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92
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ANALYSIS OF UNPROTECTED OVERCOOLING EVENTS IN THE INTEGRAL FAST REACTOR

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ABSTRACT

Simple analytic models are developed for predicting the response of a metal fueled, liquid-metal cooled reactor to unprotected overcooling events in the balance of plant. All overcooling initiators are shown to fall into two categories. The first category contains those events for which there is no final equilibrium state of constant overcooling, as in the case for a large steam leak. These events are analysed using a non-flow control mass approach. The second category contains those events which will eventually equilibrate, such as a loss of feedwater heaters. A steady flow control volume analysis shows that these latter events ultimately affect the plant through the feedwater inlet to the steam generator. The models developed for analysing these two categories provide upper bounds for the reactor's passive response to overcooling accident initiators. Calculation of these bounds for a prototypic plant indicate that failure limits - eutectic melting, sodium boiling, fuel pin failure - are not exceeded in any overcooling event.

INTRODUCTION

An innovative reactor concept known as the Integral Fast Reactor (IFR) addresses a number of the issues raised in the operating, licensing, and construction environments of the recent past. The concept envisions a nuclear reactor that relies on passive processes for safety and will never need fuel management on- or off-site during its lifetime, making it resistant to diversion of fissionable material for weapons use. The key features of the IFR concept are metal fuel, liquid-metal coolant, a pool-type reactor, and integral reprocessing.

The reactor physics issues of designing for inherent safety were reported recently.¹ Essentially one designs for four generic anticipated transient-without-scrum events - loss of flow (LOF), loss of heat sink (LOHS), rod runout transient overpower (TOP) and balance of plant overcooling. Simple methods exist for analysing the first three events. No simple methods, however, have been developed for the overcooling event. The focus of this paper is on the development of such methods.

Analysis of the unprotected overcooling event is simplified by the fact that the only way this event can influence the core is through the reactor inlet temperature. The response of the core to a change in inlet temperature is given by the quasi-static reactivity balance

$$0 = \Delta\rho = (P - 1)A + (P/F - 1)B + \delta T_{in} C + \Delta\rho_{ext} \quad (1)$$

where

P, F = normalized power and flow respectively,
 δT_{in} = change from normal coolant inlet temperature,
 $\Delta \rho_{ext}$ = externally imposed reactivity,
 A = net (power-flow) reactivity decrement (cents), and
 B = power/flow coefficient (cent/100% P/F).

This equation assumes that the cooling rate is slow enough to preclude nonequilibrium of both fuel pin temperatures and delayed neutron population. This is the case in pool type reactors. The above equation can be solved for the new power level after inherent adjustment of the core neutronics to the changed inlet temperature. Back of the envelope methods for calculating the final inlet temperature, for input into the above equation, are described below.

This paper develops simple methods for estimating the reactor inlet temperature change without having to predict complex transient phenomena. The class of overcooling initiators is shown to divide naturally into two categories, each subject to a separate analysis. The first category consists of events which do not equilibrate to a constant rate of over-cooling, such as in the case of a large steam leak. The second category consists of events which do, as in the case of a loss of feedwater heaters or excessive turbine demand. By solving in each case for a final state that is path independent, a bound for the magnitude of the reactor inlet temperature change is deduced, thereby bounding the response of the core. The first category is analysed using a non-flow control mass approach, the second using a steady flow control volume approach.

To see that there are indeed two categories of overcooling events as described above, consider a fixed boundary control volume enclosing the primary and intermediate systems and the steam generator. The volume is pierced by the steam outlet and feedwater inlet nozzles on the steam generator. The plant is at steady state. For clarity of presentation only, assume the overcooling initiator occurs as a step change. The final state of interest in the control volume will then be either an equilibrium state of constant overcooling or another as yet unknown state. In either case, the overcooling of the control volume must be the result of additional mass and energy flows across the control volume boundary. For the former case, a change in feedwater conditions is the only plausible cause, accompanied, of course, by a corresponding change in steam conditions at the steam nozzle. Specific events include a loss of feedwater heaters and excessive turbine demand. The only other cause, a sustained flow of mass and energy across the volume at points other than the steam and feedwater lines is deemed not credible. For the latter case, the initiator has to be a transient flow of mass and energy across the control volume boundary. A large steam leak is the only credible event. The final state of interest occurs when the steam pressure reaches a pressure of one atmosphere and before the inside of the control volume has time to equilibrate to a time-invariant isothermal state.

