APPENDIX 6.2B

EBASCO MODIFICATIONS TO THE CONTEMPT-LT MOD 26 COMPUTER CODE

The containment pressure and temperature transient analyses are performed with an Ebasco modified version of the CONTEMPT-LT Mod 26 computer code.

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In this computer code, the containment volume is divided into two regions, the atmosphere region (water vapor and air mixture) and the sump region (liquid water). Each region is assumed to be completely mixed and in thermal equilibrium. The temperature of each region may be different. Mass and energy additions are made to the appropriate region to simulate the mass and energy release from the Reactor Coolant System or Secondary System during and after blowdown with the contribution of the Safety Injection System (SIS) and Containment Spray System (CSS) water, and decay energy from the core. Account is taken of boiling in the liquid region and condensing in the vapor region, and mass and energy transfers between regions are considered.

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The model represents the heat conducting and absorbing materials in the containment by dividing them into segments with appropriate heat transfer coefficients and heat capacities. Thermal behavior is described by the one-dimensional, multiregion, transient heat conduction equation. The heat conducting segments are used to describe materials and surfaces in the containment which act as heat sources or heat sinks. The model includes provision for mathematically simulating cooling of the containment atmosphere by fan coolers and/or by water sprays, and cooling of the SIS sump water being recirculated to the CSS by the shutdown heat exchangers. The CONTEMPT-LT model and formulations have been shutdown to be applicable and conservative for the design of the containment vessel by simulated design basis accident tests such as the CVTR (see References 2, 3 and 4) blowdown experiment.

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Calculations are begun by computing initial steady state containment atmosphere conditions. Subsequent calculations are performed at incremental time steps. Following the pipe rupture, the mass and energy addition to the atmosphere or liquid region is determined for each time interval. Heat losses or gains due to heat-conducting segments are calculated. Then the mass, and energy balance equations are solved to determine containment pressure, temperature of the liquid and vapor region, and heat and mass transfer between regions.

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The following modifications have been made to the CONTEMPT-LT code by Ebasco:

a) An option has been added to calculate the condensing heat transfer coefficient between the containment atmosphere and the heat sink surfaces by using formulas based primarily on the work of Tagami (see References 5, 6 or 7). From this work, it was determined that the value of the heat transfer coefficient increases parabolically to peak value at the end of blowdown and then decreases exponentially to a stagnant heat transfer coefficient which is a function of the steam to air weight ratio.

Tagami presents a plot of the maximum value of heat transfer coefficient (h) as a function of "coolant energy transfer speed," which is defined by:

$$\frac{\text{total coolant energy transferred into containment}}{\text{(containment free volume) x (time interval to peak pressure)}}$$

From this the maximum value of h is calculated by:

$$h_{max} = 75 \left(\frac{E}{t_p V}\right)^{0.60}$$

where:

h_{max} = maximum value of h (Btu/hr-ft^2-F)

t_p = time from start of accident to end of blowdown (sec)

V = containment free volume (ft^3)

E = initial coolant energy (Btu)

The parabolic increase of h to its peak value is given by:

$$h = h_{max}\sqrt{\frac{t}{t_p}} \qquad o<t<t_p$$

where:

h = heat transfer coefficient between heat sink and air (Btu/hr-ft^2-F)

t_P = Time period of blowdown

t = time from start of accident (sec)

The exponential decrease of the heat transfer coefficient is given by:

$$h = h_{stag} + (h_{max} - h_{stag}) \exp(-0.05 (t-t_p)) \quad t>t_p$$

where:

h_{stag} = h for stagnant conditions = 2.0 + 50.0 X

and

X = steam to air weight ratio in containment

When the containment atmosphere is saturated or superheated and the heat sink surface is below the saturation temperature, the sink heat flux is calculated using:

$$\dot{q}_t = h\ A\left(T_{sat} - T_w\right)$$

where:

$\dot{q}_t$ = heat flux to sink

A = heat sink surface area

T_{sat} = containment saturation temperature

T_W = heat sink surface temperature

b) When either the Tagami of the Uchida[8] condensing heat transfer coefficient option is specified to calculate heat transfer to heat sink surfaces, certain containment conditions can exist for which condensing heat transfer does not occur. For this situation, the CONTEMPT-LT Mod 26 computer code has been modified to calculate the sink heat transfer using the following free convection correlation[9]:

for $T_V \left\langle T_{sat} \text{ or } T_W \right\rangle T_{sat}$

$$\dot{q}_t = h_{fc}\ A\left(T_V - T_W\right)$$

$$h_{fc} = \frac{0.13\ k_f\ \left(Gr_L\ Pr\right)_f^{1/3}}{L}$$

$$= 0.13\left(\beta_f^{\ 2}\quad g\quad \beta_f\ \Delta T\quad C_{pf}\quad k_f\ \ 2/\mu_f\right)^{1/3}$$

where:

