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CENTRAL FILES NUMBER

57-10-110

DATE: October 21, 1957

SUBJECT: Life of HRE-3 Core Tanks

TO: I. Spiewak

FROM: J. A. Larrimore

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ABSTRACT

Calculations have been made of the expected life of Zircaloy core tanks in the HRE-3 assuming 5 w/ml power density in the tank, considering possible cooling systems. If the inner surface film coefficient is 1000 Btu/hr ft² °F or better, a thick single-walled tank appears optimum. If the inner surface coefficient is only about 100, cooling of the core tank with either D₂O or ThO₂ slurry appears desirable or even necessary.

I. INTRODUCTION

The high in-pile corrosion rate of Zircaloy-2 in UO_2SO_4 solution at reactor operating temperatures has stimulated interest in means of avoiding excessive corrosion damage to the zirconium core tank. In this report, the possibility is considered of increasing the core tank lifetime by the addition of a sacrificial inner zirconium liner, with coolant between the main tank and liner being used to control the wall temperature. For comparison, expected lifetimes of single wall core tanks are also calculated. Several other methods of increasing the lifetime are also considered. The latest interpretation of existing radiation corrosion data is employed.

As the results of this investigation are primarily intended for application to HRE-3 design, a 60 mm two-region spherical reactor, with a 4 foot diameter core and a 2 1/2 foot blanket, was considered. Because of the concentric inlet and outlet to the core, the mean fuel temperature near the wall is uniform over most of the sphere, and is assumed to be 280°C . The power densities assumed are given in Table 1.1(1). Beta heating on the tank surfaces is neglected.

TABLE 1.1

<u>Location</u>	<u>Power Density (w/ml)</u>
On inner liner surface	22
Between tank and liner (for fuel solution coolant)	20
On blanket side tank surface	14
In tank and liner walls	5

Since their thicknesses and separation are small compared to the diameter of the core, the tank wall and liner are treated as parallel plates. Assuming thermal gradients are non-zero only in the radial direction over the major portion of the core wall, the system is simplified to a one-dimensional infinite parallel plate problem.

II. RADIATION CORROSION OF ZIRCALLOY-2

To estimate core tank lifetimes, the corrosion rate of Zircaloy-2 in the HRE-3 fuel solution must be known for the power density and temperature range of interest. The in-pile corrosion program, headed by G. H. Jenks, has supplied a considerable amount of pertinent data. Correlations of in-pile loop and autoclave results on Zircaloy-2 have been made by Jenks(2), and the effect of power density on corrosion rate has been expressed by equations of the general form:

$$\text{CR} = \propto \text{PD} \left(1 - e^{-\frac{\beta}{\text{CR}^{1.5}}} \right) \quad (2.1)$$

where CR is the corrosion rate, PD is the power density, β is a temperature dependent constant, and $\propto$ is a constant depending upon temperature and solution composition.

The proposed HRE-3 fuel solution contains 0.04 m UO_2SO_4 , 0.008 m CuSO_4 , and 0.02 m H_2SO_4 . Corrosion data for this solution is presently available only at 280°C . However, the data for 0.17 m UO_2SO_4 , 0.007 - 0.04 m CuSO_4 , no H_2SO_4 solution at 280°C fits the same equation as the HRE-3 solution(2), and therefore, the temperature dependence of the two solutions is assumed to be the same. Data for the 0.17 m UO_2SO_4 solution is fitted by the equation(2):

$$CR = 2.9 \text{ PD} (1 - e^{-\frac{6.5}{CR^{1.5}}}); T = 250^\circ\text{C} \quad (2.2)$$

$$CR = 2.45 \text{ PD} (1 - e^{-\frac{95}{CR^{1.5}}}); T = 280^\circ\text{C} \quad (2.3)$$

To obtain the temperature dependence of the corrosion rate, it is necessary to consider the changes of α and β with temperature. According to the present theory(3), each of these constants is associated with an activation process so that their temperature variation is represented by:

$$\begin{aligned} \alpha &\sim e^{C_1/T} \\ \beta &\sim e^{C_2/T} \end{aligned} \quad (2.4)$$

where T is the absolute temperature, and C_1, C_2 are activation energies. Fig. 2.1 shows a plot of $\ln \alpha$ and $\ln \beta$ vs. $1/T$ in which the 250°C and 280°C values have been connected by a straight line, according to Eq. (2.4). Values of α and β for various temperatures from Fig. 2.1 are used in Eq. (2.1) to find corrosion rates for the power densities of interest in this investigation. These results are plotted in Fig. 2.2 as $\ln CR$ vs. T . Although no data has been obtained to substantiate the maximum in the corrosion rate in Fig. 2.2, it is not unlikely that it exists, if there is a thermal annealing process occurring as Jenks has suggested(2).

The assumptions involved in Eq. (2.4) are subject to serious doubt, and it is expected that further experimental data will alter Fig. 2.2. Since it is the best available interpretation of the data at present, Fig. 2.2 is used in the lifetime calculations. It should be noted that between 250°C and 280°C , the range of interest in the double wall core tank calculations, Fig. 2.2 is expected to be quite accurate. It is only above 280°C that it is subject to doubt. Further mention of this problem is made during the discussion of single wall core tanks.

For purposes of calculation, the temperature dependence of the corrosion rate is expressed by an equation of the form:

$$CR \text{ (ft/yr)} = A e^{Bt(^{\circ}\text{F})} \quad (2.5)$$

where A and B are found using Fig. 2.2. The values used for a power density of 22 w/ml are given in Table 2.1.

TABLE 2.1

Temperature Range	A ft/yr	B ($^{\circ}\text{F}$)-1
$t \leq 536^\circ\text{F}$ (280°C)	3.07×10^{-7}	0.0166
$536^\circ\text{F} \leq t \leq 583^\circ\text{F}$	4.53×10^{-6}	0.0116
$t \geq 583^\circ\text{F}$ (306°C)	2.08×10^{-2}	-0.00286

The question of the applicability of Jenks' in-pile average corrosion rates to actual reactor operation has been raised. If the corrosion is pitting, or local attack, then the data cannot be applied with much certainty unless a penetration factor relating average penetration to maximum pit depth can be estimated. However, it is felt that local attack is characteristic of low corrosion rates ($< 1 \text{ mpy}$), while general attack is characteristic of high corrosion rates. Under this assumption, Jenks' data can be applied to HRE-3 operation, and the penetration factor can be taken as one.

III. SINGLE WALL CORE TANK

The lifetime of a single wall core tank depends on the corrosion rate on the inner surface, since it is assumed that the slurry corrosion of the outer side of the tank is negligible. In these calculations, the lifetime is taken as the time required for the core tank to corrode from its initial thickness to a particular final thickness. A value of 1/3-in. was used as a final thickness, since preliminary stress calculations⁽⁴⁾ indicate it is approximately the thickness required to withstand normal pressure surges between core and blanket*. Lifetimes were also calculated for corrosion to 1/4-in. final thickness and for corrosion of the full tank thickness.

The core tank lifetime is calculated in the following way. If X is the tank thickness at time τ :

$$\frac{dx}{d\tau} = -CR \quad (3.1)$$

where CR , the corrosion rate on the inner surface, can be expressed by (Eq. 2.5):

$$CR = Ae^{Bt_1} \quad (3.2)$$

For the cases of interest, it will be shown that the inner surface temperature, t_1 can be related to the wall thickness by approximate linear relations of the form:

$$t_1 = Dx + E \quad (3.3)$$

Combining these three equations, the differential equation for the lifetime, with boundary condition, is written:

$$\begin{aligned} \frac{dx}{d\tau} &= -Ae^{BDx + BE} \\ x &= \delta \quad \tau = 0 \\ x &= x_f \quad \tau = T \end{aligned} \quad (3.4)$$

where δ is the initial wall thickness, x_f is the final thickness, and T is the core lifetime.

The wall surface temperature as a function of wall thickness is found by solving for the temperature profile in the core tank wall. The wall is treated as an infinite slab with a uniform volume heat generation rate equal to the power density in the wall. In this and all of the following development, it is found most convenient to characterize the tank surface heat transfer by the surface coefficient h for the system with no volume heat source in the core fluid. This permits description of the system in terms of a relatively easily measurable quantity, and avoids the difficulties involved in attempting to rigorously include the effects of a volume heat source in the fuel solution. The heat generation in the laminar film on the core side of the wall is taken into account by considering a second conduction region of thickness $L = k/h$ where k is the thermal conductivity of the fuel solution and h is the no-volume-heat-source surface coefficient. In this equivalent

*Thermal stresses due to internal wall heating were included in this calculation.

The buoyant force due to the difference in core and blanket densities, and the force due to the change in core fluid momentum were found to give considerably smaller stresses than the allowable pressure stresses.

heat transfer region, a generation rate is assumed, equal to the power density at the wall. It is found that the effect of this heat generation on t_1 is not large for $h > 100$. Estimates of the blanket side surface coefficient, h_b , indicate that the minimum possible value lies around 100, so the generation in the blanket side laminar film is assumed to have a very small effect on t_1 , and is neglected.

