A THERMAL-HYDRAULIC ANALYSIS OF THE COOLING SYSTEM FOR THE 500 KW VIRGINIA POLYTECHNIC INSTITUTE AND STATE UNIVERSITY REACTOR

by

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I. INTRODUCTION

i) Reactor Description

The Virginia Polytechnic Institute and State University Nuclear Reactor is an Argonaut reactor designed, built and installed by Advanced Technology Laboratories, a Division of American-Standard and designated by them as the "UTR-10 reactor." The reactor is a 100 KW, H_2O- moderated, heterogeneous reactor with graphite internal and external reflectors. A plan view of the reactor core is shown in Figure 1.

The reactor core consists of a graphite block in which two aluminum tanks containing the fuel are embedded. The internal graphite reflector is about 18 inches wide. Inside the aluminum tanks are twelve fuel elements each of which contains 12 fuel plates (13 plates per element for 500 KW operation). The fuel plates are composed of U-Al alloy and are clad with aluminum. The maximum thermal neutron flux is approximately 10^{12} neutrons cm^{-2} sec^{-1}. Control of the reactor is achieved by positioning the two control rods vertically alongside the core tanks. Two safety rods are provided for rapid shutdowns.

ii) Power Upgrade of the Argonaut Reactor

In order to increase the usefulness of the reactor, it is necessary to enhance the neutron fluxes available in the experimental facilities. One method of attaining this objective is to increase the power level of the reactor.

Plans to increase the maximum power of the reactor to 500 KW have been underway for several years. A study of the "design basis accident" for the higher power operation has been completed[1,2]. The study confirmed the inherent safety of the Argonaut reactor design even under the highly unlikely postulated accident.

A recently completed study[3] has also examined the consequence of an accident comprising a fuel element broken during a reloading operation.

The Scottish Universities Research Reactor (SURR) is an Argonaut reactor located in East Kilbride, Scotland. This reactor was designed by American Standard and is virtually identical to the VPI&SU Research Reactor. The SURR was initially licensed to operate at 100 KW maximum power. In 1969 studies were undertaken to raise the maximum operating power to 300 KW. An extensive analysis of the thermal hyraulics indicated that higher power operation was feasible and the necessary system modifications were made. Since the end of 1970 the SURR has been operating routinely at 300 KW. A description of the analytical and experimental programs supporting 300 KW operation is given in Reference 4.

The UCLA Argonaut reactor has been studied to investigate the feasibility of operation at 500 KW. The study indicates that operation of an Argonaut reactor at 500 KW is feasible provided that certain system modifications are implemented.

Before the reactor power can be increased, it is necessary to insure that certain limiting conditions are not exceeded. From a thermal-hydraulic point of view, there are four such conditions:

(1) bulk or saturated boiling of the coolant must not occur;

(2) subcooled or surface boiling should not occur on any fuel plate;

(3) fuel meltdown should not occur after loss-of-coolant or loss-of-flow accident; and

(4) fuel-plate vibration problems should not be significant.

All the conditions above are dependent on the thermal-hydraulic characteristics of the reactor cooling system. Therefore, it is necessary to analyze the complete reactor cooling system before any predictions can be made regarding the increase in reactor power.

The purpose of this thesis is to present such an analysis and then demonstrate that the limiting conditions will not be exceeded for 500 KW operation provided that certain system modifications are made.

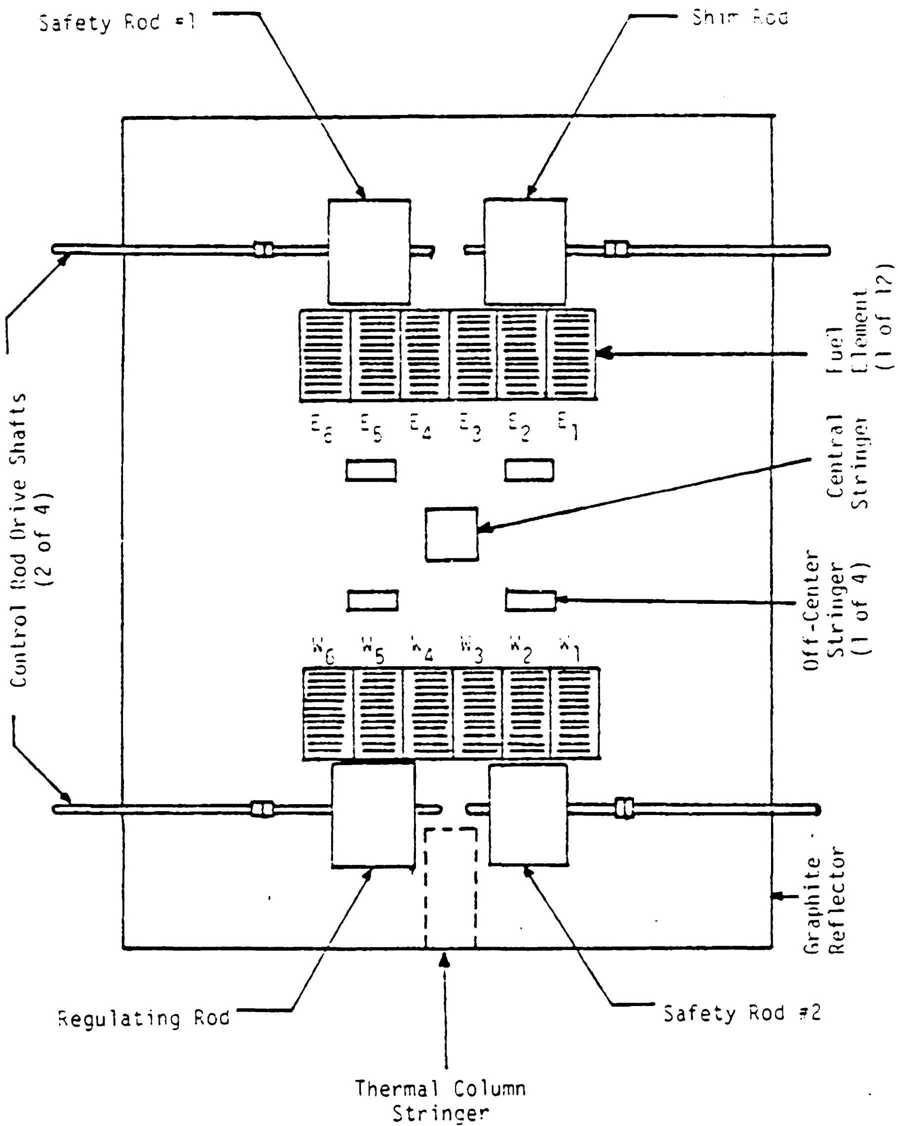


Figure 1. Reactor Core Plan

II. DESCRIPTION OF COOLING SYSTEM

The reactor cooling system installed for 500 KW operation consists of two physically separate systems linked thermodynamically by an intermediate heat exchanger. The circuit that carries the coolant through the reactor core is called the primary system; the other circuit containing the cooling tower is called the secondary system. A schematic diagram of the cooling system is shown in Figure 2.

The coolant (deionized H_2O) in the primary system is pumped through the intermediate heat exchanger in which it transfers heat to the secondary coolant and is thus cooled. The primary coolant then enters the reactor core from below, travels upwards around all the fuel plate surfaces and thus cools them. After trasversing the fuel plates, the coolant flows out of the core through a discharge line; it then travels to the dump tank from which it is pumped for recirculation. If a scram occurs, a dump value connected to the dump tank will open several minutes after reactor shutdown so that the coolant will flow directly into the dump tank.

The secondary coolant is pumped through the intermediate heat exchanger in which it receives heat from the primary coolant and thus its temperature increases. The secondary coolant then passes through the cooling tower, heat being transfered from the coolant to the atmosphere. The secondary coolant is then recirculated.

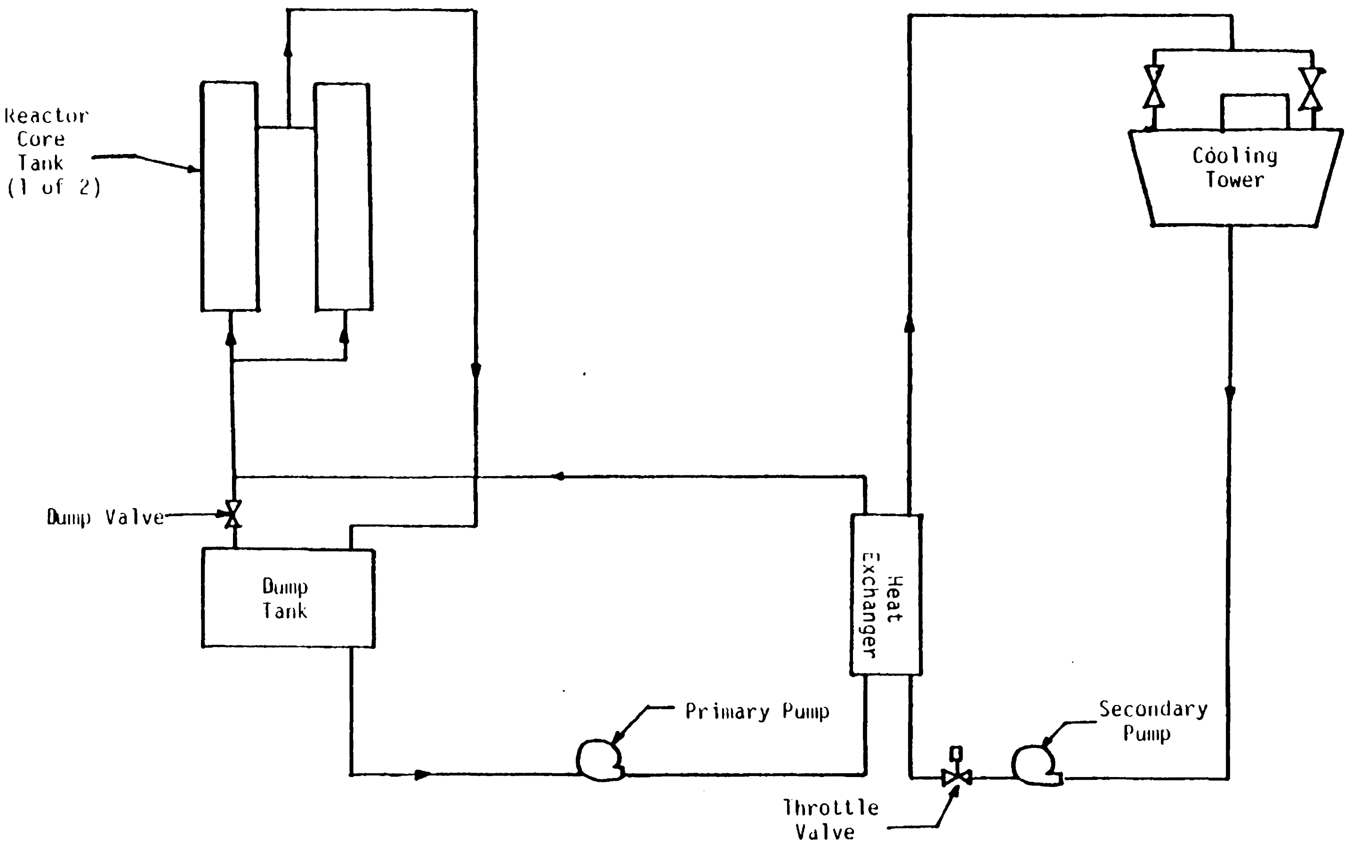


Figure 2. Schematic Diagram of the Cooling System

III. SYSTEM MODIFICATIONS

To obtain reactor operation at a power level of 500 KW., the following modifications must be made on the cooling system [2]:

(1) The primary circuit water flow rate through the core tanks is to be increased from 23 ±3 to 90 ±10 gpm. This will require a larger primary circuit pump together with some larger piping and a wider range flowmeter. It is shown in Appendix A that the existing overflow line will be of sufficient size to take this increase in the primary circuit water flow rate, although a test will be necessary to confirm this result.

(2) A cooling tower has been installed on the roof of Robeson Hall as shown in Figures 3, 4 and 5; thus the modified secondary cooling system will be a closed-loop system. The proposed temperature drop across the cooling tower is $5^{o}F$; this implies that the secondary coolant flow rate will be 685 gpm. The flow rate through the secondary system will be controlled by a throttle valve located near the output side of the pump to allow for operation of the reactor at lower power levels where the rates of heat removal are necessarily smaller.

A make-up water supply will be taken to the cooling tower to allow for water evaporation losses.

(3) The secondary circuit water flow rate will be increased to 685 gpm as previously mentioned. A larger pump is required to attain this flow rate, and this pump has to be sized appropriately. This will entail detailed pressure loss calculations on the secondary system (see Appendix B). Based on this calculation, a suitable pump as specified in Appendix C was selected and installed.

(4) The existing heat exchanger will be removed and a new plate-type heat exchanger will be installed in the northeast corner of the reactor cell.

(5) The dump valve will be operated manually so that the water can be kept in the core tanks after reactor shut down.

(6) In order to accommodate the increased primary coolant flow rate, the piping run from the dump tank to the primary pump will have to be replaced with larger piping.

The specifications of the major components of the new cooling system are summarized in Appendix C.

