)	0.01*	· 0.016	0.016				
Insulation Thickness (ft)	0.30	0.30	0.30				
Wall Heat Tr. Area to Water (ft <sup>2</sup> )	179.3	207.7	69.2			176.8	304.1**
Internal Structures: Heat Tr. Area (ft <sup>2</sup> ) Thickness (ft)		، 		, 			
Fluid Volume (ft <sup>3</sup> )°	104.5	176.3	153.3				

Injector diameter: 0.177 ft

\* Assumed

\*\* Both sides

Per cold leg

The initial fluid volume within the downcomer/lower-plenum region is uncertain. Prior to the "B" loop pump actuation the downcomer was essentially empty. There have been some estimates, however, that the pumps displaced nearly 1,000  $ft^3$  of water from the cold leg piping into the reactor vessel. This would have been sufficient to fill the downcomer completely if the core had been completely blocked, or to fill one-half of the core and downcomer volumes if free flow into the core had been possible. However, the HPI occurred nearly one-half hour later, and in all likelihood this was adequate time to allow, again, downcomer depletion. The value of  $V_f(o)$  was, therefore, taken equal to the lower plenum volume (divided by four to account for the four cold legs). The effect of any larger initial fluid volume would have been to decrease, somewhat, the cooldown rate.

The initial temperature of the downcomer structures is also uncertain. The water displaced, by the "B" pumps, from the cold leg piping (loop seal volumes principally) had been stagnant for a considerable length of time and it could have cooled somewhat from its normal operating temperature of  $\sim 530^{\circ}F$ . We ignore this efffect and take all structures (core barrel, thermal shield, vessel wall) as well as the fluid within  $V_f$  to be initially at  $530^{\circ}F$ . In addition, a parametric calculation was run assuming a core barrel temperature of  $1,000^{\circ}F$  to reflect, approximately, consistency with the highly degraded core conditions at that time.

The results are shown in Figures 16 to 18. The sensitivity to parameter  $\alpha$  is seen to be rather small. The effect of the overheated core barrel is also seen to be small. For the maximum duration of high pressure injection quoted [7], temperatures of ~ 200°F are being predicted. The breaks in the curves are due to the discretization of the structural heat input. They are useful in indicating the time-wise progression of the water level in the downcomer. In particular the last break, which has been marked in the Figures, indicates the time that the downcomer is 100% full.

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Fig. 16. Downcomer fluid temperature transients for base case.



Fig. 17. Downcomer fluid temperature transients for  $T_e = 60^\circ$  F.



Fig. 18. Downcomer fluid temperature transients for  $T_e = 70^\circ$  F.

### 4. CONCLUSIONS

If the high pressure safety injection that was initiated at ~ 200 minutes took place for ~ 15 minutes at 15 kg/s, the TMI-2 vessel wall temperatures reached levels well below 200°F. The principal factor in this conclusion is degradation of heat transfer due to the presence of noncondensibles (hydrogen, fission product gases). A mechanism leading to the accumulation of noncondensibles within the TMI-2 cold legs has been described and indicates that the above conclusion would be true even if the noncondensible concentration in the upper plenum area was rather low. Furthermore, the HPI water is predicted to have entered the empty downcomer at ~ 60°F and a mechanism for direct contact of a portion of this flow with restricted areas of the reactor vessel wall has been demonstrated.

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### Appendix A

### Hydrogen Absorption in the HPI

The mass flux of hydrogen into the turbulent HPI stream may be estimated from [A.1]

$$\dot{m} \approx CM_i \left\{ \frac{Du'}{\lambda} \right\}^{1/2}$$
 (A.1)

where D is the molecular diffusivity of hydrogen in water, u' and  $\lambda$  are the velocity and length scales of the turbulence energy-containing eddies, C is the solubility of hydrogen in water, and M is the molecular weight of hydrogen.