The temperature distribution in the core tank wall is a function of the heat generation rates, q''' and Q''' ; the wall thickness, δ ; Blanket the thermal conductivities, k_w and k ; the fluid temperature on both sides of the wall; and the heat transfer coefficients on the two surfaces. The equations governing the two regions are:

Fig. 3.1

$$\frac{d^2 t_a(x)}{dx^2} + \frac{q'''}{k_w} = 0; \quad -\delta \leq x \leq 0 \quad (3.5)$$

$$\frac{d^2 t_b(x)}{dx^2} + \frac{Q'''}{k} = 0; \quad 0 \leq x \leq L \quad (3.6)$$

At the inner surface, continuity of temperature and heat flux are required.

$$t_a(0) = t_b(0) \quad (3.7)$$

$$k_w \left. \frac{dt_a}{dx} \right|_{x=0} = k \left. \frac{dt_b}{dx} \right|_{x=0} \quad (3.8)$$

The two more necessary boundary conditions are taken as:

$$t_b(L) = t_m \quad (3.9)$$

$$q/A - \delta = h_b [t_m - t_2] = -k_w \left. \frac{dt_a}{dx} \right|_{x=-\delta} \quad (3.10)$$

where $t_a(-\delta) = t_2$. The mean fluid temperature near the core wall is 280°C or 536°F . The blanket fluid temperature is taken as 536°F also, since it is the highest expected slurry temperature.

Solving Eqs. (3.5) and (3.6) subject to the given boundary conditions, the wall surface temperatures are found to be:

$$t_1 = t_m + U \left[\frac{q''' \delta}{h} \left(\frac{\delta}{2k_w} + \frac{1}{h_b} \right) + \frac{q''' k}{2h^2} \left(\frac{\delta}{k_w} + \frac{1}{h_b} \right) \right] \quad (3.11)$$

$$t_2 = t_m + U \left[\frac{q''' \delta}{h_b} \left(\frac{\delta}{2k_w} + \frac{1}{h} \right) + \frac{q''' k}{2h^2} \left(\frac{1}{h_b} \right) \right] \quad (3.12)$$

where:
$$U = \frac{1}{\frac{1}{h} + \frac{1}{h_b} + \frac{\delta}{k_w}}$$

Estimates of h_b^* indicate that it will lie in the range 80 - 550, and so calculations were made for an h_b of 80 and of 550. Two values of h were considered, ** 1000 and 100, as these are indicative of the difference between high and low surface coefficients on the inner surface. Wall thicknesses up to 1-in. were considered. The thermal conductivity of Zircaloy-2 in the range 500 - 700°F is

8.1 $\frac{\text{Btu}}{\text{hr-ft}^2\text{-}^\circ\text{F}}$ (5). The conductivity of the fuel solution is taken as 0.336(6) for 536°F and 1500 psia operating pressure.

For $h = 100$, Eqs. (3.11) and (3.12) yield surface temperatures above the saturation temperature (596°F for 1500 psia operating pressure) for some thicknesses and particularly for $h_b = 80$. In these cases, the wall is sufficiently insulated from the fuel and slurry to permit the surface temperatures to reach the saturation temperature. When this occurs, subcooled boiling will take place. For $h = 1000$, the good heat transfer between wall and fluid keeps the inner wall temperature near the mean fluid temperature and the saturation temperature is not reached.

For $h = 100$, $h_b = 80$, subcooled boiling will occur on both surfaces, and Eqs. (3.11) and (3.12) does not hold. The bulk fluid will remain below the Blanket boiling point so that the vapor bubbles will tend to collapse as they move away from the wall. It has been found that the heat flux in such a situation is related to the t_m the difference between wall temperature and saturation temperature by an equation of the form:(7)

$$q/A = C (t_w - t_{\text{sat}})^n \quad (3.13)$$

Fig. 3.2

where t_w is the wall surface temperature, t_{sat} is the fluid saturation temperature, and C and n are constants depending on the liquid and the pressure. Using Rohsenow's 1500 psia high flux asymptote from Ref. 7, the constants are found to be:

$$C = 450 \frac{\text{Btu}}{\text{hr-ft}^2} \left(\frac{1}{^\circ\text{F}} \right)^n$$

$$n = 3.07$$

With boiling at both surfaces, the system is symmetric under the assumptions made. Laminar film heat generation is not considered since the boiling disrupts the film. Because of the symmetry, the wall heat flux is known and the wall temperature is found from:

$$q/A)_w = \frac{q''' \delta}{2} = c (t_w - t_{sat})^n \quad (3.14)$$

so that:

$$t_w = t_1 = t_2 = t_{sat} + \left(\frac{q''' \delta}{2c} \right)^{\frac{1}{n}} \quad (3.15)$$

For $h = 100$, $h_b = 550$, and $\delta \geq 1/2$ -in, Eq. (3.11) predicts $t_1 > t_{sat}$, so that in this case subcooled boiling will occur on the inner surface only. Again assuming the core side laminar film disrupted, the problem now becomes: (Fig. 3.3)

Fig. 3.3

$$\frac{d^2 t}{dx^2} + \frac{q'''}{k_w} = 0; \quad 0 \leq x \leq \delta$$

$$q/A)_o = h_b (t_m - t_2) = -k_w \left. \frac{dt}{dx} \right|_{x=0}$$

$$q/A)_\delta = C(t_1 - t_{sat})^n = -k_w \left. \frac{dt}{dx} \right|_{x=\delta}$$

Blanket

Core

(3.16)

boiling

The solution of this system of equations cannot be written down in simple form. It is found from the following equations:

$$\frac{dt}{dx} = \frac{-q'''x}{k_w} + C_1 \quad (3.17)$$

$$t_1 = t_{\text{sat}} + \left(\frac{q''' \delta}{C} - \frac{k_w C_1}{C} \right) \frac{1}{n} \quad (3.18)$$

$$(t_{\text{sat}} - t_m) + \frac{q''' \delta^2}{2 k_w} = \left(\frac{k_w}{h_b} + \delta \right) C_1 - \left(\frac{q''' \delta}{C} - \frac{k_w C_1}{C} \right) \frac{1}{n} \quad (3.19)$$

For a particular δ , Eq. (3.19) must be solved for C_1 , which can then be used in Eq. (3.18) to find t_1 .

The results of these calculations of t_1 vs. δ for $h_b = 80, 550$ and $h = 100, 1000$ are given in Fig. 3.4. From this figure, the wall temperature as a function of wall thickness may be approximated by the following linear relations for $\delta \leq 0.0833$ ft (1").

$$t_1 = 372 \delta + 536; \quad h = 1000 \quad (3.20)$$

$$\left. \begin{aligned} 4080 \delta + 536; & \quad 0 \leq \delta \leq 0.0146 \text{ ft} \\ t_1 = 598.5; & \quad 0.0146 \text{ ft} \leq \delta \leq 0.0833 \text{ ft} \end{aligned} \right\} \begin{aligned} h &= 100 \\ h_b &= 80 \end{aligned} \quad (3.21)$$

$$\left. \begin{aligned} 1620 \delta + 534; & \quad 0 \leq \delta \leq 0.0395 \text{ ft} \\ t_1 = 599 & \quad ; 0.0395 \text{ ft} \leq \delta \leq 0.0833 \text{ ft} \end{aligned} \right\} \begin{aligned} h &= 100 \\ h_b &= 550 \end{aligned} \quad (3.22)$$

Subcooled boiling is seen to have the effect of keeping the surface temperature slightly above the saturation temperature.

Equations (3.20), (3.21), (3.22) provide the constants D and E for Eq. (3.4). The corrosion constants, A and B, are tabulated in Table 2.1 for the 22 w/ml power density at the core wall. The solutions of Eq. (3.4) for $x_f = 1/3$ -in. and $x_f = 0$ are plotted in Fig. 3.5 as tank lifetime vs. initial tank thickness.

fig 3.5

While the value of h_p , within the range 80-550, has a very small effect on the tank lifetime, the value of h is seen to have a strong effect. In the case of a single wall core tank, the higher the value of h , the longer is the tank lifetime.

Because of the uncertainty in the corrosion correlations in Table 2.1 for $t > 536^\circ\text{F}$, it is of interest to see how bad a situation might be expected. If the straight line correlation in Fig. 2.2 for $t \leq 536^\circ\text{F}$ is extrapolated to 600°F and the appropriate values of A and B are used in solving Eq. (3.4), the results should give the minimum core tank lifetimes, since it is expected that the corrosion rate will not continue to increase exponentially above 536°F . In Fig. 3.6, the lifetimes for $x_f = 1/4$ -in. are plotted for this worst case (denoted as old corrosion equations) and for the corrosion correlations in Table 2.1 (denoted as new corrosion equations). The difference is seen to be significant, particularly for low h . Until more corrosion data is available, Fig. 3.6 can be used to give the range of expected lifetime for a particular initial thickness, δ , and surface coefficient, h .