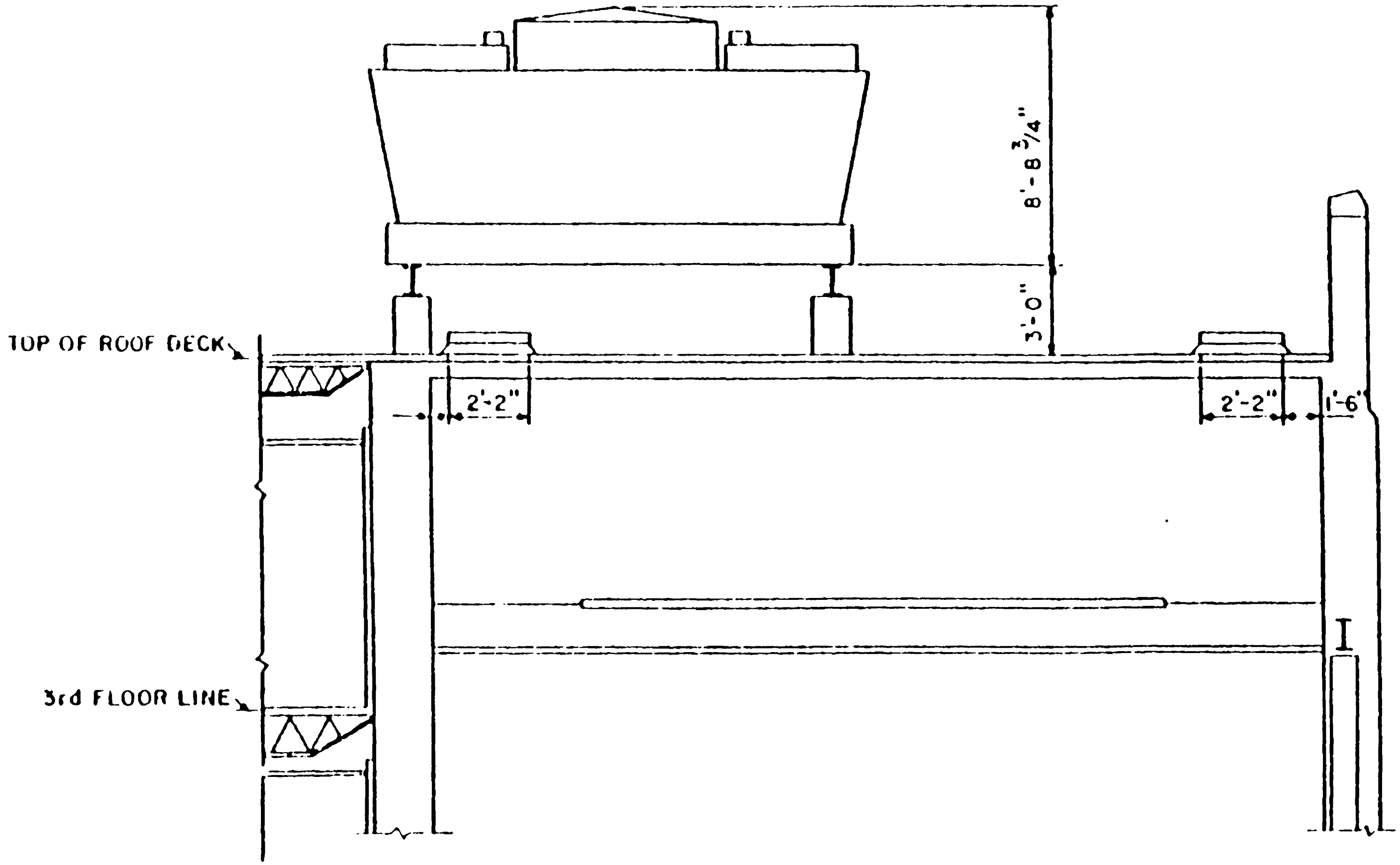
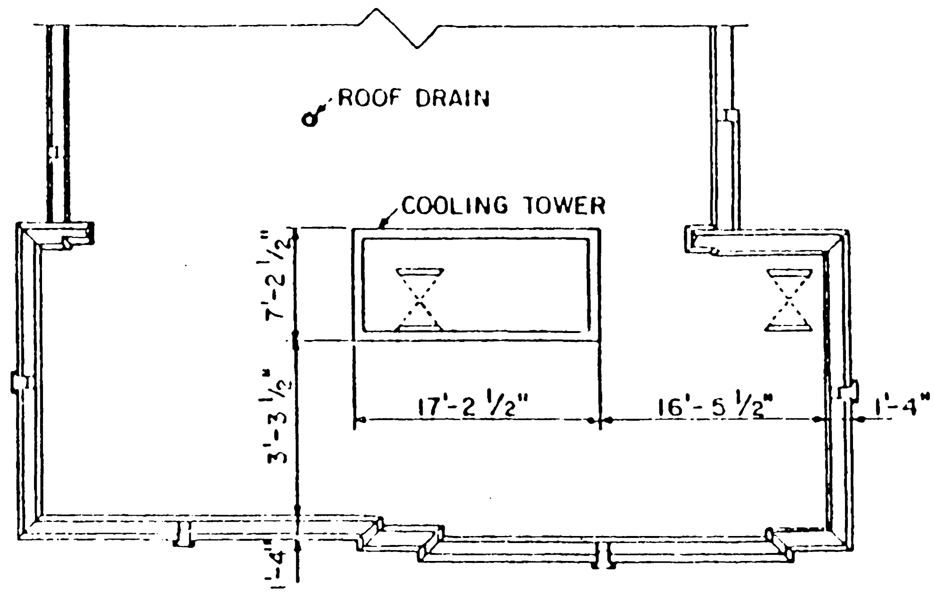


Figure 3. Cooling Tower (Elevation)



ELEVATION

PART PLAN AT ROOF

SCALE 1/8" = 1' 0"

Figure 4. Cooling Tower Location (Plan)

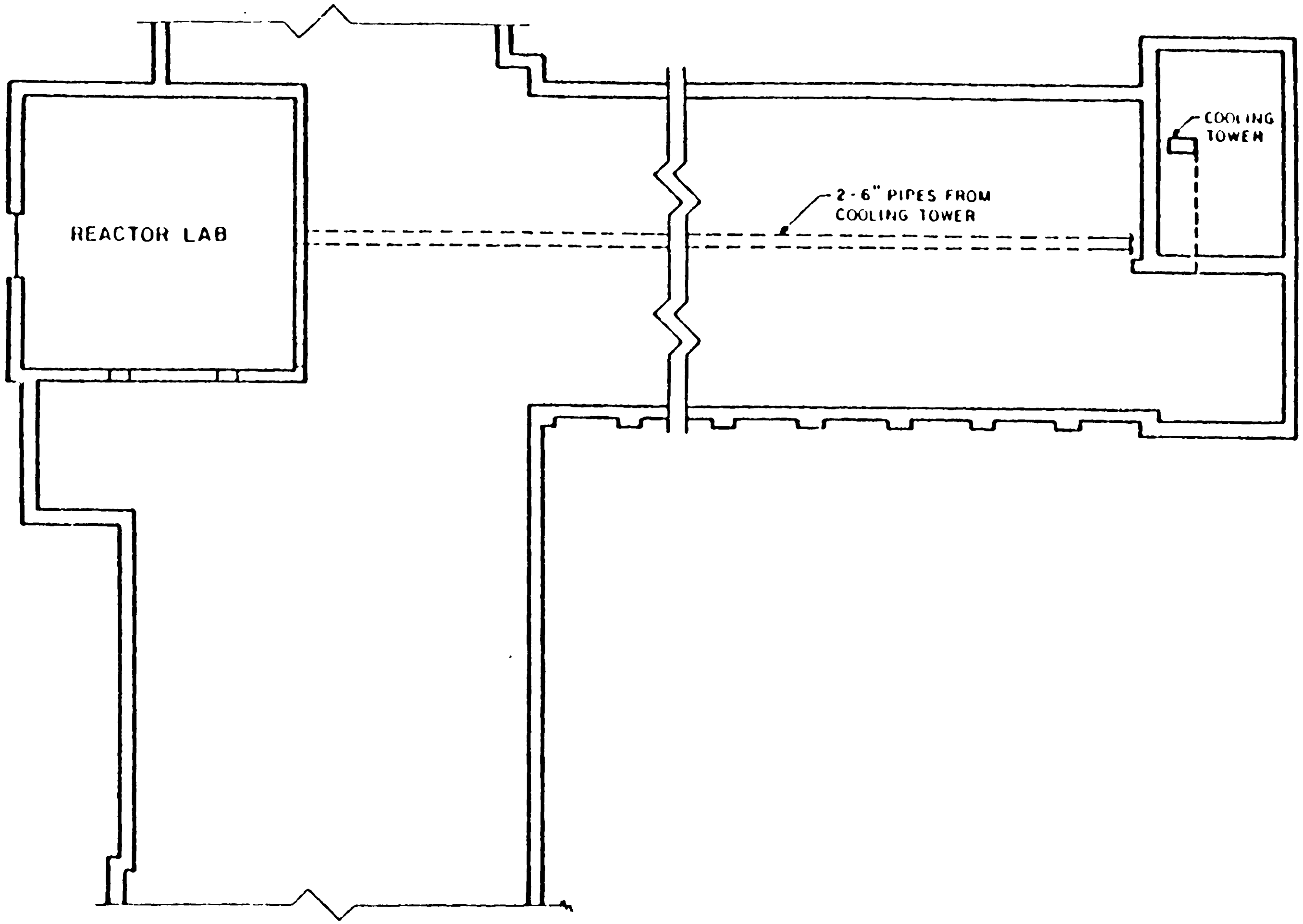


Figure 5. Cooling Tower and Piping Run

IV. STEADY-STATE THERMAL ANALYSIS

Because of the large negative void coefficient of reactivity (0.172% $\Delta k/k/\%$ void)[6], the reactor will be shutdown if boiling occurs in the core. Therefore, boiling is not allowed in the VPI&SU reactor; i.e., the following limiting conditions cannot be exceeded:

(1) bulk boiling of the coolant must not occur; and,

(2) surface boiling should not occur on any of the fuel plates.

To insure that bulk boiling does not occur, the hot channel coolant outlet temperature must be less than the saturation temperature of the coolant ($212^{o}F$). However, the safety limit of maximum coolant temperature is only $170^{o}F$ for the VPI&SU reactor[1]; therefore, it is necessary to insure that the maximum coolant temperature does not exceed $170^{o}F$. To insure that surface boiling does not occur, the maximum fuel plate surface temperature must be less than $220^{o}F$. To initiate and maintain surface boiling a small degree of superheat is required.

The aim of the following sections in this chapter will be to demonstrate that the above temperature limits will not be exceeded under normal operating conditions.

(i) Maximum Coolant Temperature ($T_{h.h.c.}$)

It is clear that the maximum coolant temperature will occur at the outlet of the hot channel. In order to obtain this temperature, it is necessary to know the coolant inlet temperature, T_c, and the temperature rise from channel inlet to outlet ($T_{h.h.c.}-T_c$). When the reactor power is 500 KW, these two quantities can be calculated as follows:

(a) Reactor Core Heat Balance

All the heat generated in the core must be removed by the coolant flow; therefore, the following relation must be satisfied:

$$3413\ P = CM_pC_p(T_h - T_c) \tag{1}$$

where M_p = primary coolant flow rate (gpm)

T_h = core outlet temperature (^oF)

C = conversion factor = 497 lbm/hr/gpm

C_p = coolant specific heat (Btu/lbm-^oF)

P = reactor power (KW)

Rearranging Eq. (1) gives

$$T_h = T_c + \frac{P}{0.146\ M_p} \tag{2}$$

(b) Intermediate Heat Exchanger Heat Balance

An expression for t_h can be derived in the same manner as in (a):

$$t_h = t_c + \frac{P}{0.146\ M_s} \tag{3}$$

where t_h = heat exchanger outlet temperature (^oF)

t_c = heat exchanger inlet temperature(^oF)

M_s = secondary coolant flow rate (gpm.)

The heat transferred in a pure counterflow heat exchanger can be expressed by the following equation:

$$3414\ P = (A \cdot U\ )\Delta T_L \tag{4}$$

where ΔT_L = log-mean-temperature difference for a pure counterflow heat exchanger(^oF)

A = effective heat transfer area (ft^2)

U = overall heat transfer coefficient ($Btu/hr\text{-}ft^2\text{-}^oF$)

ΔT_L is defined by

$$\Delta T_L = \frac{(T_h - t_h) - (T_c - t_c)}{\ln\left(\frac{T_h - t_h}{T_c - t_c}\right)} \tag{5}$$

By combining Eqs. (2), (3), (4) and (5), the following expression is obtained for $(T_c - t_c)$:

$$T_c - t_c = \frac{P}{0.146(e^x - 1)} \left(\frac{1}{M_p} - \frac{1}{M_s}\right) \tag{6}$$

where

$$x = \frac{(A \cdot U)}{498} \left(\frac{1}{M_p} - \frac{1}{M_s}\right).$$

In order to make use of Eq. (6), the overall heat transfer coefficient, U, is required. By definition, U can be written as:

$$\frac{1}{U} = \frac{1}{h_p} + \frac{1}{h_s} + \frac{x}{k} + d_{f_p} + d_{f_s} \tag{7}$$

where h = film heat transfer coefficient ($Btu/hr\text{-}ft^2\text{-}^oF$)

x = heat exchanger plate thickness (ft)

k = plate thermal conductivity ($Btu/hr\text{-}ft\text{-}^oF$)

d_f = fouling factor ($ft^2\text{-}^oF\text{-}hr/Btu$)

and the subscripts are

p = primary

s = secondary.