The diffusivity,  $D_o$  in 20°C water is given [A.2] as  $5 \times 10^{-5} cm^2/s$ . For any other temperature, T, the diffusivity may be obtained by

$$D = D_o \frac{T}{T_o} \frac{\mu_o}{\mu} \tag{A.2}$$

where  $\mu$  is the viscosity of water at T. For example at 160°C the viscosity is 0.174 cp [A.3] and  $D \sim 4.2 \times 10^{-4} cm^2/s$ .

The solubility may be related to the partial pressure of hydrogen,  $P_i$ , through Henry's law by

$$x = \frac{P_i}{K} \tag{A.3}$$

For 100°C water the Henry's law constant K is given [A.3] as  $K = 5.73 \times 10^7 mmHg$ . At a hydrogen partial pressure of 1 bar we can thus calculate an equilibrium hydrogen mole fraction of  $1.32 \times 10^{-5}$ . For a partial pressure of ~100 bar (assuming a high value of  $x_i$ ) we have  $x \sim 1.32 \times 10^{-3}$ , which corresponds roughly to  $C \sim 7.3 \times 10^{-5}$  g-moles  $H_2/cm^3H_2O$ .

The turbulence length and velocity scales are chosen as 8 cm and 30 cm/s respectively (see Appendix C).

With the above estimates Equation (A.1) yields  $\dot{m} = 5.76 \times 10^{-6} g/cm^2 s$ . The exposed area within the cold leg is of the order of  $1m^2$ . Thus a hydrogen mass of ~ 3.5g could be absorbed per minute, or ~ 52g for the 15 minute duration of the high pressure injection.

On the other hand, a steam volume of  $3.5m^3$  at 105 bar at hydrogen mole fractions of 0.2 and 0.4 would contain ~ 3,200 and ~ 6,400g, respectively. Clearly the dissolution of  $H_2$  into the water cannot provide an effective mechanism for countering the hydrogen accumulation caused by condensation.

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### Appendix B

# Degradation of Condensation Rates Due to Noncondensibles

No directly applicable experimental data could be located in this area. Our approach is, therefore, based on cautious utilization of available data, in low pressure steam/air systems.

First, let us consider how the binary diffusivity in the hydrogen/steam system compares with that of the air/steam system. At low pressures the diffusivisity may be related to the critical pressures, temperatures and molecular weights of the components [B.1]

$$P^{0}D_{1_{2}}^{0} = 3.64 \times 10^{-4} \left\{ \frac{T}{\sqrt{T_{c1}T_{c2}}} \right\}^{2.334} (P_{c1}P_{c2})^{1/3} (T_{c1}T_{c2})^{5/12} \left\{ \frac{1}{M_{1}} + \frac{1}{M_{2}} \right\}^{1/2} \quad (B.1)$$

At 533°K this yields  $P^0 D_{1}^0 \sim 4.91$  atm  $cm^2/s$ . This value may be converted to high pressure using the critical properties of the mixture