There is some question as to the effect of subcooled boiling on the corrosion rate. At present, it is felt⁽³⁾ that, if there is no depositing of fission products, the boiling may lower the corrosion rate, and certainly should not increase it. If there is depositing of fission products, the increased local power densities may increase the corrosion rate. If it is necessary to avoid boiling on the tank surfaces, Eqs. (3.11) (3.12) and Fig. 3.4 can be used to determine what conditions must be met.

IV. DOUBLE WALL CORE TANK

In a double wall core tank, several methods of cooling are possible. The most attractive method is to use part of the fuel solution which has been taken off the main stream and passed through a heat exchanger. If this coolant flow were re-introduced to the main fuel flow in the core, only a heat exchanger and pump would be required for this system. Operation could be continued after holes in the liner developed, and even until the entire liner had corroded away. Unfortunately, the corrosion of stainless steel by UO_2SO_4 solution sets a lower limit on the coolant temperature if stainless steel equipment is to be used. A change in the protective oxide coating occurs about $225 - 250^\circ\text{C}$, and below this temperature a non-protective oxide coating permits continued corrosion. Therefore, it is highly desirable to keep the coolant temperature above this changeover temperature. The minimum allowable coolant temperature was chosen as 240°C .

Other possible cooling methods include:

i) Use of fuel solution at a temperature below 240°C . This would require the use of a titanium heat exchanger in the coolant loop.

ii) Use of heavy water as coolant. This would avoid the stainless steel corrosion problem, reduce corrosion of the Zircaloy-2 in the coolant channel, and greatly decrease the heat generation in the coolant. It would not affect the nuclear characteristics of the reactor appreciably. The disadvantages are that a separate system would be required for the coolant, and that operation would have to be stopped when the first sizeable hole in the liner developed.

iii) Use of a porous liner through which clean D_2O would be forced to lower the UO_2SO_4 concentration at the liner surface and thereby reduce corrosion. This would require distilling the excess D_2O from the fuel solution somewhere in the system. Further information on the effect of UO_2SO_4 concentration on Zircaloy-2 corrosion is necessary before this system could be seriously considered.

fig 3.6

SINGLE CORE TANK

LIFETIME VS. INITIAL
THICKNESS

SHOWING EFFECT OF NEW CORROSION
EQNS FOR CASE OF CORE REPLACEMENT
AT $\frac{1}{4}$ " THICKNESS.

a) Fuel Solution at 240°C as Coolant

Considering first the use of fuel solution at 240°C as the coolant, expected lifetimes for the double wall tank are calculated in much the same way as for single wall core tanks. Since the corrosion rate on a non-cooled surface can be considerably higher than on a cooled surface, it is advantageous to design all of the tank life into the liner. Therefore, the lifetime is taken as the time for the liner to corrode entirely through. Considering the liner corrosion, the problem is similar to the single tank problem except that corrosion occurs on both surfaces. In this case, instead of specifying the channel surface coefficient, the surface temperature, t_2 , is considered fixed. This allows the coolant channel to be considered as a separate problem. Fig. 4.1 shows the infinite flat plate system considered, which is the same as in Fig. 3.1 except that t_2 is fixed. The heat generation in the laminar film is again taken into account in the equivalent heat transfer region of thickness $L =$

Fig. 4.1

The lifetime of the liner is calculated as follows. If x is the liner thickness at time γ :

$$\frac{dx}{d\gamma} = -CR_1 - CR_2 \quad (4.1)$$

where CR_1 is the corrosion rate on the inner surface and CR_2 is the corrosion rate on the coolant channel surface. Since t_2 is taken as constant, CR_2 may be found directly from Fig. 2.2 as in Eq. (2.5).

$$CR_1 = A e^{Bt_1} \quad (4.2)$$

where A and B are given in Table 2.1. In some cases of interest, it will be necessary to approximate the variation of t_1 with thickness x by a parabolic relation.

$$t_1 = -Fx^2 + Dx + E \quad (4.3)$$

The differential equation which must be solved to find the lifetime is then:

$$\frac{dx}{d\gamma} = -Ae^{-BFx^2 + BDx + BE} - CR_2$$

$$\begin{aligned} x &= \delta & \gamma &= 0 \\ x &= 0 & \gamma &= T \end{aligned} \quad (4.4)$$

To find the surface temperature as a function of wall thickness, it is again necessary to solve for the temperature profile in the wall. Using the notation of Fig. 4.1, the differential equations and boundary conditions are:

$$\frac{d^2 t_a}{dx^2} + \frac{q'''}{k_w} = 0; \quad -\delta \leq x \leq 0 \quad (4.5)$$

$$\frac{d^2 t_b}{dx^2} + \frac{Q'''}{k} = 0; \quad 0 \leq x \leq L \quad (4.6)$$

$$t_a(0) = t_b(0) = t_1 \quad (4.7)$$

$$q/A|_0 = -k_w \left. \frac{dt_a}{dx} \right|_{x=0} = -k \left. \frac{dt_b}{dx} \right|_{x=0} \quad (4.8)$$

$$t_a(-\delta) = t_2 \quad (4.9)$$

$$t_b(L) = t_m \quad (4.10)$$

The solutions for $t_b(x)$ were obtained to investigate the effect of the volume generation in the equivalent film on the surface temperature, t_1 . This effect was found to be small for $h > 100$ and negligible for $h \geq 1000$.

The solution for the wall surface temperature is

$$t_1 = \frac{t_m}{1 + \frac{k_w}{h\delta}} + \frac{t_2}{1 + \frac{h\delta}{k_w}} + \frac{\frac{q'''\delta^2}{2k_w}}{1 + \frac{h\delta}{k_w}} + \frac{\frac{Q'''k}{2h^2}}{1 + \frac{k_w}{h\delta}} \quad (4.11)$$

Fig. 4.2 shows t_1 vs. δ for $h = 100, 1000, \infty$ and $t_2 = 240^\circ\text{C}, 250^\circ\text{C}$. It is seen that for $240^\circ \leq t_2 \leq 250^\circ\text{C}$ and $h > 100$, $t_1 \leq t_m$ for $\delta \leq 1/2$ -in. This sets a limit on δ of $1/2$ -in. if the temperature t_1 is to be cooled below t_m for these conditions.

fig 4.2

In the lifetime calculations, values for h of 100 and 1000 were selected to indicate the effect of low and high heat transfer coefficient at the inner urface. Two values of t_2 were considered, to investigate the effect of coolant channel wall temperature of t_1 . Using 240°C fuel solution as a coolant, the lowest possible value for t_2 is 240°C (464°F). To achieve this, a very large surface coefficient and high velocities would be required in the coolant channel. A temperature of 250°C (484°F) could be achieved more easily.

For the cases where $h = 100$, a linear approximation is used for t_1 vs. δ . The equations used are:

$$\left. \begin{aligned} t_1 &= 1660 \delta + 464; & t_2 &= 240^\circ\text{C} \\ t_1 &= 1535 \delta + 482; & t_2 &= 250^\circ\text{C} \end{aligned} \right\} \quad 0 \leq \delta \leq 0.0416 \text{ ft} \quad (4.12)$$

For the cases where $h = 1000$, a parabolic approximation was considered sufficient for t_1 vs. δ . The equations used are

$$\left. \begin{aligned} t_1 &= -5.23 \times 10^4 \delta^2 + 3820 \delta + 464; & t_2 &= 240^\circ\text{C} \\ t_1 &= -3.94 \times 10^4 \delta^2 + 2930 \delta + 482; & t_2 &= 250^\circ\text{C} \end{aligned} \right\} \quad (4.13)$$

$$0 \leq \delta \leq 0.0416 \text{ ft.}$$

Equations (4.12) and (4.13) provide the constants D , E and F for Eq. (4.4) for the cases under consideration. The corrosion constants, A and B , are taken from Table 2.1. The channel corrosion rates, from Fig. 2.2, for a channel power density of 20 w/ml are:

$$\begin{aligned} \text{CR}_2 &= 6.34 \times 10^{-4} \text{ ft/yr}; & t_2 &= 240^\circ\text{C} \\ \text{CR}_2 &= 8.57 \times 10^{-4} \text{ ft/yr}; & t_2 &= 250^\circ\text{C} \end{aligned}$$

In the cases where $H = 100$, Eq. (4.4) can be solved explicitly for the lifetime since $F = 0$. The result is:

$$T = \frac{1}{BD \cdot \text{CR}_2} \ln \left(\frac{Ae^{BE} + \text{CR}_2}{Ae^{BE} + \text{CR}_2 e^{-BD\delta}} \right) \quad (4.14)$$

$$h = 100$$

$$0 \leq \delta \leq 0.0416 \text{ ft}$$

For the two values of t_2 considered, this equation becomes

$$T = 57.1 \ln \left(\frac{13.09}{6.75 + 6.34 e^{-27.6\delta}} \right); \quad t_2 = 240^\circ\text{C} \quad (4.15)$$

$$T = 45.9 \ln \left(\frac{17.78}{9.21 + 8.57 e^{-25.5\delta}} \right); \quad t_2 = 250^\circ\text{C} \quad (4.16)$$