A widely adopted correlation for estimating film coefficients for turbulent flow in plate exchangers is [7]:

$$h = 0.2536\left(\frac{k_{av}}{D_e}\right)(Re_{av})^{0.65}(Pr_{av})^{0.4} \tag{8}$$

where k = coolant thermal conductivity (Btu/hr-ft-^oF)

D_e = equivalent diameter (ft)

Re = Reynolds number, $\rho VD_e/\mu$, dimensionless

Pr = Prandtl number, $C_p\mu/k$, dimensionless

μ = coolant viscosity (lbm/ft-sec)

V = coolant velocity (ft/sec)

ρ = coolant density (lbm/ft^3)

C_p = coolant specific heat (Btu/lbm-^oF)

and the subscript av indicates that the above parameters are evaluated at the average temperature of the coolant.

The equivalent diameter, D_e, is defined as four times the cross-sectional area of the channel, divided by the wetted perimeter of the channel, or $D_e = (4wb)/(2w+2b)$ where w is the plate width in ft and b is the distance between plates in ft.

For an ambient air temperature of 84^oF, the inlet temperature on the secondary side will be fixed at 95^oF by cooling tower performance. The proposed temperature drop across the cooling tower is 5^oF; therefore, the outlet temperature on the secondary side will be 100^oF. With the temperatures known, the film coefficient on the secondary side can be determined by Eq. (8). However, since the temperatures on the primary side are not known, it is necessary to use the following iterative approach to calculate the overall heat-transfer coefficient:

(1) Assume the primary coolant outlet temperature T_c.

(2) Calculate the primary coolant inlet temperature T_h with Eq.(2).

(3) Calculate the film coefficients with Eq. (8).

(4) Calculate the overall heat transfer coefficient with Eq. (7).

(5) Calculate the primary coolant outlet tempature with Eq.(6).

(6) Compare the value determined in step 5 with the corresponding value assumed in step 1. If the calculated value does not agree with the assumed value, repeat the above steps, replacing the assumed value with the value calculated in step 5, until agreement.

The following numerical values are used in the calculation:

$M_p = 90$ gpm

$M_s = 685$ gpm

$A = 92.57\ ft^2$

$x = 0.002$ ft

$k = 8.65\ Btu/hrft^{o}F$

$d_{f_p} = 0.0002\ (hrft^2\ {}^{o}F)/Btu$

$d_{f_s} = 0.0002\ (hrft^2\ {}^{o}F)/Btu$

$w = 1.92$ ft

$b = 0.017$ ft

$V_p = 1.23$ ft/sec

$V_s = 4.269$ ft/sec.

All the values concerning the heat exchanger are supplied by the manufacturer. Assume $T_c = 110^{o}F$. After several iterations, the results are $T_c = 111^{o}F$, $U = 617\ Btu/(hr\text{-}ft^2\text{-}{}^{o}F)$.

(c) Hot Channel Heat Balance

A heat balance on the hot channel gives

$$m_p C_p (T_{h.h.c} - T_c) = \frac{3413\ PF}{N} \tag{9}$$

where m_p = coolant flow rate per channel, lbm/hr

N = number of flow channels

F = (mean heat flux in hot channel)/(mean heat flux in average channel) = hot channel factor.

Eq. (9) can be rearranged as:

$$T_{h.h.c.} = \frac{3413\ PF}{m_p C_p N} + T_c \tag{10}$$

The hot channel factor F is determined from the flux measurements made by Stam [8] as 1.223.

The coolant flow rate per channel, m_p, is obtained by dividing the total primary coolant flow rate (90 gpm = 45,000 lbm/hr) by the number of flow channels (156), and the computed value of m_p is 286 lbm/hr. By substituting the numerical values in Eq. (10), the maximum coolant temperature $T_{h.h.c.}$ is found to be 158^{o}F. This value is well below the safety limit of coolant temperature for the VPI&SU reactor.

(ii) Core Heat Transfer Coefficient

In order to calculate the maximum fuel surface temperature, it is necessary to determine the heat transfer coefficient in the core.

The flow within the core of the reactor is in the "mixed flow" region, that is, a region in which forced laminar flow and flow due to natural convection are superimposed. The relative importance of each is a function of the reactor power. For a given Re, the larger the reactor power the larger is the relative effect of natural convection. Further, for fuel plate temperatures in the region of $200^{o}F$, it is likely that the natural convection flow will be turbulent with its consequently larger heat transfer coefficient. In short, the heat transfer mechanism within the core is a combination of forced and free convection.

Jackson and others have reported data on combined free and forced convection in a constant-temperature vertical tube [9]. They also have derived an equation for Nusselt number which fitted the experimental data. The equation is also applicable for rectangular channels provided that the equivalent diameter is used. The equation is expressed as follows:

$$h = \frac{k_{mb}}{De} 1.128 \left\{ \left(RePr \frac{De}{x} \right)_{mb} + \left[3.05 \left(GrPr \frac{De}{L} \right)_{w} Pr_{mb} \right]^{0.4} \right\}^{\frac{1}{2}} \qquad (11)$$

where h = average heat transfer coefficient from 0 to x (Btu/ hr-ft^{2}-^{o}F)

Re = Reynolds number, $(\rho v De)/\mu$, dimensionless

Pr = Prandtl number, $C\rho\mu/k$, dimensionless

Gr = Grashof number, $\rho^{2} g\beta(T_{w}-T_{c})De^{3}/\mu^{2}$, dimensionless

De = channel equivalent diameter (ft)

k = coolant thermal conductivity (Btu/ hr-ft-^{o}F)

μ = coolant dynamic viscosity (lbm/fthr)

ρ = coolant density (lbm/ft^{3})

C_p = coolant specific heat(Btu/lbm^oF)

g = gravitational acceleration = 4.165 x 10^8 ft/hr^2

B = coefficient of volumetric expansion of coolant ($°F^{-1}$)

$\overline{T_w}$ = average fuel surface temperature (^oF)

T_c = coolant inlet temperature (^oF)

V = mean coolant velocity (ft/hr)

L = fuel plate length (ft)

x = position along fuel plate,(ft)

and the subscripts are:

w = based on surface temperature

mb = based on mean bulk temperature = $(T_{h.h.c.}+T_c)/2$.

The conditions on Eq. (11) are:

(1) $1.05 \times 10^5 < (GrPr \frac{De}{L})_w < 1.30 \times 10^6$

(2) $51 < (RePr \frac{De}{L})_{mb} < 2177$.

It is necessary to know the average surface temperature, $\overline{T}_w$, to calculate h with Eq. 11 . Unfortunately, this value is not available; therefore, the heat-transfer coefficient, h, has to be determined by an iterative approach. The following procedures were used to evaluate h:

(1) Assume an initial average surface temparature, $\overline{T}_w$.

(2) Determine the fluid properties based on the assumed value and substitute these properties together with other numerical values in Eq.11 to calculate h at x = L.

(3) Determine the new surface temperature with the following equation:

$$\bar{T}_w - T_{mb} = \frac{\bar{q}''_w}{h} \tag{12}$$

where T_{mb} = mean bulk coolant temperature (°F)

$\bar{q}''_w$ = average hot channel heat flux (Btu/hr-ft^2)

(4) Compare the value determined in step 3 with the corresponding value assumed in step 1. If the value does not agree with the assumed value, repeat the above steps, replacing the assumed value calculated in step 3 until the values agree.

The following numerical values were used to evaluate h:

T_c = 111°F

$T_{h.h.c.}$ = 158°F

T_{mb} = 135°F

L = 2.167 ft

$\bar{q}''_w$ = 12150 Btu/hr-ft^2

De = 0.06 ft

V = 617 ft/hr.

The assumed surface temperature is 195°F. By following the procedures described above the heat transfer coefficient, h, and surface temperature, $\bar{T}_w$, were found to be:

$$h = 221 \text{ Btu/ hr-ft}^2\text{-}^\circ\text{F} \quad \text{and} \quad \bar{T}_w = 189^\circ\text{F}.$$

Additional calculations were made to verify the validity of Eq. 11 . A comparison of the calculated value (189 Btu/ hr-ft^2-°F) with the experimental data obtained from the heat-transfer rig test in Reference [4] (190 Btu/hr-ft^2-°F) shows that the equation was valid in predicting the heat-transfer coefficient. However, more data are needed to provide a complete check on the equation. A further examination of Eq. (11) will be considered later in this analysis.

(iii) Position of Hot Spot

In order to determine the maximum fuel-plate surface temperature, it is essential to know where in the hot channel this temperature occurs, i.e., the hot spot position. This position can be found by performing an energy balance on the reactor hot channel. Applying the basic heat transfer equation,

$$T_w(x) - T_b(x) = \frac{q''_w(x)}{h(x)} \tag{13}$$

where $T_b(x)$ = coolant temperature (^oF)

$q''_w(x)$ = fuel-plate surface heat flux (Btu/hr-ft^2)

An enthalpy balance gives

$$T_b(x) - T_c = \frac{2\int_0^x q''_w(x')w\,dx'}{m_p\ C_p} \tag{14}$$

Now, eliminating $T_b(x)$ by combining (13) and (14) gives

$$T_w(x) = \frac{q''_w(x)}{h(x)} + \frac{2\int_0^x q''_w(x')w\,dx'}{m_p\ C_p} + T_c. \tag{15}$$

It is required to find the value of x which makes $T_w(x)$ a maximum. It is therefore necessary to differentiate Eq. (15), equate to zero and then solve for x.

$$\frac{dT_w}{dx} = \frac{d}{dx}\left[\frac{q''_w(x)}{h(x)}\right] + \frac{2w}{m_p C_p} q_w(x) = 0. \qquad (16)$$

Eq. (16) can be solved by making the following assumptions:

(a) $h(x)$ can be represented by an average constant value; and

(b) the heat flux distribution, $q''_w(x)$, has the same shape as the neutron flux distribution in the reactor hot channel. The heat flux is well represented by the following equation [4]:

$$q''_w(x) = CP(0.515 + 0.485 \sin \frac{\pi x}{L}) \frac{\text{Btu}}{\text{hr-ft}^2} \qquad (17)$$

where C is a constant and P is the reactor power in KW.

The above expression for $q''_w(x)$ is taken from the SURR report. Since the SURR is virtually identical to the VPI&SU reactor, it is justified to use Eq. (17) as the heat flux distribution in the VPI&SU reactor. The constant C is determined from the following equation.

$$\frac{3413\ PF}{N} = 2\int_0^L q''_w(x) w\,dx \qquad (18)$$

where w = fuel plate width = 0.25 ft

L = fuel plate length = 2.167 ft.

N = number of flow channels = 156

All the other terms have the same meaning as previously defined. By substituting Eq. 17 in Eq. 18 and performing the integration, C is found to be 29.5. Therefore, the heat flux distribution in the hot channel is

$$q''_w(x) = 29.5\ P(0.515 + 0.485\ \sin \frac{\pi x}{L})\ \frac{Btu}{hr\text{-}ft^2} \tag{19}$$

and

$$\frac{dq''_w(x)}{dx} = \frac{29.5P\ 0.485\ \pi}{L} \cos \frac{\pi x}{L} \ . \tag{20}$$

Putting Eqs. (19) and (20) into Eq. (16) and using assumption (a) gives

$$\frac{dT_w}{dx} = \frac{29.5P\ 0.485\ \pi}{hL} \cos \frac{\pi x}{L} + \frac{2w}{m_p C_p}\ 29.5P(0.515 + 0.485 \sin \frac{\pi x}{L}) \tag{21}$$

$$= 0$$

Upon substitution of the numerical values into Eq. (21), Eq. (21) becomes:

$$5.18 \cos \frac{\pi x}{L} + 1.380 \sin \frac{\pi x}{L} + 1.465 = 0 \tag{22}$$

Solution of Eq. (22) gives a value of x equal to 0.67L; thus, the position of the hot spot is at x = 0.67L. For 500 KW operation, this value agrees with the experimental value given in the SURR report[4]. The heat flux and temperature distributions are shown qualitatively in Figure 6.