T_V = containment atmosphere temperature

g = gravitational constant

β_f = $1/T_f$ where T_f equals the absolute temperature of T_V

ΔT = $T_V - T_W$

C_{pf} = containment atmosphere specific heat at constant pressure

k_f = thermal conductivity of containment atmosphere

μ_f = containment atmosphere viscosity

h_{fc} = free convection heat transfer coefficient

L = characteristic length of heat sink surface

ρ_f = containment atmosphere density

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G_L = Greshof number at heat sink surface

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Pr = Prandlt number of containment atmosphere

c) When either the Tagami or Uchida [8] condensing heat transfer coefficient option is used and it is determined that steam condensation does occur on the heat sink surfaces, the steam condensation rate is calculated using:

$$\dot{m} = \frac{\dot{q}_t}{h_g - h_{film}}$$

where:

$\dot{m}$ = steam condensation rate

h_g = saturated steam enthalpy at containment steam partial pressure

h_{film} = heat sink condensing film enthalpy

This approach is realistic for the following reasons:

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1) When condensation does exist on a heat sink surface, not all energy transfer is due to condensation. The total heat transfer to the sink is actually the sum of a convection and a condensation term.[11] However, the assumption is made when using the Tagami or Uchida heat transfer coefficient options that all heat transfer is due to condensation. Therefore, a conservatively high steam condensation rate is calculated.

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2) Since the net condensation at the heat sink liquid condensate film is actually the difference between the simultaneous process of evaporation and condensation [10], saturation conditions exist in the gas at the interface of the containment atmosphere gaseous boundary layer even if the bulk containment atmosphere is superheated. As can be seen in Figure 6.2B-1, a combination of the convective and diffusive effects in the gaseous boundary layer result in a gaseous interface temperature lower than the bulk containment atmosphere temperature. A 100 percent humidity or saturation condition must exist here since evaporation and condensation processes are simultaneously occurring at the gaseous-liquid interface. Since it is a complicated numerical procedure to calculate the gaseous interface temperature, and since the saturated steam enthalpy is not a strong function of pressure between 1 and 70 psia, it is assumed that the saturated steam enthalpy of the bulk atmosphere is equal to the steam enthalpy at the gaseous-liquid interface. This assumption will result in a maximum of eight percent error in the calculated saturated steam enthalpy between 1 and 70 psia.

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3) The temperature gradient in the condensate boundary layer is small compared to the gradient in the gaseous boundary layer (Figure 6.2B-1). In fact, the gradient in the condensate liquid boundary is small enough to be assumed negligible. The adequacy of this assumption can be shown by the following calculation. The total heat transfer rate from the bulk containment atmosphere to the heat sink surface is assumed to be calculated using the Tagami or Uchida condensing heat transfer coefficients:

$$\dot{q}_t = h\,A\left(T_{sat} - T_w\right)$$

The heat transfer rate in the condensate boundary layer from the gaseous - liquid interface to the heat sink wall is primarily due to conduction and can be written as:

$$\dot{q}_t = \left(-kA\frac{\partial T}{\partial \gamma}\right)_{liquid}$$

where k = thermal conductivity of condensate

therefore:

$$h\left(t_{sat} - t_w\right) = \left(-k\frac{\partial T}{\partial \gamma}\right)_{liquid}$$

Assuming that the value of k is independent of temperature:

$$\dot{q}_t \approx kA\frac{\Delta T}{\Delta \gamma}$$

Solving for ΔT

$$T = \frac{h\left(T_{sat} - T_w\right)\Delta\gamma}{k_{liquid}} \qquad = \frac{\dot{q}_t\ \Delta\gamma}{A\ k_{liquid}}$$

Using the typically most severe containment conditions resulting from a pipe break analysis which maximize the containment pressure and temperature, it can be shown that the temperature gradient across the condensate film is small and can be neglected. Assuming:

T_{sat} = 280°F

T_w = 170°F

A = 1.0 ft^2

h = 200 Btu/(hr –ft^2 -F)

Then a conservative maximum surface heat flux typical of the results expected following a pipe break accident can be calculated as:

$$\dot{q}_t = 200 \; (280 - 170) = 2.2 \text{ X } 10^4 \text{ Btu/hr.}$$

A conservative approximation of the maximum condensate boundary layer thickness can be made assuming the validity of the Nusselt condensation equation for a cool wall in the presence of pure steam [(11), (12)]. The presence of noncondensible gas as air would actually decrease the heat flux and mass condensation rate and consequently decrease the boundary layer thickness. Therefore:

$$\Delta y_{max} = \left[\frac{4 \;\; \mu_f \;\; k_f \;\; Z \left(T_{gi} - T_w \right)}{g \;\; h_{fg} \;\; \rho_f \left(\rho_f - \rho_g \right)} \right]^{1/4} = 0.0013 \text{ ft.}$$

Assuming the following values:

μ_f (at 200°F) = 0.205×10^{-3} lbm/(ft – sec)

k_f (at 200°F) = 0.394 Btu/(hr – ft – F)

Z (conservatively large heat sink height) = 150 ft.