For the cases where $h = 1000$, Eq. (4.4) is most easily solved numerically. The problem is written:

$$\left. \begin{aligned} \frac{d\tau}{dx} &= \tau' = \frac{-1}{CR_2 + Ae^{-BFx^2 + BDx + BE}} \\ x &= \delta \quad \tau = 0 \\ x &= 0 \quad \tau = T = ? \end{aligned} \right\} \quad (4.17)$$

Using Taylor expansions of the function $\tau(x)$ at successive points at intervals of length Δ , the approximate form may be derived:

$$\tau_{n+1} = \tau_{n-1} + 2\Delta \tau'_n \quad (4.18)$$

where n refers to the points of approximation. The error in Eq. (4.18) is on the order of Δ^3 . If δ/Δ is chosen equal to N , where N is an even integer, it is sufficient to calculate τ'_n from Eq. (4.17) for n odd, and τ_{n+1} from Eq. (4.18) for n even, to find $\tau_N = T$. A value of $\Delta = 0.0052$ ft was chosen for all calculations.

For the two values of t_2 considered, Eq. (4.17) becomes:

$$\tau' = \frac{-10,000}{6.83 + 6.75e^{-868x^2 + 63.4x}} ; \quad t_2 = 240^\circ\text{C} \quad (4.19)$$

$$\tau' = \frac{-10,000}{8.57 + 9.21e^{-654x^2 + 48.6x}} ; \quad t_2 = 250^\circ\text{C} \quad (4.20)$$

$$h = 1000 \quad 0 \leq x \leq 0.0416 \text{ ft.}$$

The calculated lifetimes are plotted against initial liner thickness, for the four cases considered, in Fig. 4.3.

If a coolant channel wall temperature of 240°C or 250°C can be achieved, Fig. 4.3 can be used to estimate the liner lifetime. The two factors which determine whether a particular coolant wall temperature can be obtained are:

- 1) Wall heat flux which coolant must remove.
- 2) Temperature rise of coolant due to wall heat flux and internal generation.

fig 4.3

DOUBLE WALL CORE TANK

The wall heat flux to the coolant is calculated from the relation:
(notation of Fig. 4.1)

$$(q/A)_c = k_w \left. \frac{dt_a}{dx} \right|_{x = -\delta} \quad (4.21)$$

Using the solution of Eqs. (4.5) - (4.10), this becomes:

$$(q/A)_c = q''' \delta + \frac{\frac{Q''' k}{2h^2} + (t_m - t_2) - \frac{q''' \delta^2}{2k_w}}{\frac{1}{h} + \frac{\delta}{k_w}} \quad (4.22)$$

The required heat fluxes to the coolant for $t_2 = 240^\circ\text{C}$ and 250°C and $h = 100$ and 1000 are plotted against liner thicknesses in Fig. 4.4.

This plot illustrates clearly that cooling the tank wall when there is a high surface coefficient, h , would probably require the removal of excessive amounts of heat from the core fluid. With a low h , the wall is sufficiently insulated from the core fluid to allow cooling with more reasonable heat removal requirements.

As an example of the heat transfer requirements in the coolant channel, assume $(q/A)_c = 23,000$ Btu/hr-ft². The channel surface coefficients, h_c , required for several values of channel surface temperature, t_2 , minus coolant temperature, t_c , are given in Table 4.1. This shows the desirability of a very high h_c .

TABLE 4.1

$\frac{(t_2 - t_c)}{(^{\circ}\text{F})}$	$\frac{h_c}{(\text{Btu/hr-ft}^2\text{-}^{\circ}\text{F})}$
1	23,000
9	2,600
18	1,300

To estimate the coolant temperature rise, the liner wall heat flux and the tank wall heat flux must be added to the volume generation rate in the coolant. The tank wall heat flux can be calculated from Eq. (4.22), using h_p for h , the tank wall thickness for δ , the slurry thermal conductivity for k , and the blanket wall power density for Q''' . For a 3/8-in. tank wall, and a range of 80 - 550, the heat flux to the coolant is between 21,000 - 23,000 Btu/hr-ft² for $t_2 = 240^\circ\text{C}$.

Next considering the heat generation in the coolant fluid, the volume of coolant is approximately:

$$V_c \cong 4\pi(2\text{ ft})^2 d(\text{ft}) = 50 d \text{ ft}^3$$

fig 4.4

where d is the annular spacing in feet. The exact situation at the poles is not considered. For a power density of 20 w/ml in the coolant, the total heat generation rate is:

$$Q_v = 9.65 \times 10^n d \text{ (Btu/hr)}$$

Since the coolant channel wall areas are both approximately $4 (2 \text{ ft})^2$ or 50 ft^2 , the heat fluxes at the walls add a heat input to the coolant of

$$\begin{aligned} Q_w &= 50 \text{ ft}^2 \left[23,000 \frac{\text{Btu}}{\text{hr-ft}^2} + 22,000 \frac{\text{Btu}}{\text{hr-ft}^2} \right] \\ &= 2.25 \times 10^6 \text{ Btu/hr} \end{aligned}$$

The total heat, Q , which the coolant must remove per hour for several coolant channel widths is tabulated in Table 4.2. The required flow rate of coolant for a particular average temperature is calculated from:

$$w \text{ (lb/hr)} = \frac{Q \text{ (Btu/hr)}}{C_p \Delta t \text{ (Btu/lb)}}$$

TABLE 4.2

d (in)	Q_w Btu/hr	Q_v Btu/hr	Q Btu/hr	Q Mw
1/4	2.25×10^6	2.01×10^6	4.26×10^6	1.25
1/2	"	4.02×10^6	6.27×10^6	1.84
1	"	8.04×10^6	1.03×10^7	3.02

Table 4.3 summarizes calculations of required flow rates for several cases of interest. In addition, the percentage of total fuel flow ($2.547 \times 10^6 \text{ lb/hr}$) is tabulated.

TABLE 4.3

<u>d</u> <u>(in)</u>	<u>Δt</u> <u>°F</u>	<u>w</u> <u>lb/hr</u>	<u>% of total</u> <u>fuel flow</u>
1/4	1	3.44×10^6	135
	9	3.82×10^5	15
	18	1.91×10^5	7.5
1/2	1	5.05×10^6	199
	9	5.61×10^5	22
	18	2.81×10^5	11
1	1	8.30×10^6	326
	9	9.23×10^5	36
	18	4.61×10^5	18

From this example, it is evident that cooling by this method is not simple. The most reasonable case considered in Table 4.3 requires a 1/4-in. channel and allows a temperature rise of 10C° in the coolant. Fig. 4.3 illustrates the importance of keeping t_2 as near as possible to the inlet coolant temperature of 240°C. When the two factors presented in Tables 4.1 and 4.3 are considered together, the feasibility of achieving a coolant channel wall temperature as low as 250°C is seen to be small.

In column 1 of Table 4.4, the cooling system parameters are summarized for the most likely case in Table 4.3, for straight through coolant flow. The pressure drop calculation is given in Appendix III. The surface coefficient, h_c , at the equator, was calculated from the velocity at the equator using the Colburn correlation for turbulent flow, with $2d$ used for the equivalent diameter.

$$\frac{h_c(2d)}{k} = 0.023 (N_{Re})^{.8} (N_{Pr})^{.4} \quad (4.23)$$

The N_p was taken as 1 (see Appendix II). A liner wall heat flux of 23,000 Btu/hr-ft² was used.

TABLE 4.4

	<u>240°C Fuel Solution</u>	<u>217°C D₂O</u>
Channel width	1/4-in.	1/4-in.
Flow path	Straight through	Straight through
Coolant path length	6 ft.	6 ft.
Coolant flow rate	1.91×10^5 lb/hr	2.83×10^5 lb/hr
% of total fuel flow rate	7.5 %	---
Coolant inlet temperature	240°C	217°C
Coolant outlet temperature	250°C	222°C
Pressure drop across core	~ 6 psi*	~ 11 psi
Coolant velocity (at equator)	3.8 fps	5.2 fps
h_c (at equator)	2100	2720
$t_2 - t_c$ (at equator)	6°C	5.5°C

It would be possible to obtain higher velocities and thus larger h_c by using a helical flow path. This would result in a decrease in $(t_2 - t_c)$, and an increase in the pressure drop. The most that could be achieved with a swirling flow is to make $(t_2 - t_c)$ approximately zero. Since the coolant temperature rise of 10°C would still remain, the situation would not be much better.