NON-FLOW CONTROL MASS ANALYSIS

We look first at plant overcooling resulting from a steam leak having a flowrate in excess of the rated feedwater flowrate. The magnitude of the change in reactor inlet temperature is a function of the heat capacity of the primary and intermediate system and the inventory on the water side.

General Approach

A non-flow control mass approach is used to analyse this problem. The approach avoids having to deal explicitly with the break size area, determining instead the thermodynamic maximum inlet temperature change. The control mass consists of the sodium and structure in the primary and intermediate systems and the water and structure in the water side. The fuel is excluded from the control mass. It is assumed that all the blowdown mixture enters the containment. An envelope is envisioned that contains the control mass and grows in size so as to always contain the escaping blowdown mixture. There is no mass flow across this envelope. Except for the influx of heat from the fuel, this envelope is adiabatic.

The thermodynamic maximum possible change in the control mass internal energy resulting from blowdown needs to be determined. The sodium, structure and water components of the control mass are brought into thermal equilibrium while preserving the total internal energy. With the components in thermal contact, the water is expanded reversibly to a final pressure of one atmosphere. The flow of heat from the fuel into the control mass is neglected. The change in internal energy is

$$\Delta U_{\text{rev}} = - W_{\text{rev}} \quad (2)$$

where W_{rev} is the work performed by the control mass as it expands.

The change in control mass internal energy for irreversible expansion is determined next. Specifically, the control mass components are no longer assumed to be in thermal equilibrium. They are at the temperatures found in the plant before blowdown. The water no longer expands reversibly during expansion. The internal energy change is

$$\Delta U_{\text{irr}} = - W_{\text{irr}} \quad (3)$$

where W_{irr} is the work performed by the control mass as it expands. It is convenient to represent the change in the average temperature of the control mass by ΔT_{irr} corresponding to ΔU_{irr} and to define an irreversibility factor

$$\eta = \frac{W_{\text{irr}}}{W_{\text{rev}}} \quad (4)$$

As a result

$$\Delta U_{\text{irr}} = \eta \Delta U_{\text{rev}} . \quad (5)$$

The effect of heat addition from the fuel is now considered. The first law of thermodynamics implies the following equality

$$\Delta U_{\text{irr-Q}} = \Delta U_{\text{irr}} + (Q - \delta W) \quad (6)$$

where

Q = heat flow from the fuel into the control mass, and

δW = the increase in work performed as a result of heat addition Q .

The second law implies that the additional work obtained as a result of adding heat must be less than the amount of heat added,

$$Q - \delta W \geq 0. \quad (7)$$

Combining the above equation with Eqs. (5) and (6),

$$\Delta U_{\text{irr-Q}} \geq \Delta U_{\text{irr}} = \eta \Delta U_{\text{rev}} . \quad (8)$$

The above results can be combined to bound the cold pool temperature decrease. The internal energy change of the control mass is expanded as

$$\Delta U_{\text{irr-Q}} = \Delta U_{\text{P}} + \Delta U_{\text{I}} + \Delta U_{\text{WS}} . \quad (9)$$

The dynamic nature of the blowdown event implies that at conclusion

$$\Delta T_{\text{WS}} < \Delta T_{\text{irr-Q}} , \quad (10)$$

$$\Delta T_{\text{I}} < \Delta T_{\text{irr-Q}} .$$

where the quantities on the left are the actual temperature changes while the quantity on the right is an average temperature change corresponding to $\Delta U_{\text{irr-Q}}$. The above equations imply that

$$\Delta T_{\text{P}} > \Delta T_{\text{irr-Q}} > \Delta T_{\text{irr}} . \quad (11)$$

Taking the drop in the cold pool to equal that in the primary system gives the temperature drop sought,

$$\left| \Delta T_{\text{COLD}} \right|_{\text{POOL}} < \left| \Delta T_{\text{irr}} \right|. \quad (12)$$

The specific models for computing the bound are described below.