(iv) <u>Maximum Fuel Surface Temperature</u> ($T_{w.h.s.}$)

The maximum fuel surface temperature can be written as:

$$T_{w.h.s.} = (T_{w.h.s.} - T_{b.h.s.}) + (T_{b.h.s.} - T_c) + T_c \tag{23}$$

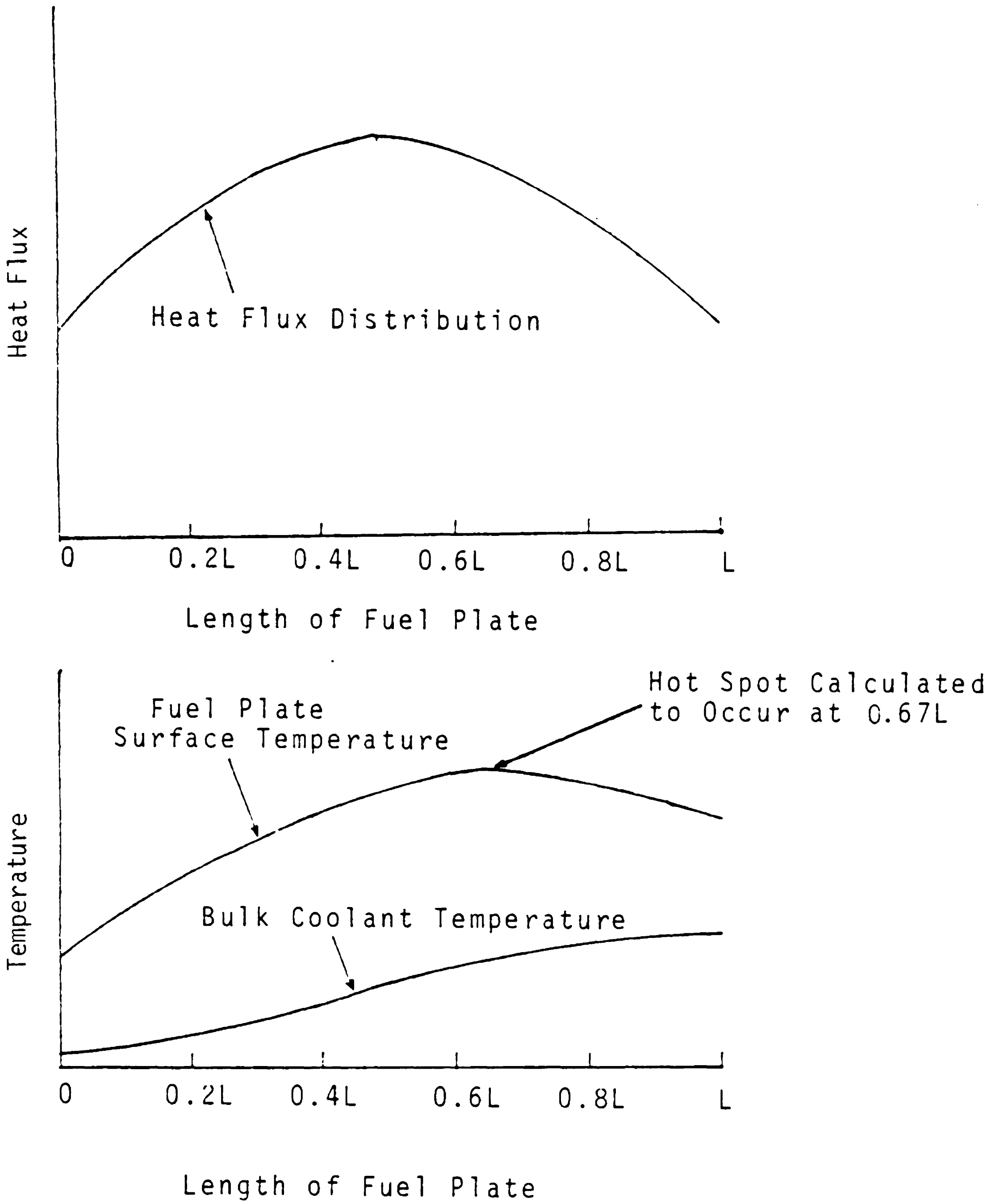


Figure 6. Hot Channel Heat Flux and Temperature Distributions

where $T_{b.h.s.}$ is the coolant temperature at the hot spot and $T_{b.h.s.}-T_c$ is obtained by an energy balance at the hot channel hot spot at 0.67L.

$$m_p C_p (T_{b.h.s.} - T_c) = 2\int_0^{0.67L} q''_w(x) w dx \tag{24}$$

By substituting the expression for $q''_w(x)$ in Eq.(24) and carrying out the integration, the following expression is obtained for $(T_{b.h.s.} - T_c)$:

$$T_{b.h.s.} - T_c = \frac{34.1 \ PWL}{m_p C_p} . \tag{25}$$

It is clear that $(T_{w.h.s.} - T_{b.h.s.})$ can be written as

$$T_{w.h.s.} - T_{b.h.s.} = \frac{q''_{w.h.s.}}{h} \tag{26}$$

where $q''_{w.h.s.}$ is the surface heat flux at the hot spot.

Substituting x = 0.67L in Eq. (19) gives

$$q''_{w.h.s.} = 27.5 \ P \tag{27}$$

By combining Eqs. (23), (25), (26) and (27), the maximum fuel surface temperature is found to be

$$T_{w.h.s.} = \frac{27.5P}{h} + \frac{34.1 \ PWL}{m_p C_p} + T_c \tag{28}$$

Upon substitution of numerical values in Eq. (28), the computed value of $T_{w.h.s.}$ is found to be 205^{o}F which is below the limiting temperature of 220^{o}F.

(v) <u>Maximum Steady-State Operating Power</u>

The aim of this section will be to determine the maximum steady-state power at which the reactor can operate without exceeding the safety limitations of the reactor.

As mentioned before, boiling is not allowed in the VPI&SU Reactor and the heat flux at the onset of nucleate boiling establishes an upper limit on reactor power level. Several techniques are available for predicting the incipient boiling heat flux. This value is obtained when a wall temperature predicted by a single-phase forced convection heat-transfer coefficient is equal to the wall temperature predicted by a subcooled nucleate boiling correlation. One simple approach is that recommended by Bergles and Rohsenow[10] in which the onset of nucleate boiling occurs when wall temperature from their nucleate boiling correlation equals wall temperature calculated by the single-phase convection equation. The boiling correlation and single-phase convection equations are given as:

$$T^*_{w.h.s.} = T_{sat} + \left(\frac{q''_{w.h.s.}}{15.6p^{1.156}}\right)^{\frac{1}{2.3\,p^{.0234}}} \tag{29}$$

$$T_{w.h.s.} = T_{b.h.s.} + \frac{q''_{w.h.s.}}{h} \tag{30}$$

where $T_{b.h.s.}$ = coolant temperature at the hot spot (^oF)

$T_{w.h.s.}$ = fuel-plate surface temperature at the hot spot predicted by convection equation (^oF)

$T^*_{w.h.s.}$ = fuel-plate surface temperature at the hot spot predicted by boiling correlation (^oF)

T_{sat} = coolant saturation temperature = 212^oF

$q''_{w.h.s.}$ = fuel-plate surface heat flux at hot spot ($Btu/hr\text{-}ft^2$)

p = atmospheric pressure = 14.7 psia

h = single-phase convection heat-transfer coefficient ($Btu/hr\text{-}ft^2\text{-}°F$).

The heat flux at the onset of nucleate boiling was calculated from the intersection of Eq. (29) and Eq. (30). Figure 7 illustrates graphically the method for calculating the heat flux at the onset of nucleate boiling.

The results of the calculation are summarized in Table 1 and plotted in Figure 7. As seen in Figure 7, the onset of boiling occurs approximately at the power level of 560 KW ($q''_{w.h.s.} = 15,400$ $Btu/hr\text{-}ft^2$). This value establishes the maximum steady-state power of the VPI&SU reactor.

As will be seen later in this analysis, the heat-transfer coefficient determined in this analysis was underestimated. Therefore, the maximum surface temperature, $T_{w.h.s.}$, should be lower than the ones shown in Table 1. Thus the maximum steady-state power should be higher than 560 KW. An estimation of the error in this calculation will be given in this analysis.

Table 1

MAXIMUM FUEL-PLATE SURFACE TEMPERATURE AT VARIOUS POWERS

P, KW	100	200	300	400	500	600
$q''_{w.h.s.}$ $\frac{Btu}{hr\text{-}ft^2}$	2,832	5,580	8,320	11,000	13,750	16,500
$T_{b.h.s.}$, ^oF	104	115	124	134	143	153
h, $\frac{Btu}{hr\text{-}ft^2\text{-}^oF}$	168	187	200	214	221	226
$T^*_{w.h.s.}$, ^oF	214	215	215.5	216	216.5	217
$T_{w.h.s.}$, ^oF	121	144	158	185	205	226

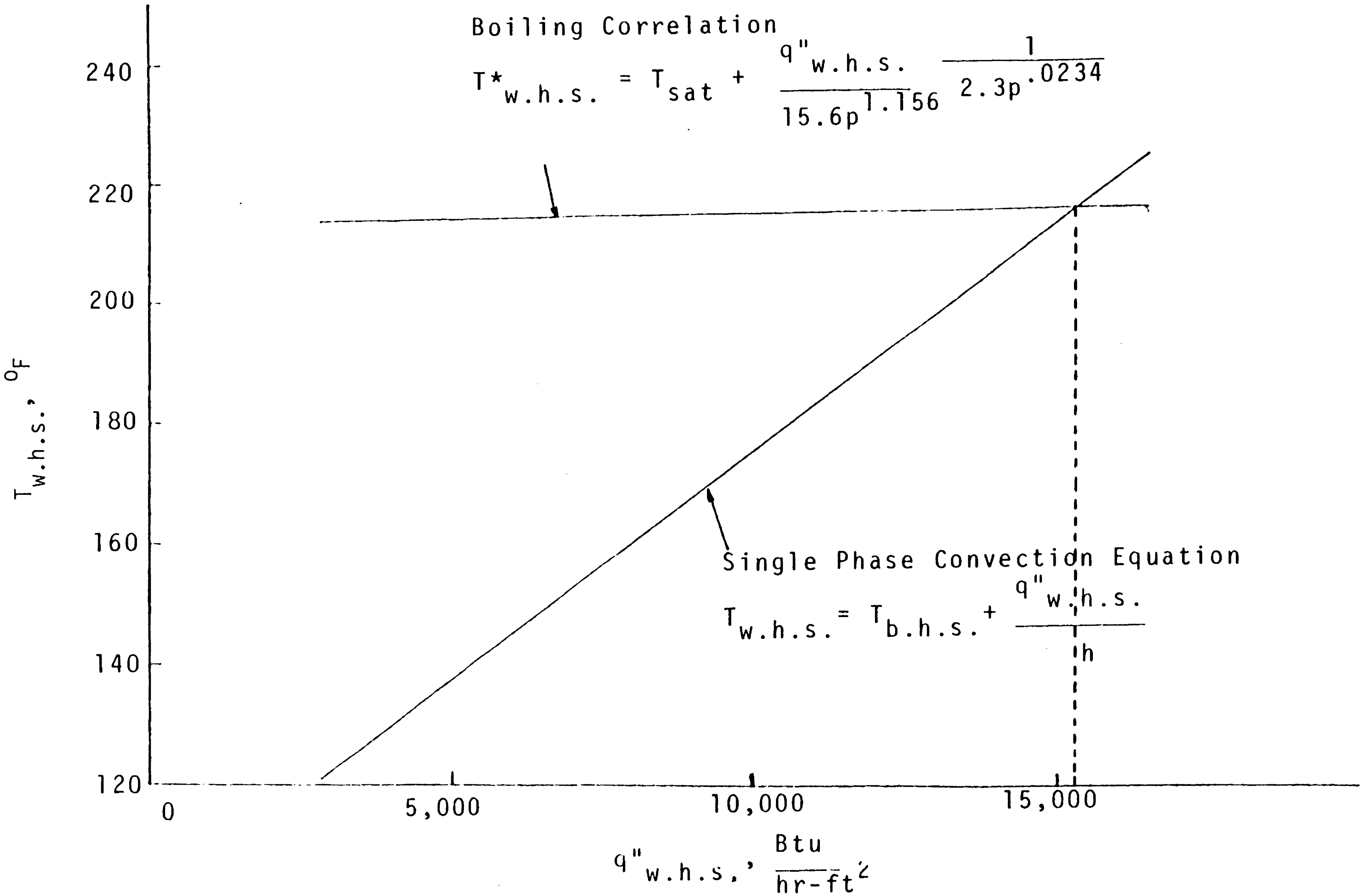


Figure 7. Maximum Fuel Plate Surface Temperature Versus Hot Spot Heat Flux

V. TRANSIENT THERMAL ANALYSIS

The major hypothetical abnormal transients occur as a result of the following accidents:

(1) Loss-of-flow accident (LOFA) due to primary pump failure; and

(2) Loss-of-coolant accident (LOCA) due to a major rupture of the primary cooling system.

In order to maintain the fuel element integrity, fuel melt-down should not occur during the transients. To insure this, the maximum fuel-plate surface temperature must be less than the aluminum melting point of $1220^{0}F$.

(i) Loss-of-Flow Accident (LOFA)

As mentioned before, the reactor is cooled by free and forced convection. Failure of the primary cooling pump may thus lead to inadequate cooling. Such a failure can arise due to mechanical breakdown or loss of electric power.

If a pump failure occurs, the primary system flow begins to coast down when the reduced flow is sensed, the control rods scram, and the reactor is shutdown rapidly. After a few minutes, the heat released by the reactor is due only to fission product decay. Although this heat release is very small when compared to the original power level, it is high enough to cause fuel melt-down unless provision is made for heat removal. In a water-cooled reactor, heat is removed by natural-circulation of the coolant[11]. It is necessary to demonstrate that the natural-circulation cooling is adequate so that the fuel temperatures remain within their desired limits.

Because of the small characteristic dimension of the fuel plate, a lumped system analysis is adequate to calculate the fuel temperature during the transient [12].

In the following analysis, two simplifications can be made: (1) the spatial temperature variation can be neglected; and (2) when the pump loses power, the system pressure does not change significantly and the coolant temperature remains approximately constant [13]. The situation where there is a step reduction in the coolant heat transfer coefficient and a subsequent reduction in the heat generation rate due to a reactor scram is considered.

The rate equation for the heat transfer from the fuel to the coolant can be written as:

$$\rho c_p v \frac{dT_w}{dt} = q(t) - hA(T_w - T_{mb}) \tag{31}$$

where ρ = fuel density (lbm/ft^3)

c_p = fuel heat capacity (Btu/ lbm-^oF)

v = fuel volume (ft^3)

T_w = fuel temperature (^oF)

$q(t)$ = decay power (Btu/hr)

h = coolant heat transfer coefficient (Btu/ hrft^2 ^oF)

A = heat-transfer area (ft^2)

T_{mb} = mean bulk coolant temperature (^oF)

Rearranging Eq. (31) gives:

$$\frac{d\theta}{dt} + a\theta = q' \tag{32}$$

where $\theta = T_w - T_{mb}$

$a = hA/\rho c_p v$

$q' = q(t)/\rho c_p v.$

Employing the Laplace transform, Eq. (32) becomes:

$$S\bar{\theta}(S) + a\bar{\theta}(S) - \theta(0) = q'(S) \tag{33}$$

where $\theta(0)$ is the initial value of $\theta(t)$.