This example shows that it may be possible to use fuel solution at 240°C as a coolant in a 1/4-in. channel between liner and core tank. The comparison of calculations for the single and double wall tanks shown in Fig. 4.5 indicates that a lengthening of tank lifetime would only be obtained if there is a relatively low core side surface coefficient. In Fig. 4.5, the case of a single core tank with replacement after corrosion to 1/3-in. (from Fig. 3.5) is compared with a double wall tank with a 3/8-in. thick main tank (from Fig. 4.3). Permitting the single core tank to corrode until rupture occurs, may increase its lifetime by several years (see Fig. 3.5). However, permitting the main tank in the double wall case to corrode till rupture increases its lifetime equivalently. Four things must be remembered in conjunction with Fig. 4.5. First, it may be very hard to achieve a $t_2 = 250^\circ\text{C}$. Second, the corrosion correlations for the single tank case are in some doubt and the lifetimes for this case may be significantly smaller (see e.g. Fig. 3.6). Third, boiling on the core and slurry surfaces has been considered as allowable in single core tank calculations. Fourth, the calculations have assumed a uniform h over the spherical surface. Appendix II indicates that this may be very incorrect.

COMPARISON OF SINGLE AND DOUBLE WALL CORE TANKS.

fig 4.5

LIFETIME VS. TOTAL
INITIAL THICKNESS

The conclusion reached from these calculations is that the use of 240°C fuel solution as coolant will not give greatly increased lifetimes over equivalent single wall tanks for expected HRE-3 conditions. There seem to be two conditions when this cooling system might offer important advantages. First, if the core side surface coefficient were low over the entire inner tank surface so that wall cooling would not require removal of excessive amounts of heat from the core. Secondly, if boiling had to be avoided on the inner tank surface. In a full scale power reactor, the high power density in the wall may well require a double wall cooling system just to maintain the inner surface temperature below the saturation temperature.

b) D₂O at 225°C as Coolant

While the use of 240°C fuel solution as coolant does not appear very advantageous, many of the difficulties can be overcome by using either fuel solution below 240°C or D₂O as coolant. Of these two, D₂O seems to have a definite advantage since the corrosion in the coolant channel is reduced to a negligible amount and the heat generation in the coolant is also greatly reduced. In addition, the stainless steel corrosion problem which would be present with UO₂SO₄ solution below 240°C is avoided. The need for a separate system for the D₂O coolant may not be too severe a disadvantage. Since it is a separate system, there is no limitation on the possible coolant flow rate. The heat removed by the coolant could be used elsewhere in the system - perhaps in the boiler feed water heaters. The calculations for a D₂O cooling system indicate it could give long lifetimes under the worst assumptions.

In making calculations on such a D₂O cooled system, Fig. 4.1 and the following development are applicable.

The only change is in the value of t_2 . As an example, t_2 is chosen as 225°C. In Fig. 4.6, the solutions for t_1 from Eq. (4.11) are plotted for liner thicknesses up to 1/2-in. and for $h = 100$ and 1000. The approximations used for the lifetime calculations are:

$$t_1 = 1920 \delta + 437; \quad h = 100 \quad (4.25)$$

$$t_1 = -69,500 \delta^2 + 5100 \delta + 437, \quad h = 1000 \quad (4.26)$$

Since the channel corrosion is assumed negligible, the differential equation is:

$$\left. \begin{aligned} \frac{dx}{d\tau} &= -CR_1 = -Ae^{Bt_1} \\ x &= \delta \quad \tau = 0 \\ x &= 0 \quad \tau = T \end{aligned} \right\} \quad (4.27)$$

Although the wall temperatures go below 250°C, the lowest temperature for which corrosion data is available, it is expected that the corrosion correlation in Table 2.1 for this region is good. The solution of Eq. (4.27) for $h = 100$ is

$$T = 72.5 (1 - e^{-31.9\delta}) \quad (4.28)$$

The solution of Eq. (4.27) for $h = 1000$ is best done numerically as was done for Eq. (4.17). For this purpose, Eq. (4.27) is written in the form:

$$\frac{d\gamma}{dx} = \frac{-2330}{e^{-1150x^2 + 84.6x}} \quad (4.29)$$

The calculated lifetimes for these two cases are also plotted in Fig. 4.6. The lifetimes shown there cannot be obtained, since the liner cannot be allowed to corrode below a minimum thickness necessary to keep it from rupturing.

There is also the problem of the heat flux required to maintain t_2 at 225°C in regions of high h . Equation (4.22) may be used to calculate this heat flux with 437°F (225°C) used for t_2 . The liner wall heat flux to the coolant is also plotted on Fig. 4.6.

Although the temperature difference between the channel wall and the coolant is not an important problem, since the coolant temperature can be adjusted, it is desirable to keep the coolant temperature rise quite small, so as to keep the wall temperatures as uniform as possible. The power density in the D_2O is due to the gamma flux from the core and is estimated to be no more than 10 per cent of the power density at the inner liner surface⁽¹⁾. Therefore, the volume generation of heat in the coolant is small compared to the heat added through the walls. The tank wall heat flux can be calculated from Eq. (4.22) as was done before. For a $3/8$ -in. tank wall and a range of h_b of 80-550, the heat flux to the coolant is between 22,000 and 28,000 Btu/hr-ft². The heat added to the coolant through the walls (for $1/2$ -in. thick liner) is then

$$\begin{aligned} Q_w &\cong 50 \text{ ft}^2 \left[27,000 \frac{\text{Btu}}{\text{hr-ft}^2} + 25,000 \frac{\text{Btu}}{\text{hr-ft}^2} \right] \\ &= 2.6 \times 10^6 \text{ (Btu/hr)} \end{aligned}$$

Using 2 kw/l for the power density in the coolant, the total heat generation rate is

$$Q_v = 9.65 \times 10^6 d \text{ (Btu/hr)}$$

where d is the channel width in feet. The equations relating the parameters of this cooling system for straight through coolant flow are:

$$Q(\text{Btu/hr}) = Q_w + 9.65 \times 10^6 d \quad (4.30)$$

$$w(\text{lb/hr}) = \frac{Q}{C_p \Delta t} \quad (4.31)$$

$$\frac{h_c(2d)}{k_c} = 0.023 \left(\frac{w}{2\pi \mu c} \right)^{.8} \left(\frac{C_p \mu}{k} \right)_c^{.3} \quad (4.32)$$

$$v(\text{ft/hr}) = \frac{w}{4\pi \rho c d} \quad (4.33)$$

where the velocity of the coolant at the equator is considered, since it will indicate the lowest h_c . The values used for the coolant are: (6)

$$\begin{aligned} C_p &= 1.1 \text{ Btu/lb-}^\circ\text{F} \\ \rho_c &= 58 \text{ lb/ft}^3 \\ k_c &= 0.38 \text{ Btu/ft-}^\circ\text{F-hr} \\ \mu_c &= 0.31 \text{ lb/hr-ft} \end{aligned}$$

Combining Eqs. (4.30) - (4.32) and substituting the values for the coolant properties, the following relation can be derived.

$$h_c d = 312 \left(\frac{1 + 3.7d}{\Delta t} \right)^{.8} \quad (4.34)$$

It is seen that small Δt , small d , and large h_c are mutually compatible. However, Δt cannot be chosen too small or w will become excessive. A thorough analysis would provide the best values for these parameters, but for this example Δt is chosen as 9°F (5°C) and d as $1/4$ -in. For these values, the system parameters are summarized in column 2 of Table 4.4.

This system is seen to be quite satisfactory. As the tank corrodes, the increasing wall heat flux in regions of high h may cause difficulties. To investigate this, using Fig. 4.6, the average q/A_c through the liner for a $1/8$ -in. thickness is estimated at $40,000 \text{ Btu/hr-ft}^2$. If the flow rate remained $2.83 \times 10^5 \text{ lb/hr}$ and the coolant inlet temperature was kept at 217°C , the outlet temperature would rise to 223°C and $t_2 - t_c$ at the equator would be less than 8°C . This example indicates that the increase in coolant temperature rise as the liner corrodes will not be a problem of liner thicknesses of $1/8$ -in. or more.

It does not seem to be possible at the present time to estimate the liner thickness required to prevent rupture in this case. This is partly because the design pressure difference between these regions has not been set, and partly because the effect of thermal stresses on collapse of the tank is not known. As an optimistic guess, corrosion to $1/8$ -in. may be allowable. In this case, the lifetime of a $1/2$ -in. liner could be expected to be 19 years, even with regions of h around 1000.

This means of cooling the core tank shows considerable promise in cases where long lifetimes are desired. The D_2O temperature could be further reduced to increase the lifetimes, until thermal stresses in the tank and liner walls became excessive. There is a need for thorough stress calculations for such systems to be performed before such lifetime calculations can be continued.

The use of a double wall system with D_2O coolant may also prove to be a possible way of achieving reasonable lifetimes in large reactors with high heat generation in the core tank walls.