Thermal Equilibrium

The components of the control mass are brought into thermal equilibrium. The resulting state is obtained by solving conservation of energy and volume equations. For the volume of the water inventory, assuming negligible sodium and structure volume change,

$$\begin{aligned} M_{\text{SG}}(v_{\text{f-SG}} + x_{\text{SG}}v_{\text{fg-SG}}) + M_{\text{FW}}v_{\text{FW}} + M_{\text{C}}v_{\text{C}} \\ = M_{\text{W}}(v_{\text{f}} + x_0v_{\text{fg}}) \end{aligned} \quad (13)$$

where

$$M_{\text{W}} = M_{\text{SG}} + M_{\text{FW}} + M_{\text{C}}.$$

For the internal energy of the control mass,

$$\begin{aligned} (M_{\text{NA-P}}T_{\text{NA-P}} + M_{\text{NA-I}}T_{\text{NA-I}})Cv_{\text{NA}} + M_{\text{SG}}(Cv_{\text{W}}T_{\text{SG}} + x_{\text{SG}}U_{\text{fg-SG}}) \\ + (M_{\text{FW}}T_{\text{FW}} + M_{\text{C}}T_{\text{C}})Cv_{\text{W}} + (M_{\text{S-P}}T_{\text{NA-P}} + M_{\text{S-I}}T_{\text{NA-I}} + M_{\text{S-SG}}T_{\text{SG}})Cv_{\text{S}} \\ = (M_{\text{NA}}Cv_{\text{NA}} + M_{\text{S}}Cv_{\text{S}})T_{\text{SATO}} + M_{\text{W}}(Cv_{\text{W}}T_{\text{SATO}} + x_0U_{\text{fg}}) \end{aligned} \quad (14)$$

where

$$M_{\text{NA}} = M_{\text{NA-P}} + M_{\text{NA-I}}, \text{ and}$$

$$M_{\text{S}} = M_{\text{S-P}} + M_{\text{S-I}} + M_{\text{S-SG}}.$$

The above two equations are solved for x_0 and T_{sat0} .

Isentropic Expansion

The control mass is now expanded reversibly without heat addition. Isentropic expansion of this mixture of sodium, structure and water requires that

$$T_m ds_m = dh_m - v_m dP_m = 0. \quad (15)$$

From the definition of mixture enthalpy

$$h_m = \frac{(M_{Na} C_{pNa} + M_w C_{pw} + M_s C_{ps}) T_{SAT} + M_w x h_{fgw}}{M_w + M_{Na} + M_s}. \quad (16)$$

Assuming properties independent of temperature, the differential of the above equation is

$$dh_m = \frac{(M_{Na} C_{pNa} + M_w C_{pw} + M_s C_{ps}) dT_{SAT} + M_w h_{fgw} dx}{M_w + M_{Na} + M_s}. \quad (17)$$

If the volume of the sodium, structure and liquid water are neglected, then

$$(M_{Na} + M_w + M_s) v_m = x M_w v_{gw} = \frac{x M_w R T_{SAT}}{P_m}. \quad (18)$$

Along the saturation line

$$dP_m = \frac{dP}{dT} dT_{SAT} = \frac{h_{fgw}}{T_{SAT} v_{fgw}} dT_{SAT}, \quad (19)$$

so that assuming $v_{fg} \approx v_g$,

$$dP_m = \frac{h_{fgw} P}{R_w T_{SAT}^2} dT_{SAT}. \quad (20)$$

If the above three equations are substituted into the equation above them, we obtain

$$\left[M_{Na} C_{pNa} + M_w C_{pw} + M_s C_{ps} - \frac{x M_w h_{fgw}}{T_{SAT}} \right] dT_{SAT} + M_w h_{fgw} dx = 0. \quad (21)$$

But since

$$T d\left(\frac{x}{T}\right) = \frac{-x}{T} dT + dx, \quad (22)$$

the above equation can be written as

$$\left[M_{Na} C_{pNa} + M_w C_{pw} + M_s C_{ps} \right] \frac{dT_{SAT}}{T_{SAT}} + M_w h_{fgw} d\left(\frac{x}{T_{SAT}}\right) = 0. \quad (23)$$