T_{gi} (gas-liquid interface temperature) $\approx T_{sat} = 280°\text{F}$

Note: T_{gi} is actually lower that T_{sat} due to the gas-liquid interface resistance which is large when a noncondensible gas as air is mixed with steam [(13)]. h_{fg} (represents the actual enthalpy drop from the gas to the liquid)=

1173.8 - 138.08 = 1035.72

ρ_f (at 200°F) = 1/0.01663 = 60.13 lbm/ft^3

ρ_g (at 280°F) = 1/8.644 = 0.1157 lbm/ft^3

Using an average value for Δy of 0.00065 ft., the value of ΔT becomes 36°F resulting in an average condensate film temperature of about 188°F (conservatively assuming that the temperature gradient in the film is linear). Therefore, the assumption that the condensate film average temperature is 170°F results in a conservative maximum error of about 10 percent at the time of peak heat flux. In reality, the heat sink surface heat flux and temperature gradient across the condensate film is a function of time and normally much less than these assumed maximum conservative values. Thus, the resulting error in the assumption of condensate film temperature is considerably less than 10 percent throughout the major portion of the transient.

d) An option has been added to calculate the heat removal efficiency of the containment sprays when the containment atmosphere is saturated using:

$$e = \frac{h_e - h_n}{h_f - h_n} \approx \frac{T_e - T_n}{T_f - T_n}$$

where:

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e = spray system efficiency ratio

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T_e = spray water temperature entering sump region, °F

h_e = spray water enthalpy entering sump region

T_n = spray water temperature at spray nozzle exit, °F

T_f = containment vapor region temperature, °F

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h_f = containment vapor region saturated liquid enthalpy

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h_n = spray water enthalpy at spray nozzle exit

Spray thermal efficiency data are taken from Reference 6. These efficiency data are specified as a function of the steam/air mass ration in the containment. The data taken from Reference 6 are for a spray system with a mean spray drop diameter of 600 microns. A conservatively short drop fall height of approximately five meters is used. The spray efficiency is shown in Figure 6.2B-2.

The energy removal rate from the containment atmosphere is then computed from the thermal efficiency and energy transfer by

$$\dot{q} = e\ \dot{m}\left(h_f - h_n\right)$$

where:

$\dot{q}$ = spray energy removal rate, Btu/sec

$\dot{m}$ = spray flow rate, lbm/sec

If the containment atmosphere is superheated, the value of h_e can be solved for using the efficiency data and the definition of spray efficiency. Then h_e and the containment partial steam pressure are used to solve for the final quality (x) of the spray water after interaction with the vapor region. For this case, the energy removal rate is calculated using:

$$\dot{q} = \dot{m}\ (h_f (l - x) - h_n)$$

and the mass addition to the containment atmosphere and sump are calculated as:

$$\dot{m}_S = \dot{m}\,x$$

$$\dot{m}_1 = \dot{m}(1 - x)$$

where: $\dot{m}_s$ = steam addition rate to atmosphere

$\dot{m}_l$ = liquid addition rate to sump region

e) An optional method has been included that determines the steam condensation rate of the containment fan coolers by an interpolation in a table of containment atmosphere saturation temperature versus the fan cooler steam condensation rate.

This additional table is merged with the existing CONTEMPT-LT Mod 26 input of containment atmosphere saturation temperature versus fan cooler heat removal rate.

If the values of fan cooler mass condensation rate input into the fan cooler table are zero, the code will calculate the steam mass condensation rate using the original CONTEMPT-LT Mod 26 assumption.

Following the pipe break inside the containment, the mass and energy of the containment atmosphere, and the mass of the containment sump are updated using the rates interpolated from the input table. Additionally, consistent with the assumptions of the CONTEMPT-LT Mod 26 code, the temperature of the fan cooler condensate is conservatively assumed to be at the containment atmosphere saturation temperature.

Therefore, the condensate energy addition rate to the containment sump is calculated by:

$$\dot{q}_{sump} = \dot{m}_{condensate}\, h_f$$

Where:

$\dot{q}_{sump}$ = steam condensate energy addition rate to the sump region

$\dot{m}_{condensate}$ = fan cooler steam condensate rate obtained from the table interpolation or calculated using the CONTEMPT-LT Mod 26 methods

h_f = saturated liquid enthalpy of containment atmosphere

APPENDIX 6.2B: REFERENCES

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