Rearranging Eq. (33), one obtains:

$$\theta(S) = \frac{\theta(0)}{S+a} + \frac{q'(S)}{S+a} \tag{34}$$

The function $\theta(t)$ can be determined from the inverse transform of Eq. (34) provided that expressions for q(t) and the coolant heat transfer coefficient are available. Approximate expressions that may be used for the decay heat in the fuel are [14]:

$$\frac{q(t)}{q_o} = 0.056\ e^{-0.004t} + 0.20\ e^{-0.1t} \qquad t \leq 60 \text{ sec}$$

$$= 0.031 + 0.025\ e^{-0.009t} \qquad 60 \text{ sec} \leq t \leq 350 \text{ sec} \tag{35}$$

where q_o is the original power level (Btu/hr)

The natural-convection heat-transfer coefficient can be evaluated with the following equation [15]:

$$h = \frac{1}{24}\frac{k_w}{L}(Gr\ Pr)_w \left[1 - e^{-\frac{35L}{b(Gr\ Pr)_w}}\right]^{3/4} \tag{36}$$

where k = coolant thermal conductivity (Btu/hr-ft-°F)

L = fuel-plate length (ft)

b = channel width (ft)

Gr = Grashof number, $b^3 g\rho^2 B(T_w - T_c)/\mu^2$, dimensionless

Pr = Prandtl number, $C_p \mu / k$, dimensionless

ρ = coolant density (lbm/ft^3)

g = gravitational acceleration = 4.156×10^8 ft/hr^2

B = volumetric expansion coefficient ($1/^oF$)

μ = coolant dynamic viscosity (lbm/ ft-hr)

T_w = fuel temperature (^oF)

T_c = hot channel inlet temperature (^oF)

C_p = coolant specific heat (Btu/lbm-°F)

and the subscript w is based on the fuel-plate surface temperature.

With the fluid properties known, the heat-transfer coefficient can be calculated; the computed value for h is 121 Btu/ (hr-ft^2-°F).

The following values were used in the calculation:

$\theta(0)$ = 54°F

q_o = 13165 Btu/hr

T_{mb} = 135^oF

ρ = 169 lbm/ft^3

c_p = 0.225 Btu/lbm^oF

v = 0.0036 ft^3

A = 1.09 ft^2.

Substitution of the numerical values into Eq. (34), followed by the inverse transformation, yields θ as function of time:

$$\theta(t) = 17.54\, e^{-0.268t} + 5.67\, e^{-0.004t} + 31.79\, e^{-0.1t} \qquad t \leq 60 \text{ sec}$$

$$= 3.09 - 1.13\, e^{-0.268(t-60)} + 2.58\, e^{-0.009(t-60)} \qquad (37)$$

$$60 < t \leq 350 \text{ sec.}$$

The temperature difference between $T_w(t)$ and T_{mb} versus time is plotted in Figure 8. As seen, the fuel temperature decreases with time during the transient. This result agrees with the experimental data obtained after reactor shutdown with delayed water dump in the Scottish University report [4]. This finding is significant as it demonstrates that a loss-of-flow accident (LOFA) for the 500 KW model should not result in a fission product release through fuel melting.

(ii) Loss-of-Coolant Accident (LOCA)

The most serious accident generally considered in safeguards analysis is the hypothetical loss-of-coolant accident occasioned by the rupture of the primary coolant piping or core tank.

If rupture of the core tank or piping occurs, the moderator will drain from the core tank, thus reducing k_{eff} by 30 percent and shutting the reactor down [6]. After shutdown, the reactor continues to generate decay heat. Because of the loss of coolant water, the temperature of the fuel elements will rise from their normal values at a rate that depends, in part, on the decay heating rate and the mass and specific heat of the fuel. As the temperatures increase, heat will be lost from the fuel elements to the surroundings through the combined mechanisms of conduction, thermal radiation and natural

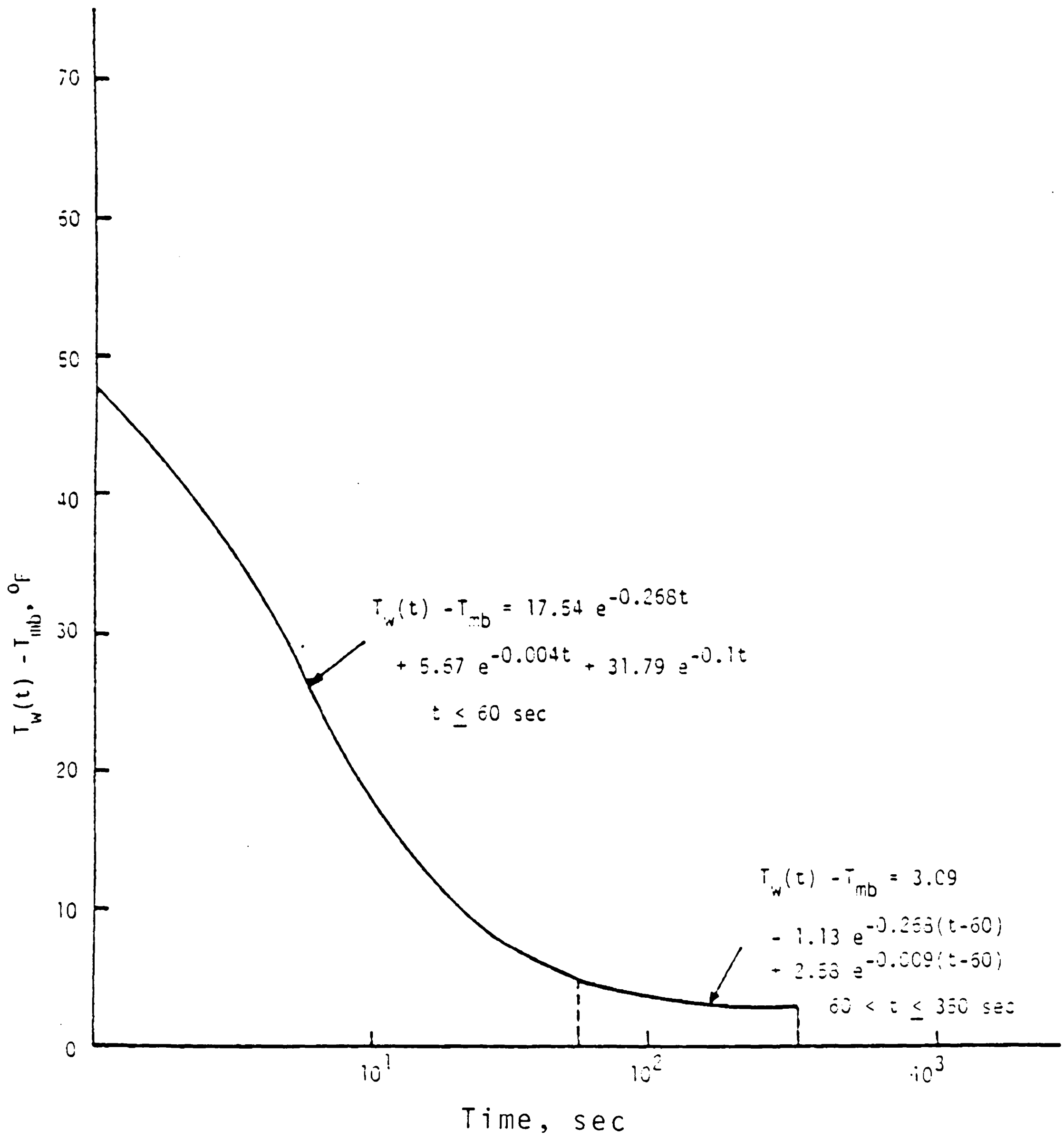


Figure 8. Temperature Difference $T_w(t) - T_{mb}$ Versus Time After LOFA

convection of the air. Depending on many variables, the rate of heat loss will at some point in the transient become exactly in balance with the rate of heat generation from fission product decay and core temperatures will then begin to decrease. It is necessary to show that the maximum temperature of the fuel elements will not exceed the aluminum melting point during the accident. A temperature curve following the accident is shown in Figure 9.

G. E. Cort has performed a computer-modeled analysis to evaluate the maximum core temperatures after such accident[16]. The analysis was conducted with an existing two-dimensional finite-element computer program[17] with some modifications. Heat transfer in the horizontal direction parallel to the surface of the fuel plates is assumed negligible. Four different cases have been run on the computer model at 500 kw. Two cases were run with natural convection air flow between the fuel plates and two with stagnant air in the space between the plates. Two cases were run with the core in the uncrushed condition and two with the core crushed. The maximum calculated fuel temperatures for the four cases are summarized in Table 2.

As can be seen in Table 2, some fuel melting may take place under the extreme condition of complete blockage of air flow through the fuel elements. It is quite possible that refinement of some of the conservative assumptions in Cort's analysis, such as a three-dimensional rather than two-dimensional models would show that melting would not occur at 500 KW even under such highly unlikely postulated accident.

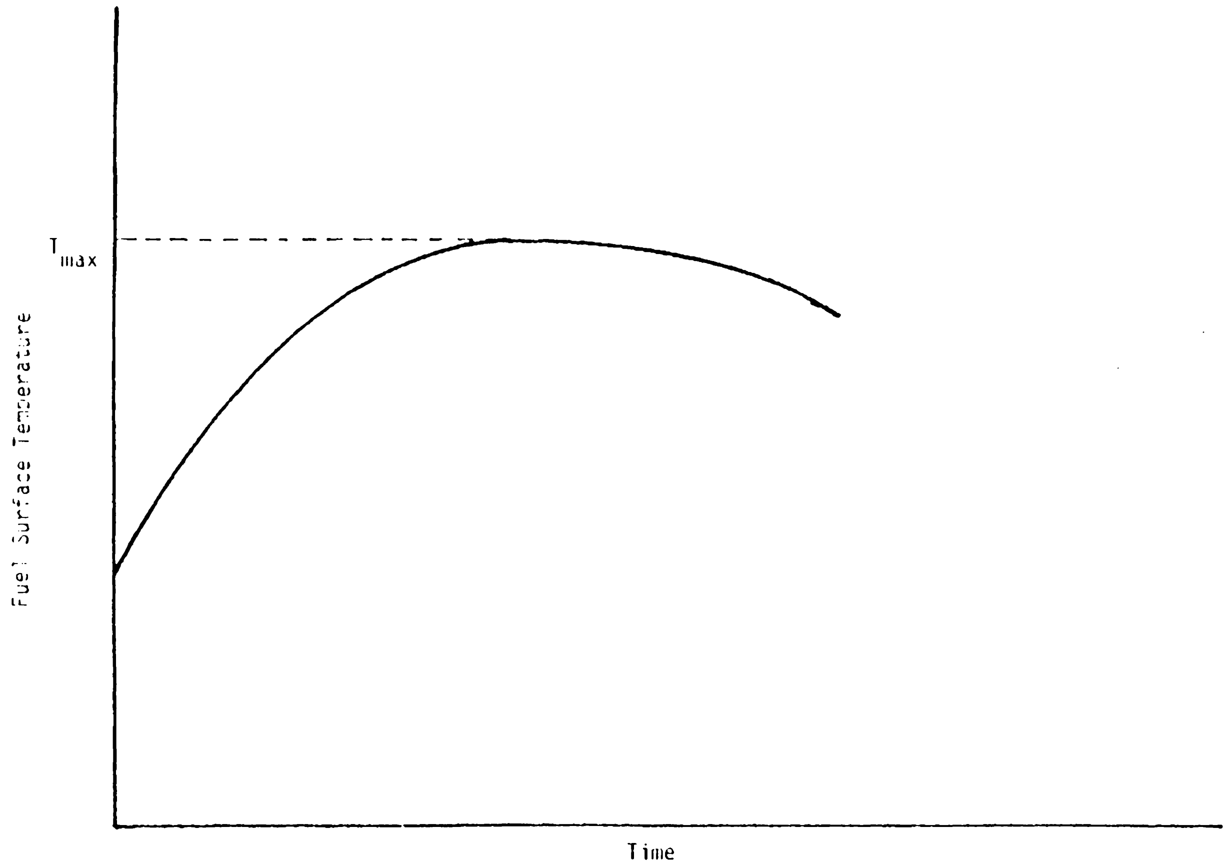


Figure 9. Fuel Surface Temperature Versus Time After LOCA

Table 2

Calculated Peak Temperature in Argonaut Fuel Following Loss-of-Coolant Accident

	Uncrushed	Crushed[a]
Natural Convection Air Flow	673 ^{o}F	1125 ^{o}F
No Air Flow	1809 ^{o}F[b]	1335 ^{o}F[b]

(a) The crushing of the fuel plate is in the lateral direction and caused by an earthquake. The coolant gap between fuel plates is reduced to one-half the nominal value.

(b) Fuel would melt before reaching these temperatures.

VI. Fuel Plate Vibration

Vibration of the fuel plates is caused by the flow of the coolant. When a high-speed flow passes through a gradually narrowing passage, the pressure head of the stream is converted into a velocity head, which creates a suction force on the wall. If the wall is movable or flexible, the passage will be reduced to zero and the flow will be stopped. As soon as the flow becomes stagnant, the stream pressure increases to its maximum value, pushing the wall back, thus providing a wide passage again. These suction and pushing forces can act periodically and thus vibrate the core structure. This mechanism appears to govern the vibration of parallel fuel plates.