An expression for the final quality is obtained by integrating this equation,

$$x_1 = \left(\frac{T_{SAT1}}{T_{SATO}} \right) x_0 + \left(\frac{M_{Na} Cp_{Na} + M_w Cp_w + M_S Cp_S}{M_w h_{fg_w}} \right) T_{SAT1} \ln \left(\frac{T_{SATO}}{T_{SAT1}} \right). \quad (24)$$

It may be that all the water is converted to saturated vapor before the final state is reached. This will be the case if the above equation yields a quality greater than one.

If the saturated vapor state is reached, then the above equation is solved for T_{sat1} with x_1 set to unity. This defines the state point from which superheat begins. The model for expansion through the superheat region is similar to that for the saturated region. The mixture enthalpy is

$$dh_m = \frac{M_{Na} Cp_{Na} + M_w Cp_{gw} + M_S Cp_S}{M_{Na} + M_w + M_S} dT \quad (25)$$

where the dependence of Cp_{gw} on temperature and pressure is assumed small. The specific volume of the water vapor is

$$v_m = \left(\frac{M_w}{M_{Na} + M_w + M_S} \right) \frac{R_w T}{P}. \quad (26)$$

Substituting the above two equations into Eq. (15) gives

$$\left(\frac{M_{Na} Cp_{Na} + M_w Cp_{gw} + M_S Cp_S}{R M_w} \right) \frac{dT}{T} = \frac{dP}{P}. \quad (27)$$

Defining the quantity in the brackets to be m , the final state point (P_{SH} , T_{SH}) expressed in terms of the initial state point is

$$T_{SH} = T_{SAT1} \left(\frac{P_{SH}}{P_{SAT1}} \right)^{1/m}. \quad (28)$$

The change in internal energy is

$$\Delta U_{rev} = [(M_{Na} Cv_{Na} + M_S Cv_S) T_{SH} + M_w u_{SH}] - [(M_{Na} Cv_{Na} + M_S Cv_S) T_{SATO} + M_w u_{SATO}] \quad (29)$$

where u_{SH} is the specific internal energy of the superheated vapor and u_{SATO} is the specific internal energy of the water at the initial conditions T_{SATO} and x_0 .

Irreversible Expansion

The change in internal energy for irreversible expansion is related to reversible expansion through Eq. (4). Expanding this equation we obtain

$$\begin{aligned} (M_{Na} C_{vNa} + M_S C_{vS})(T_{irr} - T_{SATO}) \\ + M_w (u_{irr} - u_{SATO}) = n\Delta U_{rev} \end{aligned} \quad (30)$$

where

T_{irr} = final system temperature at one atmosphere, and

u_{irr} = final water specific internal energy at one atmosphere.

This equation is solved for T_{irr} and u_{irr} . An upper bound for the efficiency factor is estimated in the Appendix. We then calculate

$$\Delta T_{irr} = T_{irr} - T_{SATO}$$

and compute the change in cold pool temperature from Eq. (12). The reactor power level at the end of blowdown is then given by Eq. (1).

STEADY FLOW CONTROL VOLUME ANALYSIS

As discussed earlier, all initiators other than the large steam generator leak affect the plant through the steam generator feedwater nozzle. The sign of typical values for A, B, and C in Eq. (1) are such that the reactor will come to a new equilibrium consistent with the new feedwater conditions. It is a simple matter to determine the worst set of core conditions that can be caused by a change in feedwater conditions. The dynamic characteristics of the plant are such that for a step change in feedwater conditions, the core inlet temperature approaches a new equilibrium monotonically. Furthermore, irregardless of how low the feedwater temperature drops, the heat removal from the primary sodium circuit will cease upon sodium freezing in the intermediate sodium loop. The worst case occurs then at steady state with the cold leg of the intermediate system equal to the sodium solidus temperature. The associated reactor inlet temperature is obtained by first writing energy balances

$$Q = (wC_p)_P (T_{PH} - T_{PC}) , \quad (31)$$

$$Q = (wC_p)_I (T_{IH} - T_{IC}) .$$

The log mean temperature model is

$$\frac{T_{PH} - T_{IH}}{T_{PC} - T_{IC}} = \exp UA \left(\frac{1}{(wC_p)_P} - \frac{1}{(wC_p)_I} \right) = K . \quad (32)$$