$$P'_c = \sum_j x_j P_{cj}$$
 and  $T'_c = \sum_j x_j T_{cj}$ 

and a graphical representation of the ratio

$$\frac{P \ D}{P^0 \ D^0} = K$$

given by Bird et al. [B.1]. For example for  $x_i = 0.1$  we have  $T'_c = 585.3^\circ$  K and  $P'_c = 197.3$ bar. Thus K = 0.6 and with the previously obtained value of  $P^0D_{1\ 2}^0$  we obtain, for P =102 bar,  $D \sim 2.9 \times 10^{-2} cm^2/s$ . At 373 K the same procedure yields  $D \sim 1 \times 10^{-2} cm2/s$ . Very similar results are obtained also for  $x_i = 0.2$ . At the other extreme of  $x_i = 0.8$ we have  $T'_c = 155.8^\circ K$ ,  $P'_c = 54$  bar, and  $K \sim 1$ . Thus the  $D \sim 4.8 \times 10^{-2} cm^2/s$  and  $2 \times 10^{-2} cm^2/s$  values are obtained for the high and low temperature cases respectively. These values are to be contrasted with a value of  $D \sim 0.239 cm^2/s$  for an 8°C air/steam system and a value of  $D = 0.634 cm^2/s$  for the hydrogen/air system at 0°C and 1 bar. At 100°C the above values are multiplied by a factor of x2. It is clear, therefore, that the high pressure steam/hydrogen system (present application) would give rise to considerably higher diffusional resistance to condensation than the low pressure steam/air systems which have been previously studied experimentally. Stein et al. [B.2] carried out experiments of steam condensation in the presence of air. Condensation took place in the underside of a horizontal cooled copper plate, at system pressures of 3.1, 6.2 and 12.4 bar. We have correlated these data in the manner shown in Figure B.1. An exponential decay with  $x_{1/2} = 0.05$  is indicated. This is consistent with much older data obtained in vertical condensing plate geometries [B.3].



Fig. B.1. Correlation of the Stein, et al. data [B.2] in terms of mole fraction of noncondensibles.

On the other hand, Stein et al. [B.2] have argued that natural convection effects were important in mitigating some of the condensation inhibition efffects of the noncondensibles. If this were true in the air/steam system it would be even more important in the hydrogen/steam system. Unfortunately, the information available does not allow the reliable evaluation of such effects, which are therefore left outside the scope of the present study. In an effort to cover such uncertainties the slower exponential decays of  $x_{1/2} = 0.1$ and 0.2, as shown in Figure B.1 were also considered in the parametric evaluations. Furthermore, if natural convection effects are important their onset should take place at low enough condensation rates where the suction due to condensation is somewhat diminished. Indeed the deviation seen in the data at  $q/q_o \sim 0.1$  may be due to this type of behavior. To cover this possibility and acknowledging the absence of data for  $q/q_o < 0.1$  we have also carried out additional parametric calculations whereby for  $x_i > x_L$  (where  $x_L$ in some specified, limiting value) the  $q/q_o$  decay is specified to linearly approach zero at  $x_i = 1$ . Except for using  $x_{i\infty} = 0.3$  or 0.4 all other parameters were fixed at the base case values. The condensation transient for this kind of behavior is shown in Figures B.2 to B.7. Again a rapid shut-off of condensation is observed, although in this case the hydrogen mole fraction continues to increase approaching values close to unity.



Fig. B.2. Hydrogen concentration transient,  $x_{i\infty} = 0.3$ ,  $x_L = 2x_{1/2}$ .







Fig. B.4. HPI temperature transient,  $x_{i\infty} = 0.3$ ,  $x_L = 2x_{1/2}$ .



Fig. B.5. Hydrogen concentration transient,  $x_{i\infty} = 0.4$ ,  $x_L = 2x_{1/2}$ .



Fig. B.6. Concentration rate transient,  $x_{i\infty} = 0.4$ ,  $x_L = 2x_{1/2}$ .



Fig. B.7. HPI temperature transient,  $x_{i\infty} = 0.4$ ,  $x_L = 2x_{1/2}$ .

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### Appendix C

### The Heat Transfer Coefficient in the Absence of Noncondensibles

In the absence of noncondensibles heat transfer is controlled by the turbulence in the liquid stream. A formulation developed for mass absorption by Theofanous et al. [C.1] about 10 years ago was found also appropriate for condensation by the experimental work of Bankoff [C.2] and Thomas [C.3]. For heat transfer, using  $Pr \sim 1$ , the equations are

$$St_t = 0.25Re_t^{-1/4}$$
 for  $Re_t > 500$  (C.1)

$$St_t = 0.70 Re_t^{-1/2}$$
 for  $Re_t < 500$  (C.2)

The turbulence Stanton and Reynolds numbers are given by:

$$St_t = \frac{h_o}{U'\rho c}$$
 and  $\operatorname{Re}_t = \frac{u'\lambda}{\nu}$  (C.3)

The u' and  $\lambda$ , also called intergral velocity and length scales have been related to the mean flow velocity, U, and hydraulic diameter, D, in a couple of different ways. Theofanous et al. [C.1] used the relations known to apply for pipe flow:  $u' \sim 0.05U$  and  $\lambda \sim 0.03D_h$ . Bankoff [C.2] suggested that for stratified flow of depth  $\delta$ : u' = 0.3U and  $\lambda \sim \delta$ .