The analysis by Miller[18] applies the Bernoulli theorem for incompressible flow to determine the pressure difference which would be developed across the plates by modifications of local fluid velocities as a result of small plate deflections. Equating these pressure differences to those which would be required to produce the corresponding deflections gives an expression for the critical velocity in terms of plate and fluid parameters. Once the flow velocity exceeds this critical velocity, the plates will collapse.

In VPI&SU reactor, the fuel plates are fixed at both ends. Figure 10 shows a side view of the deflected plates. This mode of collapse would be approximated in such manner that the transverse curvatures would be small in comparison

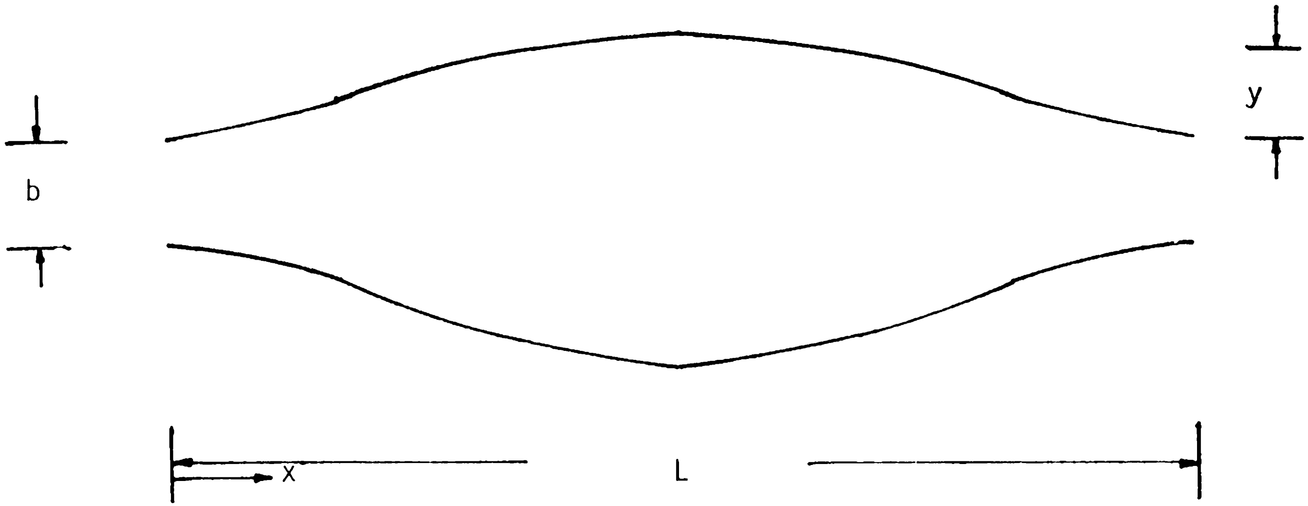


Figure 10 Side View of the Deflected Plates

with longitudinal curvature. In this case all plates would tend to deform in a similar manner over their full length with adjacent plates deflecting in opposite directions at any longitudinal position.

The velocity at the largest cross-section of an enlarged channel V_1 is

$$V_1 = \frac{V_o b}{b+2y} \tag{38}$$

where V_o = coolant velocity in the undeformed channel

b = normal flow channel thickness

y = maximum deflection of the plate relative to supports

and, in a constricted channel, is

$$V_2 = \frac{V_o b}{b-2y} \tag{39}$$

The pressure difference across the plates, from Bernoulli's theorem and Eqs. (38) and (39) is:

$$p = \frac{\rho V_o^2}{2g}\left[\left(\frac{1}{1-\frac{2y}{b}}\right)^2 - \left(\frac{1}{1+\frac{2y}{b}}\right)^2\right] \tag{40}$$

In the limit as y/b approaches zero, Eq. (40) becomes

$$p = \frac{4\rho V_o^2 y}{gb} \tag{41}$$

The equation for the deflection curve of one of the plates is

$$\frac{EI}{(1-\nu^2)} \frac{d^4y}{dx^4} = p \tag{42}$$

where E = Young's modulus of elasticity

I = Moment of inertia of the beam cross-section per unit width of the beam

ν = Poisson's ratio

x = distance along the plate

Since the plates are fixed at both ends, the solution of Eq. (42) must satisfy the following boundary conditions:

(1) $y(0) = 0$

(2) $\frac{dy(0)}{dx} = 0$

(3) $\frac{dy(L/2)}{dx} = 0$

(4) $y(L) = 0$

(5) $\frac{dy(L)}{dx} = 0$

The solution satisfying Eq. (42) and the above boundary conditions is

$$y(x) = \frac{px^2(1-\nu^2)(x^2-2Lx+L^2)}{24EI} \tag{43}$$

The maximum deflection occurs at x = L/2 and is equal to

$$y = \frac{p(1-\nu^2)L^4}{384EI} \tag{44}$$

Substitution of Eq. (44) and the expression for I in terms of plate thickness ($I = d^3/12$) in Eq. (41) gives the critical velocity

$$V_c = \left[\frac{8Ed^3gb}{\rho L^4(1-\nu^2)} \right]^{1/2} \tag{45}$$

Wambaganss[19] extended the work to include second order terms in Eq. (40) and concluded that the critical velocity could be from 0.63 to 0.85 times those derived by Miller.

For conservative purpose, a correction factor of 0.63 was used to calculate the critical velocity; the following values were substituted in Eq. (45) :

$E = 6.7 \times 10^6$ psi

$d = 6.7 \times 10^{-3}$ ft

$\rho = 61.58$ lbm/ft^3

$L = 2.167$ ft

$\nu = 0.33$

$b = 0.03$ ft

Upon substitution of the above values, the critical velocity is found to be 1.026 ft/sec. The proposed flow velocity for 500 KW operation is 0.171 ft/sec. This value is considerably below the critical velocity; hence, it is concluded that the fuel plate vibration problem will not be significant for 500 KW operation.

VII. DISCUSSION

By extrapolating the temperature data given in the SURR report for a power level of 500 KW, it is found that the temperature difference at the hot spot ($T_{w.h.s.} - T_{b.h.s.}$) is approximately 51^{o}F which is considerably lower than the value of 62^{o}F predicted by Eq. (26). This result shows that the heat-transfer coefficient used to predict ($T_{w.h.s.} - T_{b.h.s.}$) is underestimated by Eq. (11) and the maximum fuel-plate surface temperature should be lower than the value predicted in this analysis. The physical reason for this result can be explained as follows:

The natural-convection coefficient is proportional to the surface temperature; i.e., the higher the surface temperature, the larger the natural-convection coefficient. Equation (11) was derived for constant-surface-temperature condition; hence, the natural-convection effects are the same along the whole heating length. Actually, the surface temperature varies along the fuel-plate length. At the position of the hot spot, the surface temperature is much higher than the average surface temperature. Therefore, the actual heat-transfer coefficient at the hot spot will be larger than the value determined by Eq. (11). The experimental curve used for extrapolation is plotted in Figure 11.

By using the temperature data in Figure 11 and following the method as described in Section IV.v, the maximum steady-state power of the VPI&SU reactor is found to be 660 KW. This value is about 17 percent higher than that previously determined in Section IV.v of 560 KW.

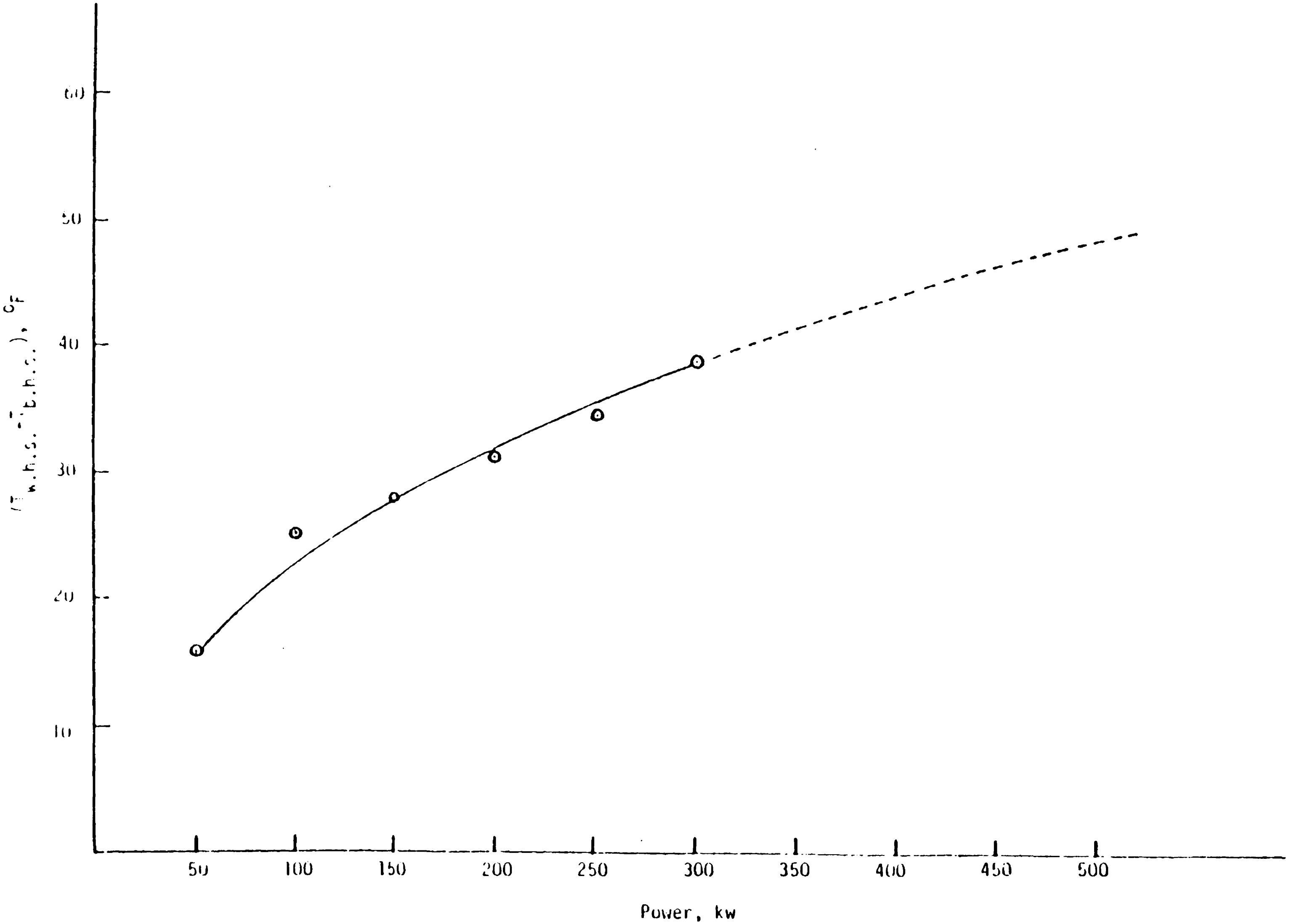


Figure 11. Temperature Difference ($T_{w.h.s.}$-$T_{b.h.s.}$) Versus Reactor Power (from Ref. 4)

VIII. CONCLUSIONS AND RECOMMENDATIONS

A thermal-hydraulic analysis has been performed on the cooling system of the VPI&SU Research Reactor, and the results show that the reactor can be operated at a power level of 500 KW provided that some modifications of the cooling system are made.

The major modifications are to the reactor secondary cooling curcuit and essentially entail increasing the water flow rate to 685 gpm and installing a cooling tower. The primary coolant flow rate is also to be increased to 90 gpm.

The analysis shows that for a fixed cooling tower outlet temperature of 95°F the reactor core water inlet temperature would be 111°F and the hot channel outlet temperature would be 157°F. The maximum fuel-plate surface temperature would be 205°F. These results demonstrate that boiling will not occur in the core for 500 KW operation.

The analysis also shows that the fuel-plate surface temperature will decrease after a loss-of-flow accident. This finding indicates that the loss-of-flow accident should not result in a fission product release through fuel melting. Some fuel melting may take place under the extreme condition of complete blockage of air flow through the fuel elements after a loss-of-coolant accident. However, there is a strong possibility that a calculation based on a three-dimensional model would show that no fuel melts in the core.

Finally, the analysis reveals that the increased primary coolant flow rate will not cause the fuel plates to collapse. In short, the results of the analysis indicate that operation of the VPI&SU Research

Reactor at 500 KW is feasible as far as thermal-hydraulics is concerned. Several recommendations regarding the operation of the reactor at 500 KW are summarized as follows:

(1) Due to the uncertainty of the heat transfer coefficient, it is suggested that an experimental program be carried out to determine the heat transfer coefficient accurately. One alternative is to raise the power level of the VPI&SU reactor to 500 KW temporarily for the experiment consisting of various temperature measurements, overflow line test and flow-induced vibration tests. The results of the experiment can be used to confirm the calculations made in this analysis.

(2) Reference to figure 8 shows that the peak fuel-plate temperature that occurs after reactor shutdown can be reduced significantly by delaying the water dump while keeping the secondary cooling system in operation. It is recommended that when possible this procedure should be used in practice.

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Appendix A

MAXIMUM COOLANT FLOW RATE OF THE DISCHARGE LINE

The maximum flow rate occurs when the discharge line is filled completely with water, as shown in Figure A-1.