The above three equations yield

$$T_{PC} = K_1 T_{PH} + K_2 T_{IC} \quad (33)$$

where

$$R = \frac{(wC_p)_P}{(wC_p)_I},$$

$$K_1 = \frac{1 - R}{K - R}, \text{ and}$$

$$K_2 = \frac{K - 1}{K - R}.$$

One can also write

$$T_{PC} = T_{PC_0} + \delta T_i. \quad (34)$$

The quasi-static reactivity model of Eq. (1) at constant flowrate is

$$P = 1 - \frac{\alpha_i \delta T_i}{A + B}. \quad (35)$$

The above three equations yield

$$\delta T_i = \frac{\Delta T_o - \left(\frac{1 - K_1}{K_1}\right) T_{PC_0} + \frac{K_2}{K_1} T_{IC}}{\frac{\alpha_i \Delta T_o}{A + B} + \frac{1 - K_1}{K_1}}. \quad (36)$$

It is interesting to note then that there are actually two cases for the blowdown event. In the first, the blowdown heat loss is insufficient to freeze the intermediate system sodium. The other case is where the sodium does freeze. In that case the core inlet temperature can never drop below the value predicted by the steady flow analysis with the intermediate system cold leg temperature equal to the sodium solidus.

APPLICATION

These simple analytic models provide clear insight into plant behaviour. They can be used to determine how closely safety limits are approached, to determine the role of sodium freezing in mitigating the outcome, and to determine how the outcome is controlled by design parameters. The results of an analysis performed for a prototypic metal-fueled liquid-sodium cooled pool type plant are described. The plant design parameters are summarized in Table I.

The essential result of the analysis is that safety limits are not exceeded in any overcooling event. The safety limits considered were cladding eutectic limit, sodium boiling temperature, and fuel pin failure. This last limit is the overpower at which metal pins have failed in experiments. Some of the results are summarized in Table II.

It is evident from Table II that the inlet temperature drop predicted for a complete water side blowdown (feedwater and condensate systems and steam generator) exceeds that predicted for a frozen intermediate system. As a result overcooling in the blowdown case is terminated early by sodium freezing. The normalized power will never exceed 2.25. However, a blowdown of a lesser inventory, for example only that of the once through steam generator, would not challenge the freezing limit.

APPENDIX IRREVERSABILITY FACTOR

The Marviken Test T-11 results² are used to estimate the irreversibility factor. The test consisted of a vessel containing an equilibrium mixture of liquid and vapor water at an initial pressure of 5.0 MPa and a quality of 0.0333. A valve on the vessel was opened for a period of 80 seconds. Saturated vapor was discharged until the valve closed at which time the vessel pressure was 1.88 MPa. It was determined that the mixture inside the vessel remained in thermodynamic equilibrium for the duration of blowdown.

The value of η is the ratio of the irreversible work performed on the atmosphere which we deduce from the experiment to the reversible work which we calculate from an isentropic analysis. The value of η in the Marviken Test T-11 is believed to lie at the high end of the values that might occur in a pool plant. This is because all the conditions that maximize the value of η appear to have been met in the test but in general are not met in a pool plant. First, depressurization was stopped before atmospheric pressure was reached. The fraction of mass that underwent irreversible expansion was less than if blowdown had terminated at atmospheric pressure. Second, the fluid entering the vessel nozzle was saturated vapor. The steam tables can be used to show for an iso-enthalpic nozzle that the work performed by the nozzle on the atmosphere on a unit mass basis is greatest for saturated vapor inlet conditions. One finds that the ratio of this work to the reversible maximum is about 0.35. In a pool plant, the mixture entering the nozzle will in general be a less efficient two phase mixture. Third, in the Marviken Test T-11 a negligible amount of stored energy was conducted into the water mixture from the vessel during blowdown. By a contrast, a significant amount of heat will be conducted from the structure and sodium in a pool plant. The fraction of the initial mass that exits the vessel increases as the stored energy absorbed by the water increases. Since this mass now expands irreversibly instead of reversibly, the overall efficiency is reduced. The above three points suggest that the irreversibility factor for a pool plant will be less than 0.30.