V. FURTHER SINGLE WALL CORE TANK CONSIDERATIONS

a) Economics of Core Tank Thickness

The core tank lifetime can be increased by increasing the thickness, but there is an accompanying decrease in the breeding ratio. Although the primary goal for the HRE-3 is to achieve breeding, it is of interest to estimate the optimum tank thickness from the standpoint of power costs. To do this, the effect of thickness on fuel burnup costs, core tank costs, replacement costs, and investment costs must be compared. In such a comparison, only those parts of the power cost which vary with tank thickness are of interest.

The fuel burnup costs for the reactor may be written as:

$$C_f = \frac{\text{fuel cost}(\$/\text{gm}) \times \text{fuel consumption}(\frac{\text{gm}}{\text{hr} + \text{MWday}})}{(24 \frac{\text{hr}}{\text{day}}) (\text{Plant efficiency})} (1 - \text{BR}) \quad (5.1)$$

where C_f = fuel burnup costs (mills/kw-hr)

BR = Breeding ratio $(\frac{\text{gms fuel produced}}{\text{gms fuel consumed}})$

Using present fuel costs, normal fuel consumption and efficiency, this is found to be⁽¹⁾:

$$C_f = 3.2 (1 - \text{BR}) \quad (5.2)$$

The effect of core thickness on breeding ratio for the HRE-3 has been reported⁽⁸⁾ as

$$\frac{\partial \text{BR}}{\partial \delta} = -0.048 (\text{in})^{-1} \quad (5.3)$$

and using the value given for the breeding ratio for a 1/3-in. thick tank, it is found that:

$$\text{BR} = 1.139 - 0.048 \delta (\text{in}) \quad (5.4)$$

so that

Since only the variable part of this cost is of interest in determining minimum power costs, it is convenient to write:

$$C_f, \delta = 0.154 \delta \quad (5.6)$$

where C_f, δ = part of fuel burnup cost dependent on core tank thickness (mills/kwhr).

Next, estimating the core tank costs,

$$C_t = \frac{\text{material cost (\$/lb)} \times \text{tank wt (lb)}}{\text{lifetime (yr)} \times 8760 \left(\frac{\text{hr}}{\text{yr}}\right) \times (15 \text{ el MW}) \times \text{PF}} \quad (5.7)$$

C_t = reactor core tank cost (mills/kwh)

PF = plant factor = percentage of time plant is in operation

$$= \frac{\text{Downtime}}{\text{Period}} = \frac{\text{Lifetime}}{\text{Lifetime} + \text{Downtime}} = \frac{T}{T + D}$$

D = Downtime (yrs)

The material cost for Zircaloy-2 is taken as \$150/lb installed⁽⁹⁾. The tank weight is approximated by considering a full sphere:

$$\text{Tank Weight} = 4 \pi (2 \text{ ft})^2 \times \left(\frac{\delta}{12} \text{ ft}\right) \times 400 \text{ lb/ft}^3$$

Substituting these values in Eq. (5.7), the core tank cost becomes:

$$C_t = 1.9 \frac{T+D}{T^2} \quad (5.8)$$

The lifetime may be approximated by the equation:

$$T = a (\delta - x_f) \quad (5.9)$$

Where a is found from Fig. (3.5) or (3.6) for a particular x_f and h . The tank cost can now be written:

$$C_t = C_{t, \delta} = 1.9 \frac{a(\delta - x_f) + D}{a^2(\delta - x_f)^2} \quad (5.10)$$

The investment costs of the reactor may be written as

$$C_{RA} = \frac{I(\$/\text{el kw}) \times H(\$/\text{yr}-\$) \times 10^3 \left(\frac{\text{kw}}{\text{MW}}\right)}{8760 \left(\frac{\text{hr}}{\text{yr}}\right) \times \text{PF}} \quad (5.11)$$

C_{RA} = investment part of power cost (mills/kw hr)

I = investment cost of reactor

H = fixed charges (taken as 15% per year)

PF = plant factor

The investment costs of the HRE-3, for 15 mw electrical output, are expected to be about \$1000/installed el. kw with present technology, and about \$300/el-kw installed with advanced technology. Then:

$$\left. \begin{aligned} C_{RA} &= 17.1 \left(1 + \frac{D}{T}\right), \text{ (present technology)} \\ C_{RA} &= 6.0 \left(1 + \frac{D}{T}\right), \text{ (advanced technology)} \end{aligned} \right\} \quad (5.12)$$

The variable part of this cost may be written, using Eq. (5.9) as:

$$C_{RA, \delta} = \left. \begin{aligned} &\frac{17.1 D}{a(\delta - x_f)} \quad \text{(present technology)} \\ &\frac{6.0 D}{a(\delta - x_f)} \quad \text{(advanced technology)} \end{aligned} \right\} \quad (5.13)$$

The maintenance and replacement costs are affected by the tank lifetime, and thus by the tank thickness, but it is not presently known how to estimate this effect.

Combining the parts of the total power cost which are affected by tank thickness, an equation is obtained which can be plotted to find the optimum tank thickness.

$$\begin{aligned} C_{p, \delta} &= C_{f, \delta} + C_{t, \delta} + C_{RA, \delta} \\ &= 0.154 \delta + 1.9 \frac{a(\delta - x_f) + D}{a^2(\delta - x_f)^2} + \frac{bD}{a(\delta - x_f)} \end{aligned} \quad (5.14)$$

where $b = 17.1$ for present technology and 6.0 for advanced technology.

In Fig. 5.1, Eq. (5.14) is plotted for the particular case:

$$x_f = 1/4\text{-in.}$$

$$b = 17.1 \text{ (present technology)}$$

$$a = 22.3; \quad h = 100$$

$$a = 30.7; \quad h = 1000$$

$$D = 1 \text{ yr}$$

In estimating a value for the downtime, it is assumed that the major part of the downtime will occur during core tank replacement. The rather pessimistic value of 1 year has been chosen.

The lifetime of the core tank, from Fig. 3.6, and the breeding ratio, from Eq. (5.4), have also been plotted on Fig. 5.1. When the information used in Fig. 5.1 is more accurately known, a plot of this sort can be used to choose the tank thickness. A core tank thickness of about 1-in. seems to have definite advantages on the basis of present information. Significant decreases in the factors b , x_f , or D , or increases in a , in Eq. (5.14) may alter this conclusion.

b) Cooling with Slurry

While Fig. 5.1 strongly suggests using core tanks with thicknesses of 1-in. or more, the occurrence of sub-cooled boiling at regions of low h (see Fig. 3.4) may limit the possible thickness if boiling is to be avoided. Therefore, a means of keeping wall surface temperatures below saturation for thick walled tanks is highly desirable. The use of slurry as coolant may provide such a system.

In investigating a slurry cooling system, the optimistic assumption of a Newtonian slurry is made. The system considered is shown in Fig. 5.2, where t_s is the slurry temperature. The equations governing the system are the same as Eqs. (3.5) - (3.10) except for:

$$q/A)_{-\delta} = h_b(t_s - t_2) = -k_w \frac{dt_a}{dx} \Big|_{x=-\delta} \quad (5.15)$$

Fig. 5.2

The solution of these equations for temperature t_1 gives:

$$t_1 = t_s + U \left[\frac{q''' \delta}{h} \left(\frac{\delta}{2k_w} + \frac{1}{h_b} \right) + \left(\frac{Q''' k}{2h^2} + t_m - t_s \right) \left(\frac{\delta}{k_w} + \frac{1}{h_b} \right) \right] \quad (5.16)$$

$$\text{where } U = \frac{1}{\frac{1}{h} + \frac{1}{h_b} + \frac{\delta}{k_w}}$$

The solution for the heat flux to the slurry, $q/A)_{-\delta}$ is found to be:

$$q/A)_{-\delta} = U \left[\frac{Q''' k}{2h^2} + q''' \delta \left(\frac{1}{h} + \frac{\delta}{2k_w} \right) + (t_m - t_s) \right] \quad (5.17)$$

The temperature of the slurry, t_s , and the slurry side surface coefficient, h_p , are the parameters of the cooling system which can be varied. The surface coefficient is fixed by the annular separation of the core tank and the slurry baffle (see Appendix I), for a constant slurry flow rate. Taking as an example, a velocity at the equator of 10 fps, the annular separation at the equator must be 0.83 in. for a flow rate of 3.69×10^6 lb/hr. Using the Colburn equation, and the correction for asymmetrically heated channels⁽¹⁰⁾

$$\frac{hb(2d)}{k_s} = 2/3 (0.023) (N_{Re})_s^{.8} (N_{Pr})_s^{.4} \quad (5.18)$$

where d is the annular separation in feet and the subscript s refers to slurry. Since:

$$(N_{Re})_s = \frac{2\rho vd}{\mu} = \frac{w}{\pi R \mu} \cong 2.4 \times 10^5 \quad (5.19)$$

for HRE-3 conditions,

$$h_p d = 110 \quad (5.20)$$

For an annular separation of 0.83 in., Eq. (5.21) gives $h_p = 1600$.

