Enclosure 7

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RSS/wmm
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Participation in Three Mile Island Calculations

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Beginning on the day after the malfunction at TMI I sat in on some very early discussions with Ted Mott of TEC and Bob Hedrick of SAI. The objective was to extrapolate from fuel temperatures then being measured and so predict the temperatures to be expected if the core cooling was limited to natural convection. Working from dimensions in the TMI PSAR, Mott calculated the net buoyant driving force available as a result of heating in the core and cooling in the heat exchanger. This was then equated to the frictional resistance to derive the flow rate and to determine the ΔT necessary to close the loop. See simplified representation in Fig. 1. Difference between pressure in the two legs is

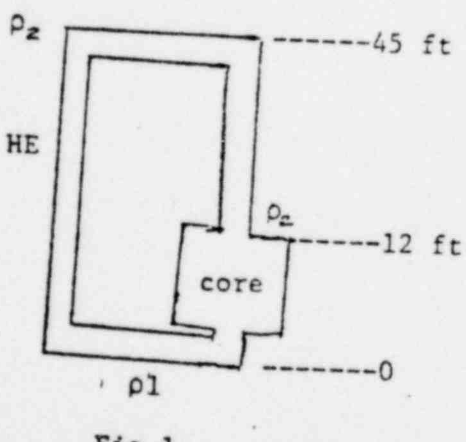

Fig 1

$$45\left(\frac{\rho 1+\rho_2}{2}\right)g - 33\left(\rho_2\right)g - 12\left(\frac{\rho 1+\rho_2}{2}\right)g$$

$$= 33\left(\frac{\rho 1+\rho_2}{2} - \rho_2\right)g = 33\left(\frac{\rho 1-\rho_2}{2}\right)g$$

$$= 16.5\left(\rho 1-\rho_2\right) \quad lb_f/ft^2$$

$$= 0.115 \;\Delta\rho \;\; lb_f/in^2.$$

1006 261

7909240 466

$$\Delta\rho \approx \frac{\partial\rho}{\partial T}\,\Delta T.$$ In the temperature and pressure range measured at the shutdown TMI, $\frac{\partial\rho}{\partial T} \approx .0436\ \mathrm{lb_m/ft^3\ {}^oF}$, so $\Delta\rho = 0.0436\ \Delta T$, and $\Delta P = 0.115 \times 0.0436\ \Delta T = 0.00500\ \Delta T$.

Ted Mott has been using a computer code which includes the friction factor for the entire loop for a typical B&W plant. Using this, $\Delta P = 8.9 \times 10^{-6}\ Q^2$, where Q is flow in lbm/sec. Using this value for ΔP in the relation above,

$$\Delta P = 0.00500\ \Delta T = 8.9 \times 10^{-6}\ Q^2,$$

$$\text{or } Q = \sqrt{\frac{.005\ \Delta T}{8.9 \times 10^{-6}}} = 23.7\ \sqrt{\Delta T}$$

In the core, power is added to the water at a rate

$$\phi = Q\ c_p\ \Delta T.$$

Again at TMI conditions, $c_p \approx 1$, and at a day and a half after shutdown, ϕ = 10Mw = 9483 BTU/sec. Equating the two expressions for Q,

$$Q = \frac{9483}{\Delta T} = 23.7\ \sqrt{\Delta T}$$

$$\Delta T = \left(\frac{9483}{23.7}\right)^{2/3} = 54.3^oF$$

This is a manageable temperature rise. With an intact core and an operational heat exchanger there should be no problem running on natural convection cooling at the 10 Mw level. The concern is that the core may not be intact. At the time of our analysis, thermocouples in one corner were indicating temperatures well above those of the rest of the core. The fear of course was that the high temperature region had blockage and restricted flow.

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At this point the group disbanded to realign with other discussion groups, and with Mott to continue calculations using a core percolation model in place of the core flow assumption. I decided to try some further calculations based on the assumption of quarter-core blockage. The data used by Ted Mott give 0.512×10^{-6} for the friction factor of the core alone. With 3/4 of the core completely blocked (in the extreme case) we would expect to get 3/4 as much flow for the same ΔP. Since flow varies inversely with the square root of the resistance, to change the flow to 3/4 requires increasing the resistance by $\frac{16}{9}$, so I have assumed a resistance of $\frac{16}{9} \times .512 \times 10^{-6} = .91 \times 10^{-6}$ for 3/4 of a normal core.

With one pump forced flow there is a ΔT of about 20° in the "undamaged" part of the core.

So $$Q_u = \frac{P_u}{\Delta T_u} = \frac{3/4 \times 9483}{20} = 356 \ \frac{lb_m}{sec}$$

In the "damaged" part of the core, ΔT is about 160°,

So $$Q_d = \frac{P_d}{\Delta T_d} = \frac{1/4 \times 9483}{160} = 14.8 \ \frac{lb}{sec}$$

Since both sections must have the same ΔP,

$$R_d \, Q_d^2 = R_u \, Q_u^2 \, ,$$

or $$R_d = R_u \left(Q_u/Q_d\right)^2 = .91 \times 10^{-6} \left(356/14.8\right)^2$$
$$= 0.525 \times 10^{-3}$$

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Resistance for damaged and undamaged core sections in parallel is then 0.908×10^{-6}, and for the entire circulation loop would be $(8.9 - .512 + .908) \times 10^{-6} = 9.3 \times 10^{-6}$.

As before, for convection cooling

$$.005\ \Delta T = 9.3 \times 10^{-6}\ Q^2 \quad \text{and} \quad Q = 9483/\Delta T,$$

So

$$Q^3 = \frac{.005 \times 9483}{9.3 \times 10^{-6}}, \qquad \text{or} \qquad Q = 172\ lb_m/\ sec$$

$$\Delta P_{core} = 0.908 \times 10^{-6}\ (172)^2 = 0.0269\ psi$$

$$Q_d = \sqrt{\Delta Pc/.525 \times 10^{-3}} = 7.16\ lb_m/\ sec$$

$$\Delta T_d = \frac{P_d}{Q_d} = \frac{9483/4}{7.16} = 331\ ^\circ F$$

Using 280°F as the inlet temperature prevailing at the time of the comp a- tion, 331°F temperature rise would exceed saturation temperature for the 1050 psi at which the core was then running. For atmospheric pressure at the top of the loop, $\Delta T = 331^\circ F$ would produce boiling in the damaged quadrant for ar inlet temperature.

Since the region showing elevated outlet temperature seemed to shift with changes in the choice of coolant pump, the assumption that part of the core is obstructed has been less defensible. If the cause is simply inertial channeling, there should be no problem under natural convection.

Even if part of the core were obstructed to the extent calculated, ΔT will be proportional to power level. At a month after shutdown, if we assume 2 Mw afterheat, ΔT_d would be $331/5 = 66.2^\circ F$, a level which could be tolerated. There should be some concern over accuracy of ΔT in the hot spot

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region. As better measurements of temperature and flow become available the above numbers would have to be modified; the methodology should be ok.