Since the diameter of the pipe is relatively small compared to the height of the tank, it is reasonable to assume that the pressure is constant along the position x ($0 \leq x \leq 2R$). Then the velocity V(x) can be derived from the Bernoulli Equation as:

$$V(x) = \sqrt{2gx} \qquad 0 \leq x \leq 2R. \tag{A-1}$$

The flow rate, Q, can be determined by carrying out the following integration:

$$Q = \int V(x)\, dA(x).$$

This integral can be separated into two parts as shown in Figure A-2.

$$Q = \int_{top} V(x)\, dA(x) + \int_{bottom} V(x)\, dA(x)$$

The velocity V(x) expressed in Eq. (A-1) can be rewritten as:

$$\begin{aligned} V(x) &= \sqrt{2g(R-X)} \qquad 0 \leq x \leq R \qquad \text{(for top part of the integral)} \\ V(x) &= \sqrt{2g(R+x)} \qquad 0 \leq x \leq R \qquad \text{(for bottom part of the integral).} \end{aligned} \tag{A-2}$$

As can be seen in Figure A-2:

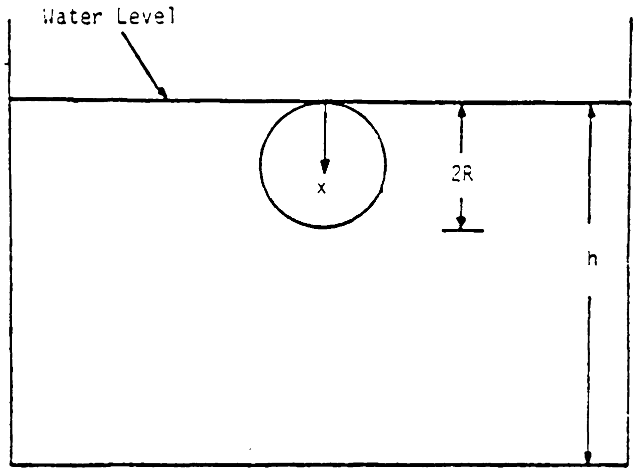

Figure A-1. Position of the Discharge Line

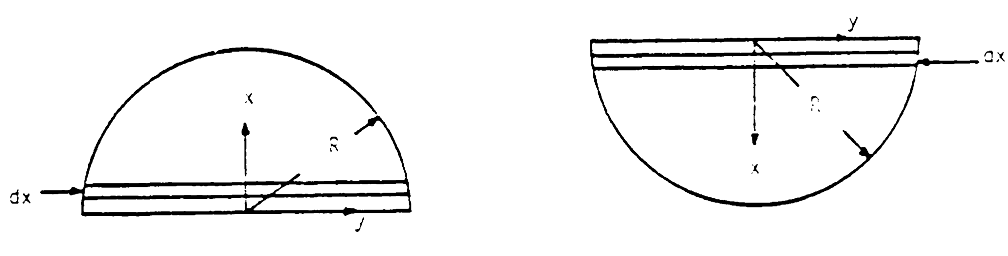

Figure A-2. Cross-Section of the Discharge Line

$$dA(x) = 2ydx$$

$$y = \sqrt{R^2-x^2}$$

$$dA(x) = 2\sqrt{R^2-x^2}\ dx. \tag{A-3}$$

Substituting Eqs. (A-2) and (A-3) into the integrals gives:

$$\int_{top} V(x)\ dA(x)$$

$$= 2\int_0^R \sqrt{2g(R-x)}\ \sqrt{R^2-x^2}\ dx$$

$$= 2\int_0^R \sqrt{2g}\ (R-x)\ \sqrt{R+x}\ dx$$

$$= 2\sqrt{2g}\left(\int_0^R R\ \sqrt{R+x}\ dx - \int_0^R x\ \sqrt{R+x}\ dx\right)$$

$$= 1.15\sqrt{2g}\ R^{5/2}$$

$$\int_{bottom} V(x)dA(x)$$

$$= 2\int_0^R \sqrt{2g(R+x)}\ \sqrt{R^2-x^2}\ dx$$

$$= 2 \int_0^R \sqrt{2g}\,(R+x)\,\sqrt{R-x}\;dx$$

$$= 2\sqrt{2g}\left(\int_0^R R\,\sqrt{R-x}\;dx + \int_0^R x\,\sqrt{R-x}\;dx\right)$$

$$= 1.87\,\sqrt{2g}\,R^{5/2}$$

$$Q = 1.15\,\sqrt{2g}\,R^{5/2} + 1.87\,\sqrt{2g}\,R^{5/2}$$

$$= 3.02\,\sqrt{2g}\,R^{5/2}$$

$$= 24.19\,R^{5/2} \qquad \text{(A-4)}$$

$$R = 0.167 \text{ ft.}$$

Substituting this value into Eq. (A-4), we have

$$Q = 0.275 \text{ ft}^3/\text{sec}$$

$$= 123 \text{ gpm}$$

this value is larger than the core inlet coolant flow rate (90 gpm); therefore, the discharge line is of sufficient size to prevent overflow.

Appendix B

PRESSURE LOSS CALCULATION ON THE SECONDARY COOLANT SYSTEM

(1) Required Pump Head

The pumping head determination procedure for the "open" tower piping loop differs from the conventional "closed" loop piping circuit used for most hydraulic applications. The difference involves consideration of "open" loop static heads.

The closed loop circuit has no need for consideration of static heads for pump selection because of a balance or cancellation of static heads between the supply and return risers. The only pressure loss is caused by flow friction.

The cooling tower circuit differs slightly from the basic "open" circuit in that the discharge piping is seldom connected to a discharge tank. The tower discharge is generally to atmosphere and then into a distribution pan. A typical "open" tower piping circuit is shown in Figure B-1.

For the tower piping circuit, the pump must overcome the piping flow friction loss (piping, heat exchanger, valves, etc.). It must also provide the static head between the open discharge and the water level, (H_0).

The required pump head can be calculated as follows: The proposed temperature drop across the cooling tower is $5^{o}F$ at 500 KW, which means the secondary flow rate will be 685 gpm.

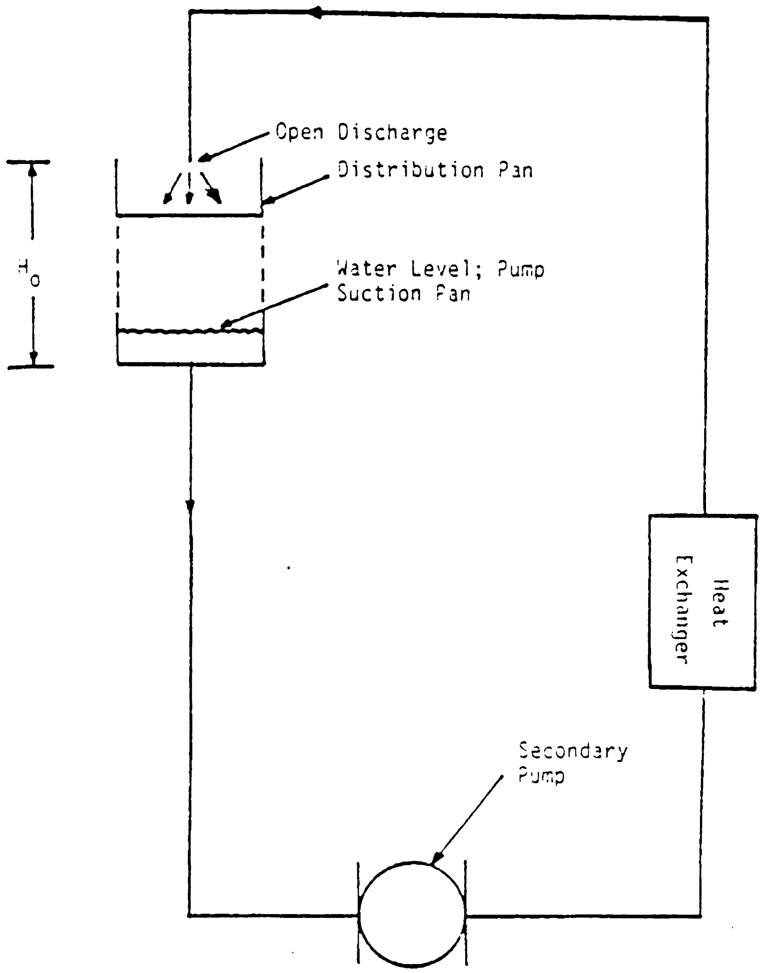

Figure 3-1. "Open" Tower Piping Circuit

Flow rate = 685 gpm = 1.53 ft^3/sec

Flow area = $\frac{\pi d^2}{4}$ (d = 6.025 in.) = 0.2 ft^2

Velocity = $\frac{\text{Flow rate}}{\text{Flow area}}$ = 7.63 ft/sec.

(i) Flow-friction loss in the piping

For the 16 ft of the piping, the flow rate is 342.5 gpm (v_1 = 3.82 ft/sec)

$$Re_1 = \frac{v_1 d}{\nu}$$

$$d = 0.5 \text{ ft}$$

$$v_1 = 3.82 \frac{ft}{sec}$$

$$\nu = 0.736 \times 10^{-5} \frac{ft^2}{sec} \text{ @ } T = 95^oF \simeq 100^oF$$

$$Re_1 = 2.60 \times 10^5.$$

Assuming that the piping is of similar roughness to that of commercial steel:

$$\frac{e}{d} = .0003 \text{ [20]} \qquad Re_1 = 2.60 \times 10^5.$$

The friction factor, f_f, can be determined from the Moody curve [20] as:

$$f_{f_1} = 0.0046.$$

The pressure loss can be calculated with the following equation:

$$h_{L_1} = 2f_{f_1} \frac{L_1}{d} \frac{v_1^2}{g_c}$$

$$= 0.13 \text{ ft}.$$

For the remainder of the piping (L_2 = 534 ft), the flow rate is 685 gpm (v_2 = 7.63 ft/sec).

$$Re_2 = \frac{v_2 d}{\nu} = 5.72 \times 10^5$$

$$f_{f_2} = 0.0044$$

$$h_{L_2} = 2f_{f_2} \frac{L_2}{d} \frac{v_2^2}{g_c}$$

$$= 17 \text{ ft}$$

(ii) Friction Loss in Fittings

The number of 90^o elbows (flow rate = 685 gpm) is 23. The friction loss can be calculated with the following equation:

$$h_{L_3} = 23 \frac{kv_2^2}{2g_c} \; ; \; k = .82 \text{ for } 90^o \text{ elbow}$$ [10]

$$h_{L_3} = 17.08 \text{ ft.}$$

The number of 90^o elbows (flow rate = 342.5 gpm) is 4.

$$h_{L_4} = 4 \frac{kv_1^2}{2g_c}$$

$$= 0.74 \text{ ft.}$$

The number of 45^o elbows is 5.

$$h_{L_5} = 5 \frac{kv_2^2}{2g_c} \; ; \; k = 0.35 \text{ for } 45^o \text{ elbow}$$ [10]

$$h_{L_5} = 1.58 \text{ ft.}$$

The number of gate valves (flow rate = 685 gpm) is 4.

$$h_{L_6} = 4\,\frac{kv_2^{\,2}}{2g_c} \qquad k = 0.16 \text{ for gate valve}$$ [10]

$$h_{L_6} = 0.58 \text{ ft.}$$

The number of gate valves (flow rate = 342.5 gpm) is 2.

$$h_{L_7} = 0.07 \text{ ft.}$$

The number of Tee's is 1.

$$h_{L_8} = \frac{kv_2^{\,2}}{2g_c} \qquad k = 0.4 \text{ for Tee}$$ [10]

$$h_{L_8} = 0.36 \text{ ft.}$$

The number of butterfly valves is 3.

For a wide open 6 in. butterfly valve, a flow rate of 1150 gpm will create a pressure drop of 2.3 ft[21]. Since the flow-friction loss changes as the square of the flow changes, the friction loss, h_{L_9}, at 685 gpm will be 3 x 2.3 x $(685/115^{o})^2$ = 2.46 ft.

(iii) Pressure Loss Through Heat Exchanger

$h_{L_{10}}$ = 12.78 psi = 29 ft. (according to the information supplied by the heat exchanger manufacturer)

$$\text{Total friction loss } h_L = \sum_{i=1}^{10} h_{L_i} = 69 \text{ ft.}$$

Static Head H_o = 10 ft

Required Pump Head = $h_L + H_o$ = 79 ft.

Having determined the required pump head of the design flow rate, the pump manufacturer can now supply a pump based on this information. The specifications of the pump are summarized in Appendix C.

(2) Net Positive Suction Head (NPSH)

The NPSH is the difference in absolute pressure measured at the center line of the suction nozzle and the lowest pressure that exists in the liquid passage within the pump before the liquid is acted upon by the pump. This parameter is important in order to prevent cavitation (formation of vapor bubbles) within the pump during operation. If P_1 is the inlet pressure to the pump, cavitation is prevented when [10]

$$P_1 - \mathrm{NPSH} > P_v \qquad \text{(B-1)}$$

where P_v is the liquid vapor pressure at the pumping temperature.

The pump inlet pressure P_1 is calculated as follows: The schematic diagram of the piping run from the cooling tower to the secondary pump is shown in Figure B-2.