The solutions for t_1 , from Eq. (5.16) are plotted in Fig. 5.3 for $h_p = 1600$ and $t_s = 280^\circ\text{C}$, 270°C , 26°C and 250°C . Comparing this figure with Fig. 3.4 shows that the slurry cooling significantly increases the wall thickness at which boiling occurs for $h = 100$. The slurry cooling has a small effect on the wall temperature for $h = 1000$. For the $h = 100$ cases, the lifetime was calculated from Eq. (3.4) for tank thicknesses for which boiling does not occur. Values of 1/3-in. and 1/4-in. were used for stopping thicknesses. The constants D and E used are tabulated in Table 5.1.

TABLE 5.1

h	$t_s(^{\circ}\text{C})$	$D (^{\circ}\text{F}/\text{ft})$	$E (^{\circ}\text{F})$	Range
100	280	1560	524	$1/4\text{-in.} \leq \delta \leq 1/2\text{-in.}$
100	270	1680	510	$1/4\text{-in.} \leq \delta \leq 5/8\text{-in.}$
100	250	1890	478	$1/4\text{-in.} \leq \delta \leq 3/4\text{-in.}$

The calculated lifetimes are given in Table 5.2. By using 250°C slurry and increasing the tank thickness to 3/4-in., it is possible to get lifetimes over 10 years in regions of low h without boiling. Regions of high h would have longer lifetimes since the surface temperature t_1 would be lower.

TABLE 5.2

$t_s(^{\circ}\text{C})$	$\delta(\text{in})$	$T(\text{yr})$	
		$x_f = 1/3\text{-in.}$	$x_f = 1/4\text{-in.}$
280	1/2	3.8	6.1
	3/8	1.0	3.3
	1/3	0	2.3
	1/4	---	0
270	5/8	6.9	9.5
	1/2	4.2	6.8
	3/8	1.2	3.7
	1/3	0	2.6
	1/4	---	0
250	3/4	10.4	13.8
	5/8	8.7	12.2
	1/2	5.6	9.0
	3/8	1.6	5.0
	1/3	0	3.5
	1/4	---	0

To investigate the heating of the slurry in the slurry baffle, the wall heat flux (Eq. 5.17) and the volume generation must be added. Heat transfer through the baffle wall is neglected. The solutions of Eq. (5.17) are summarized in Table 5.3 for two values of t_s .

TABLE 5.3

$t_s(^{\circ}\text{C})$	h Btu/hr-ft ² °F	$q/A)C, \text{ Av}$ Btu/hr-ft ²	$q/A)C, \text{ max.}$ Btu/hr-ft ²	Range
270	100	18,000	22,500	$1/4\text{-in.} \leq \delta \leq 5/8\text{-in.}$
	1000	13,000	15,000	"
250	100	22,000	27,500	$1/4\text{-in.} \leq \delta \leq 3/4\text{-in.}$
	1000	20,000	21,500	"

The average and maximum wall fluxes are tabulated for the ranges used in the lifetime calculations of Table 5.2. Assuming a complete spherical baffle with annular separation 0.83-in., and a power density of 14 kw-l in the coolant channel, the heat generated is 4.6×10^6 Btu/hr. The maximum possible heat added to the slurry by the wall flux for the cases in Table 5.3 is 1.3×10^6 Btu/hr. Therefore, the slurry temperature rise in the coolant channel is found to be less than 2°C for all cases in Table 5.3. The total temperature rise of the slurry remains fixed at 10°C, but the use of the baffle makes most of this rise occur in the slow downward flow of the slurry. These calculations indicate that the cooling conditions used for the lifetime calculations in Table 5.2 can be readily attained.

The present design data for the HRE-3 gives a slurry inlet temperature of 270°C. If a narrow channel baffle is used, as in the example, the expected tank lifetimes can be estimated from Table 5.2. Thus, a 1/2-in. tank can be expected at last at least four years. With more exact information about h , it is probable that this estimate can be increased at least 50 per cent.

If the temperatures of the slurry cycle are reduced to 250°C inlet temperature and 260°C outlet temperature, the tank thickness could be increased to at least 3/4-in. without surface boiling, and the expected lifetime would be at least 10 years. By further decreasing the slurry temperatures, longer lifetimes could be attained.

This method of increasing the tank lifetime seems to show great promise. If regions of low h do exist, the greater cooling occurring there will tend to make the wall temperature, t_1 , uniform over the sphere and thus tend to eliminate hot regions. By effectively cooling the low h regions, subcooled boiling will be avoided and thicker core tanks can be used. The corrosion rate is not reduced to a negligible amount, but instead to a livable level which can be allowed for in extra tank thickness. The system has the particular advantage of requiring no additional equipment not presently planned for the HRE-3. If a Newtonian slurry is achieved, this method shows the most promise for use on the HRE-3. It may also be suitable for higher power reactors, if heat removal requirements from the tank wall does not become excessive.

VI. DISCUSSIONS AND RECOMMENDATIONS

It is recognized that much of the development in this report is based on questionable grounds. The present state of information is not adequate to allow final conclusions to be drawn. However, it is felt that several of the cooling systems investigated merit further development. Any further investigation should include a determination of reliable corrosion correlation, of core tank strength requirements, and of heat transfer conditions.

The present theory of radiation corrosion of Zr-2 may be essentially correct, but it requires further experimental verification. It is particularly important to obtain corrosion data at temperatures above 280°C and below 250°C. The observation of a maximum in the temperature dependence of the corrosion rate, as in Fig. 2.2, would greatly increase the acceptability of the present corrosion theory.

Thorough stress calculations of cooled and non-cooled core tanks are needed. This would require the establishment of a definite maximum allowable pressure surge between the reactor regions. The effect of thermal stresses, resulting from wall cooling, on collapse of the tank must be determined. Hopefully, the minimum core tank thickness necessary to prevent failure can be determined within a narrow range.

Heat transfer conditions in the core present a difficult problem. In this report, natural convection effects were neglected, and volume generation in the fluid was included in an equivalent heat transfer region. It was hoped that in this way the heat transfer properties of the system could be characterized by a measurable quantity, h . Attempts at more rigorous analysis of core conditions(11) have indicated that natural convection effects in conjunction with volume generation may alter the system considerably.

If natural convection is not important, measurement of the surface coefficient, h , with no volume heat source, in a core mockup will provide the information necessary to design a cooling system.

If h is greater than 1000 over the entire tank surface, the temperature t_1 will remain close to the mean fluid temperature and cooling is not advisable. (Calculations by Oliver(11) for the HRT indicate this may be the case for the HRE-3) The lifetime can be increased by increasing the tank thickness without danger of boiling on the surface

If h is relatively low in some regions of the tank surface, then cooling is possible and desirable. For the HRE-3, slurry cooling is recommended as being simple and effective. If a lifetime of more than five years is not required, the present slurry design temperature range, using a narrow coolant channel, will be adequate. For longer lifetimes, the slurry operating temperatures can be lowered.

For the design of a high power homogeneous reactor having a large power density in the tank wall, a double wall D₂O cooled system may prove desirable. This will require experimental work on flow and heat transfer in the spherical annulus and development of the necessary cooling system components.

VII. APPENDICESAPPENDIX IEstimate of Slurry-Side Surface Coefficient

In order to avoid caking of the slurry on the core vessel, relatively high velocities near the core wall are desirable. In the present HRE-3 design, the slurry flows in a 3-in. spherical annulus between the core wall and a spherical slurry baffle. For the flow rate of 3.69×10^6 lb/hr⁽¹²⁾, the mean slurry velocity at the equator in the annulus is 2.5 fps.

The assumed properties of the slurry at operating conditions are:⁽¹²⁾

$$\rho = 119 \text{ lb/ft}^3$$

$$C_p = 0.515 \text{ Btu/lb-}^\circ\text{F}$$

$$k = 0.444 \text{ Btu/hr-ft-}^\circ\text{F}$$

$$\eta = 1 \text{ cp} = 2.4 \text{ lb/hr-ft}^{(13)}$$

$$\tau_y \sim 0^{(13)}$$

The Prandtl number and Reynolds number for flow at the equator are found to be:

$$N_{PR} = 2.8$$

$$N_{RE} = 2.2 \times 10^5$$

The flow in the annulus is turbulent.

In estimating the heat transfer coefficients, it is assumed⁽¹³⁾ that an upper limit is given by the turbulent flow forced convection surface coefficient which would occur for a Newtonian fluid having the same properties as the slurry. A lower limit is found by assuming laminar flow of a Newtonian fluid of the same properties.