Assuming open channel critical flow at the cold leg exit (Fr ~ 1) we can estimate a liquid depth  $\delta \sim 8cm$  and a velocity  $U \sim 1m/s$ . With the Theofanous et al. choices for u' and  $\lambda$  and for 100°C water ( $\nu = 0.0029cm^2/s$ ) we obtain  $Re_t \sim 826$ . The choice of Bankoff yields even higher values. Thus, the regime of Equation (C.1) is applicable. We will first show that both choices for u' and  $\lambda$  lead to very similar results.

We can define a mean flow Stanton number by

$$St = \frac{h_o}{U\rho} \equiv \frac{Nu}{RePr} \sim \frac{Nu}{Re}$$
 (C.4)

and in combination with Equations (C.1) and (C.3) we obtain

$$St = St_t \frac{U'}{U} = 0.25 Re_t^{-1/4} \frac{u'}{U}$$
(C.5)

or

$$h_o = 0.25\rho \frac{\nu^{1/4} {u'}^{3/4}}{\lambda^{1/4}} \tag{C.6}$$

Denoting by  $h_o^T$  and  $h_o^B$  the heat transfer coefficients based on the Theofanous and Bankoff choices of turbulence parameters respectively we have:

$$\frac{h_o^T}{h_o^B} = \left[\frac{5}{30}\right]^{3/4} \left[\frac{100}{3}\right]^{1/4} = 0.62 \tag{C.7}$$

That is a difference between the two predictions of only 38%.

Using  $Re_t = 826$  in Equations (C.1) and (C.2) we obtain  $St_t \sim 4.66 \times 10^{-2}$  and  $2.43 \times 10^{-2}$ , respectively. Let us choose the value  $St_t \sim 3 \times 10^{-2}$ . In combination with Equation (C.5) we obtain  $St \sim 1.5 \times 10^{-3}$  which utilized in Equation (C.4) finally yields  $h_o^T \sim 1,004$  BTU/hr ft2°F. The Bankoff prediction would then be  $h_o^B \sim 1,620$  BTU/hr ft2°F. An in-between value of  $h_o \sim 1,300$  BTU/hr ft2°F will be considered as a best estimate.

These predictions should be viewed, however, with a certain degree of reservation. The reason is that the limited experimental data available in this area were obtained at low pressures. The concern is that at the high pressures, high subcoolings and high heat fluxes of interest here the condensate fluid may be difficult to disipate by turbulent mixing, yielding stratafication and hence a significant reduction of the heat transfer coefficients from the values predicted above.

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# Appendix D

## Parametric Results for the Condensation Transient

Each case is represented by three figures, a, b, and c, depicting the hydrogen mole fraction, the condensation rate, and the HPI exit temperature,  $T_e$ , respectively. Each case is identified by the parameter value varied from the base case. All other parameters remain the same.



D.2



D.3



D.4







Fig. D.4 (a), (b), (c)  $x_{\infty} = 0.10$ 

D.5



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Fig. D.5 (a), (b), (c)  $x_{\infty} = 0.30$ 



Fig. D.6 (a), (b), (c)  $h_o = 5,000 \text{ BTU/hr ft}^{2\circ}F$ 



Fig. D.7 (a), (b), (c)  $h_o = 50,000 \text{ BTU/hr ft}^{2\circ}F$ 





Fig. D.9 (a), (b), (c)  $P_{\infty} = 2,000$  psia





Fig. D.10 (a), (b), (c)  $V = 2.5m^3$ 





D.12

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