As seen in Figure B-2, the pump inlet pressure P_1 can be written as:

$$P_1 = P_a + H_o - h_L. \qquad \text{(B-2)}$$

The total length of pump suction piping is 197.25 ft.

$$h_{L_1} = 2f_f \frac{L}{d} \frac{v^2}{g_c}$$

$$f_f = 0.0044$$

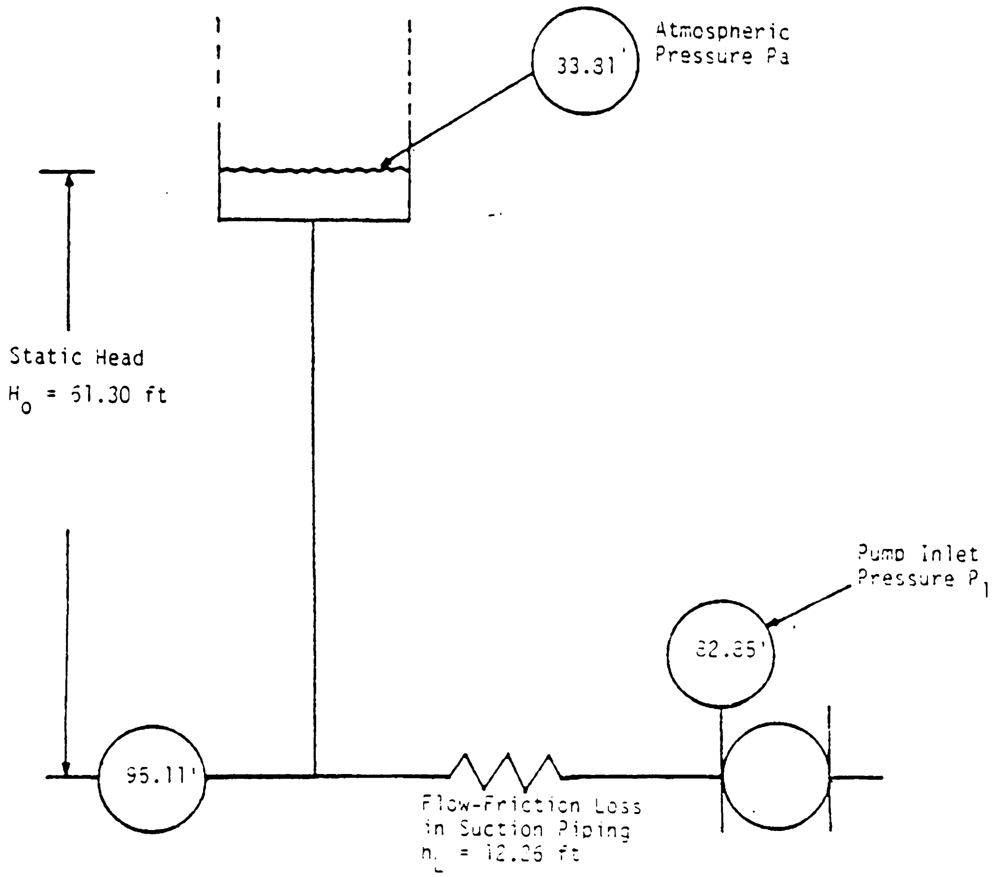

Figure 3-2. Pump Suction Pressure

$$L = 197.25 \text{ ft}$$

$$d = 0.5 \text{ ft}$$

$$V = 7.83 \frac{\text{ft}}{\text{sec}} \text{ (flow rate = 685 gpm)}$$

$$h_{L_1} = 6.29 \text{ ft.}$$

The number of 90^o elbows is 7.

$$h_{L_2} = 7 \frac{kv^2}{2gc}$$

$$k = 0.82$$

$$h_{L_2} = 5.20 \text{ ft.}$$

The number of 45^o elbows is 2.

$$h_{L_3} = 2 \frac{kv^2}{2g_c}$$

$$k = 0.35$$

$$h_{L_3} = 0.63 \text{ ft.}$$

The number of gate valves is 1.

$$h_{L_4} = \frac{kv^2}{2g_c}$$

$$k = 0.15$$

$$h_{L_4} = 0.14 \text{ ft}$$

$$h_L = \sum_{i=1}^{4} h_{L_i} = 12.26 \text{ ft}$$

$$Pa = 14.7 \text{ psi} = 33.81 \text{ ft}$$

$$H_o = 61.3 \text{ ft.}$$

Substituting h_L, Pa and H_o in Eq. (B-2) gives

$$P_1 = 82.85 \text{ ft}$$

The liquid vapor pressure, P_v, is 1.68 ft at the pumping temperature of $95^{o}F$, and the value of NPSH is 7 ft according to the information supplied by the pump manufacturer. By substituting P_1, P_v and NPSH in Eq. (B-1), it is seen that the inequality is easily satisfied. Therefore, cavitation will not occur within the pump during operation.

Appendix C

SPECIFICATIONS OF THE MAJOR COMPONENTS OF THE COOLING SYSTEM

Primary Cooling Pump

Manufacturer:	Crane Co., Chempump Division
Model No.:	GC-1½K-152H-LS
Flow Rate:	100 GPM with 40 ft head
Impeller Diameter:	4-1/8 in.
Volts:	230/460, 3 phase A.C. 60 Hz, 3450 RPM

Secondary Cooling Pump

Manufacturer:	Carotek Inc., Allis-Chalmers Industrial Pump Division
Model No.:	CSO
Flow Rate:	750 GPM with 100 ft head
Impeller Diameter:	6.4 in.
Volts:	60/460, 3 phase A.C. 60 Hz, 3600 RPM

Cooling Tower

Manufacturer:	Marley Cooling Tower Co.
Flow Rate:	375,000 lb/hr
Inlet Temperature:	$100^{o}F$
Outlet Temperature:	$95^{o}F$
Fluid Circulation:	(PH 6.0) Potable Water
Gravity:	1.0
Specific Heat:	1.0 Btu/lbm ^{o}F

Plenum Reserve:	1600 gallons
Heat Exchanged:	1,875,000 Btu/hr
Operating Pressure:	Atmospheric

Intermediate Heat Exchanger

Manufacturer:	Alfa-Laval Inc.
Type:	Counterflow
Model No.:	AM10HBM

	Primary Side	Secondary Side
Flow Rate, gpm	90	685
T-in, ^{o}F	149	95
T-out, ^{o}F	111	100

Effective Heat-Transfer Area:	92.57 ft^2
Operating Pressure	100 psi
Plate Material	AISI 304 SS
No. of Plates	22
Heat Load	1,706,500 Btu/hr

Flowmeter System

Flow Sensor

Manufactor:	Horizon Ecology Co.
Type:	Paddle Wheel
Model No.:	C-5618-10
Flow Range:	1 ft/sec minimum, 50 ft/sec maximum

Flow Meter

Manufacturer:	Horizon Ecology Co.
Type:	Digital averaging display (D.A.D.)
Model No.:	5622-35-VH

Installation Fittings

Manufacturer:	Horizon Ecology Co.
Type:	316 Stainless Steel Tee
Model No.:	C-5620-45
Length:	4-3/4 in.

Throttle Valve

Manufacturer:	Keystone International Inc., Keystone Valve Division
Type:	Butterfly
Size:	6 in.

Primary Cooling Piping

Material:	Aluminum
Type:	6061-T6
Schedule:	80
O.D.:	2.875 in.
I.D.:	2.323 in.

Appendix D

UNEVEN POWER SPLIT IN THE CORE TANKS

E. Stam has made various flux measurements for the 10 KW VPI&SU reactor[6]. One of the interesting results is that the average flux in the East core is considerably lower than that in the West core. This result is probably due to the fact that the shim rod is half withdrawn from its position adjacent to the East core during reactor operation. (see Figure 1.) The relative flux distributions in the two cores are shown in Figure D-1.

As can be seen in Figure D-1, the averages in the West and East cores are in the ratio of 85.7 to 75.5 or 1 to 0.88. This means that average flux in the West core is approximately 16% higher than that in the East core. Since the reactor power is proportional to the flux, the average power level in the West core is 16% higher than that in the East core. Accordingly the fuel assembly that has the highest average temperature is located in the core tank position W-4.

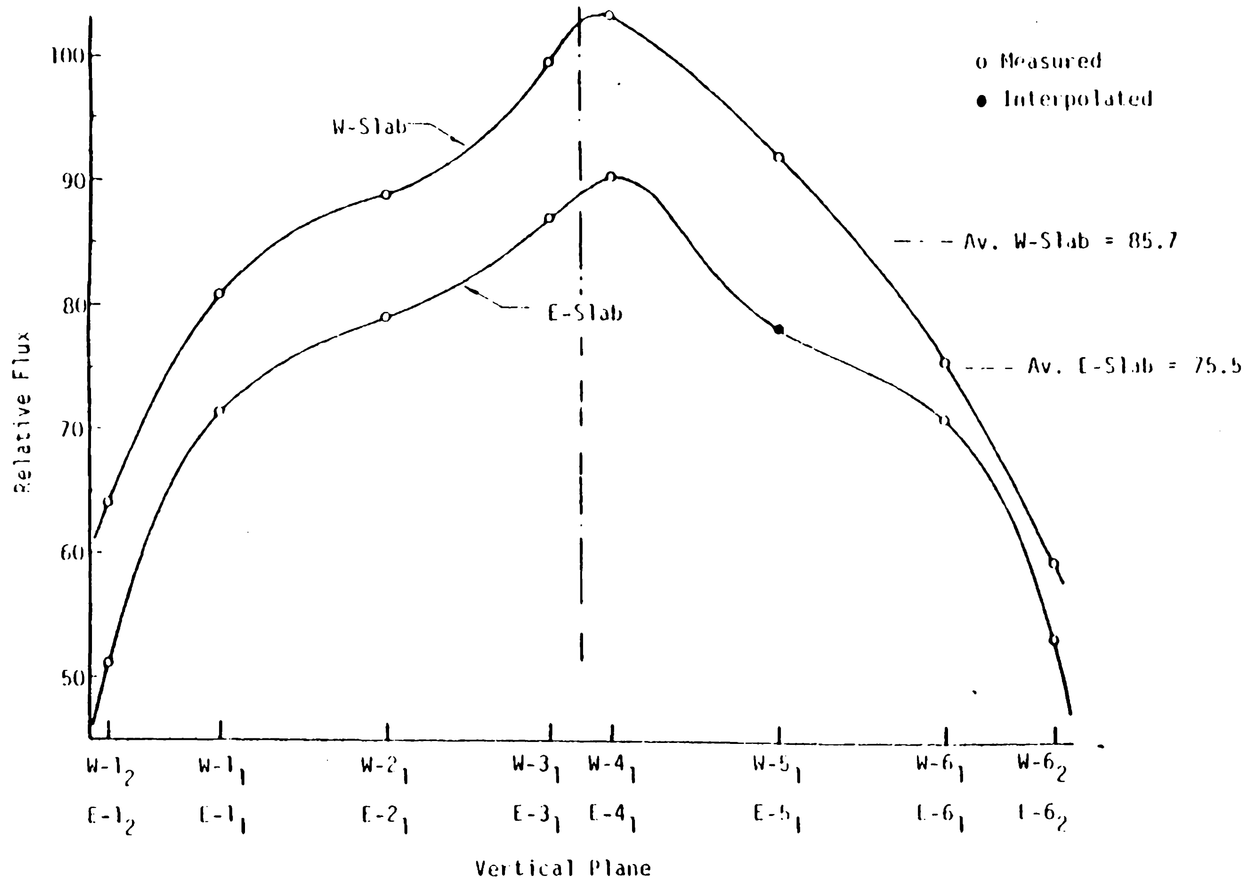

Figure D-1. Relative Flux Distributions in the Core Tanks (from Ref. 8)

Appendix E

SPECIFICATIONS OF 12- AND 13-PLATE FUEL ASSEMBLIES

As previously mentioned, the thirteen-plate elements will be needed for the 500 kw operation. However, at the start of the operation only twelve-plate elements will be loaded in the core. These elements will then be removed from the core gradually and replaced with fully loaded thirteen-plate elements to compensate for fuel burnup and fission product poison buildup.

The design specifications for the twelve- and thirteen-plate elements are listed in Table E-1.

Table E-1

DESIGN SPECIFICATIONS FOR THE TWELVE- AND THIRTEEN-PLATE ELEMENTS

	Twelve-Plate Elements	Thirteen-Plate Elements
Fuel Material:	UAl_4	UAl_4
Weight of U-235 per Plate:	22 grams	22 grams
Plate Length:	26 in.	26 in.
Plate Width:	3 in.	3 in.
Plate Thickness:	0.08 in.	0.08 in.
Water Gap:	0.4 in.	0.36 in.
Active Dimensions:	23 in. x 2.75 in. x 0.04 in.	23 in. x 2.75 in. x 0.04 in.
Total Weight of U-235:	264 grams	286 grams

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A THERMAL-HYDRAULIC ANALYSIS OF THE COOLING SYSTEM FOR THE 500 KW VIRGINIA POLYTECHNIC INSTITUTE AND STATE UNIVERSITY REACTOR

by

Ai-Tai Lo

(ABSTRACT)

In order to increase the usefulness of the nuclear research reactor, it is necessary to enhance the neutron fluxes available in the experimental facilities. One method of attaining this objective is to increase the power level of the reactor.

The VPI&SU 100 KW research reactor has been studied to investigate the feasibility of operation of 500 KW. From a thermal-hydraulic point of view, the following limiting conditions should not be exceeded for a 500 KW operation:

1) bulk boiling of the coolant should not occur,
2) surface boiling should not occur on any fuel plate,
3) fuel meltdown must not occur after a loss-of-flow or loss-of-coolant accident,
4) fuel plate vibration problems should not be significant.

The results of the investigation indicate that the above limiting conditions will not be exceeded provided that certain system modifications are made. Thus the operation of the VPI&SU reactor at 500 KW is feasible as far as thermal-hydraulics is concerned.