The Martinelli forced heat transfer solution can be used to obtain the Nusselt number for circular pipe flow for the Reynolds and Prandtl numbers calculated above. The result is:⁽¹⁴⁾

$$N_{NU} \cong 650$$

Alternatively, the Colburn equation, Eq. (4.23) can be used to estimate the Nusselt number. This gives:

$$N_{NU} \cong 610$$

Harrison and Menke⁽¹⁰⁾ have shown that the Nusselt number for an asymmetrically heated parallel plate channel, for the Reynolds and Prandtl numbers of this case, is about two-thirds that for symmetrical heating in circular pipes. However, to estimate an upper limit, the maximum N_{NU} was assumed to be about 630, corresponding to an h_p of 560. For calculations, the upper limit on h_p was taken as 550.

It is expected that the lower limit on h_b estimated on the basis of assumed laminar flow will be considerably lower than the actual coefficient. Therefore, the roughness of the approximations is not too important. First using the Colburn form for laminar flow in pipes,

$$N_{NU} = 1.86 (N_{Re})^{1/3} (N_{Pr})^{1/3} \quad (I.1)$$

a N_{NU} of 80 is obtained.

Using the equation for laminar flow of a Bingham plastic⁽¹⁵⁾

$$N_{NU} = 1.615 \left(\frac{WC_p}{kL} \right)^{1/3} \left(1 + \frac{gc\tau_y D}{24\eta v} \right)^{1/3} \quad (I.2)$$

A $N_{NU} \cong 120$ is estimated. In (I.2), W is the mass flow rate, and L is the length of the channel.

Averaging these two results, the minimum N_{NU} was assumed to be 100, corresponding to a minimum h_b of about 80.

It is of interest to see if natural convection is of importance. At the equator, a natural convection coefficient can be estimated from the vertical plate equation⁽¹⁶⁾.

$$N_{NU} = 0.59 (X)^{1/4}, \quad 10^4 \leq X \leq 10^9$$

$$\text{where } X = \frac{L^3 \rho^2 g \beta \Delta t}{\mu^2_f} \quad (I.3)$$

A value for β of 1.16×10^{-3} is found from compressed D_2O density data⁽⁶⁾ reduced by the volume fraction of D_2O^* in the slurry (.88). The result is

$$h \cong 0.7 (\Delta t)^{1/4} \quad (I.4)$$

Therefore, for Δt up to $60^\circ F$ (up to saturation temperature), natural convection predicts a much lower surface coefficient.

On the basis of these estimates, it is expected that the slurry-side surface coefficient will lie in the range:

$$80 < h_b < 550.$$

*Using a specific gravity of 10 for ThO_2 .

APPENDIX IIEstimate of Core Side Surface Coefficient with no Volume Heat Source in Core Fluid

With a concentric inlet and outlet as in the HRE-3 core, the flow in the annular region next to the tank wall has been found to be turbulent and well mixed. As an approximation, the flow at the equator of the sphere is considered as fully developed turbulent flow in a cylindrical annulus, and the Colburn equation, (4.23), is used to calculate the Nusselt number.

The properties used for the fuel solution are:⁽¹²⁾

$$C_p = 1.24$$

$$\mu = 0.26 \text{ lb/hr-ft}$$

$$k = 0.32 \text{ Btu/hr-ft}^\circ\text{F}$$

$$\rho = 53 \text{ lb/ft}^3$$

$$w = 2.547 \times 10^6 \text{ lb/hr}$$

At the equator of the sphere, the mean velocity is 4070 ft/hr, assuming the inner diameter of the annulus to be 1 ft. The equivalent diameter of the system, using the definition:

$$D_e \equiv \frac{4(\text{flow area})}{\text{wetted perimeter}}$$

is found to be 3 ft. Calculations of the Prandtl and Reynolds numbers for the system yield:

$$N_{Pr} = 1$$

$$N_{Re} = 2.5 \times 10^6$$

The Nusselt number calculated from Eq. (4.23) is 3010, giving a surface coefficient of 322. Similar calculations at other cross sections of the sphere yield values of h up to 1500. In these calculations the effects of curvature, and of the forced turbulence due to the flow path are not considered. Since in addition to the assumption of pipe flow is not really applicable, the calculations are much in question. However, the value of h is expected to vary with latitude over the sphere surface, and it is important to study the effects of this variation on the tank surface temperature. For lifetime calculation a range of h is chosen:

$$100 \leq h \leq 1000$$

to cover the whole range from low to high surface coefficient. Because the value of h has such an important effect on core lifetime, and on the feasibility of wall cooling, it should be experimentally measured in a core mockup.

APPENDIX III

Calculation of Pressure Drop for Straight Through Flow in a Spherical Annulus

The pressure drop for straight through flow in a small spherical annulus is due to frictional losses and to losses resulting from the changing area of the channel. The former can be estimated quite accurately, while the latter is approximated from empirical data.

To calculate the frictional losses, the pressure gradient is integrated along the flow path, using the relation:

$$\frac{dp_f}{dx} = - 4f \left(\frac{\rho}{4r_h} \right) \left(\frac{v^2}{2g_c} \right) \frac{\text{lb/ft}^2}{\text{ft}} \quad (\text{III.1})$$

The notation used in Eq. (III.1) and Fig. III.1 is:

f = Fanning friction factor

ρ = density of fluid (lb/ft³)

r_h = hydraulic radius = $d/2$ (ft)

d = annular separation (ft)

v = velocity (ft/hr)

$g_c = 4.17 \times 10^8 \frac{\text{lb mass}}{\text{lb force}} \frac{\text{ft}}{\text{hr}^2}$

R = radius of sphere (ft)

x = distance along annulus from north pole (ft)

θ = azimuthal angle

θ_i = azimuthal angle of coolant input

θ_o = azimuthal angle of coolant outlet

w = mass flow rate of coolant (lb/hr)

μ = fluid viscosity (lb/hr-ft)

The friction factor may be related to the velocity in the range of interest by: (17)

$$f = \frac{0.046}{\left(\frac{\rho v r_h}{\mu} \right)^{.2}} \quad (\text{III.2})$$

The velocity and the flow area are related by:

$$w = \rho A v \quad (\text{III.3})$$

The flow area as a function of azimuthal angle is given by:

$$A(\theta) = 4\pi R r_h \sin \theta \quad (\text{III.4})$$

The azimuthal angle is related to the arc length by:

$$\theta = \frac{x}{R} \quad (\text{III.5})$$

Substituting these relations into Eq. (III.1), the following equation may be derived.

$$\frac{dp_f}{d\left(\frac{x}{R}\right)} = \frac{-1.84 \times 10^4}{\rho_{gc} r_h^3} \left(\frac{w}{R^4}\right)^{.2} \frac{1}{\left(\sin \frac{x}{R}\right)^{1.8}} \quad (\text{III.6})$$

Assuming $\theta_i = 10^\circ$ and $\theta_o = 170^\circ$, and using the symmetry of the integrand about $x = 90^\circ$, Eq. (III.6) may be written:

$$\Delta P_f = \int_{P_i}^{P_o} dp_f = -2A \int_{\frac{\pi}{18}}^{\frac{\pi}{2}} \frac{d\left(\frac{x}{R}\right)}{\left[\sin\left(\frac{x}{R}\right)\right]^{1.8}} \quad (\text{III.7})$$

where

$$A = \frac{1.84 \times 10^4}{\rho_{gc} r_h^3} \left(\frac{w}{R^4}\right)^{.2} \quad (\text{III.8})$$

Eq. (III.7) may be solved numerically. It is written in the form:

$$\frac{dp_f}{d\left(x/R\right)} = \frac{-A}{\left(\sin \frac{x}{R}\right)^{1.8}} \quad (\text{III.9})$$

$$P_{n+1} = P_{n-1} + 2\Delta P_n \quad (\text{see Eq. 4.18})$$

$$P_f = P_i, \quad \frac{x}{R} = \frac{\pi}{18}$$

$$P_f = ?, \quad \frac{x}{R} = \frac{\pi}{2}$$

Choosing $\Delta = \pi/18$, and using the method outlined in Section 4, the solution is found to be:

$$\Delta P_f = -8.05A \left(\frac{\text{lb/ft}^2}{ft}\right) \quad (\text{III.10})$$

Using values for the fuel solution given in Appendix II, A becomes:

$$A = 3.66 \times 10^{-15} \left(\frac{w^{1.8}}{r_h^3}\right)$$

For a 1/4-in annular separation and a coolant flow rate of 1.91×10^5 lb/hr, the frictional pressure loss is found to be:

$$\Delta P_f = -0.58 \text{ psi}$$

To estimate the pressure drop due to the changing flow area, the pressure drop is assumed proportional to the initial velocity head.

$$\Delta P_a = k \left(\frac{V_1^2}{2g}\right) \quad (\text{III.11})$$

The values of k assumed are:

$k = 0.5$ for diverging region

$k = 0.2$ for converging region

For the particular cooling case under consideration, the calculation yields:

$$\Delta P_a = 5.2 \text{ psi}$$

Therefore, the total pressure drop is given by:

$$\Delta P = \Delta P_f + \Delta P_a = 0.6 + 5.2$$

$$\cong 6 \text{ psi.}$$

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