The actual efficiency in the Marviken test can be deduced. It is useful to conceptualize the blowdown process in terms of three control volumes. The first control volume is the space occupied by the liquid-vapor mixture in the vessel. The two phases expand in thermodynamic equilibrium while the fluid exiting the vessel is saturated vapor. The second control volume consists of an imaginary surface drawn around the region in which the

discharged vapor expands to atmospheric pressure. This expansion region is assumed fixed in space. Because no shaft work is done by the control volume, the expansion process is isoenthalpic. The third control volume is defined to enclose the vapor that has already expanded into pressure equilibrium with the atmosphere. The surface is assumed adiabatic and moves so as to always contain the expanded vapor.

We calculate the irreversibility factor from Eq. (4). The actual irreversible work is that done by the third control volume as it expands. On a unit mass basis this equals the pressure times the specific volume of the vapor in the control volume. The specific volume is evaluated at one atmosphere and at a specific internal energy equal to that of the vapor exiting the isoenthalpic expansion region. With saturated vapor exiting the vessel, this is in turn very nearly equal to the specific internal energy exiting the vessel, which was measured. (This follows from the assumption of isoenthalpic expansion and steam properties.)

The maximum work possible in arriving at the final vessel and ex-vessel conditions in the experiment is also calculated. To do so, two separate masses, the end state vessel mass and the ex-vessel mass, are expanded reversibly to the end state conditions measured in the experiment.

The irreversibility factor calculated from the above irreversible and reversible work is 0.522. Based on the discussion at the start of this appendix, a value of 0.30 is believed to be more appropriate for complete blowdown in a pool plant. This latter value was adopted for the calculations described in the body of this paper.

Nomenclature

C _p	Constant-pressure specific heat
C _v	Constant-volume specific heat
h	Enthalpy
M	Mass
P	Pressure or normalized power
Q	Heat flowrate
R	Ideal gas constant
S	Entropy
s	Specific internal entropy
T	Temperature
U	Internal energy
u	Specific internal energy
v	Specific volume

Greek Letters

ρ	Reactivity
η	Efficiency
χ	Thermodynamic quality

Subscripts

C	Condenser
f	Property of saturated liquid
fg	Difference in property for saturated vapor and saturated liquid
FW	Feedwater
g	Property of saturated vapor
I	Intermediate system
irr	Irreversible
m	Mixture
Na	Sodium
P	Primary system
Q	Heat
rev	Reversible
S	Structure
SAT	Saturated
SG	Steam generator
SH	Superheated
w	water
WS	Water side
0	Initial state of thermal equilibrium
1	Final state of thermal equilibrium

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2. G. A. Grolmes, "Level Swell Analysis of the Marviken Test T-11," Nuclear Science and Engineering, 93, 229-239, 1986.

Table I. Parameter Values for Representative Pool Plant

M_{NA-P} , kg	800,000
M_{NA-I} , kg	47,000
M_{SG} , kg	2,300
M_{FW} , kg	35,000
M_C , kg	140,000
M_{S-P} , kg	390,000
M_{S-I} , kg	27,000
M_{S-SG} , kg	280,000
A + B, cents	-50
C, cents/°C	-0.33
ΔT_0 , °C	150
T_{i0} , °C	357
$(wC_p)_I$, Joules/S	5.788×10^6
$(wC_p)_P$, Joules/S	5.890×10^6

Table II. Summary of Overcooling Results for
Representative Pool Plant

	<u>Normalized Power, P</u>
Thermodynamic analysis of blowdown	
- entire water side ^a	2.28
- steam generator only	1.01
Sodium freezing analysis	2.25
Pin failure in overpower experiments	
(P ₀ = 12 kw/ft) >	4.0

^aSteam generator, feedwater piping, condenser hotwell