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Heat Exchanger Design Study for the ISHR

I. GENERAL DISCUSSION This memorandum presents the results of a study to determine reliable heat transfer data for the design of a suitable heat exchanger for the ISHR. It is desirable that a heat exchanger to remove energy from the core of a circulating fuel homogeneous reactor shall be capable of generating steam at sufficient pressure to be attractive as a power producer. However, if possible, such an exchanger should also be designed to function as a cooler when operating the reactor at low temperature in order to provide for the contingency that the corrosion rate of UO_2SO_4 solution on Type 347 stainless steel at high temperatures and high concentration may be beyond the range of tolerance. Thus, the reactor may still be made to function as a plutonium producer or thorium converter even though the temperature of operation may be too low to function also as a power producer.

In general, it has been postulated that the requirements to be met in the design of a heat exchanger for the ISHR shall be as listed below:

(1) Reactor power	48,000 KW
(2) Reactor solution flow rate	5,000 GPM
(3) Reactor solution conditions	250°C at 1000 psia
(4) Allowable tube-side pressure drop	20 psi
(5) Tube-side corrosion allowance	~0.030 in.
(6) U-Tube, all-welded construction	1/2 in. O.D. - 16 BWG tubes
(7) Tube-side material	Type 347 stainless steel
(8) Shell-side material	Carbon steel
(9) Tube-side design pressure & temperature	1000 psi at 550°F

- (10) Shell-side design pressure & temperature   600 psi at 500°F
- (11) Shell-side corrosion allowance   1/16 inch
- (12) Code requirements: ASME 1950 on Shell; 1949 Par. U-68 on tube side, except that allowable stress on tube side shall not exceed stress-corrosion range of 10,000 psi.
- (13) The maximum allowable flux for the vaporization of water using forced or natural circulation shall not exceed 30,000 Btu/(hr)(ft)² to preclude the possibility of vapor blanketing.

A study is currently being made by Leon F. Goode of the development group for the design of an exchanger to remove 200,000 KW at 20,000 GPM tube side flow from a horizontal tube evaporator. This study may be compared to the one presented herein for vertical tube evaporators in order to aid in evaluating the relative merits of a horizontal versus a vertical tube heat exchanger for future large scale homogeneous circulating fuel type reactors.

Two basic mechanisms of heat removal from the outer surfaces of the tubes are considered.

- (A) Sensible heat from the fuel solution is transferred by convection and conduction to the outer tube wall surface and, hence, to the shell side fluid by nucleate boiling. With this mechanism, the outer wall heat transfer coefficient is a function of the temperature difference between the tube wall and the boiling liquid and is affected by such variables as (1) the nature of the surface (2) the absolute pressure of the boiling liquid (3) the arrangement of tubes (vertical or horizontal) and (4) the amount of agitation.
- (B) Heat is transferred from the outer tube wall surface to the shell

side fluid by the process of convection. With this assumption, the outer tube-wall heat transfer coefficient may be predicted by the equation of Colburn and is substantially independent of the temperature difference between the tube wall and the fluid.

II.(3) PROPERTIES OF REACTOR FUEL SOLUTION - The tube side heat balance and magnitude of the inside tube heat transfer coefficient are evaluated from the following fuel solution properties obtained from CF-52-1-124 (Some Physical Properties of UO_2SO_4 - D_2O Solutions by R. Van Winkle to File, Jan. 18, 1952.)

Fuel solution	UO_2SO_4 - D_2O
Uranium concentration	250 gm/l
* Temperature	466°F
Density of fuel solution	$\begin{cases} 75 \text{ lb/ft}^3 \\ 1.2 \text{ gm/cc} \end{cases}$
Specific heat	0.84 Btu/(lb)(°F)
Viscosity	0.42 lb/(Hr)(ft)
Thermal conductivity	~ 0.35 Btu/(hr)(ft ²)(°F/Ft)

III.(4) PROPERTIES OF WATER AT ELEVATED TEMPERATURES - The shell side exchanger characteristics are evaluated by means of graphs prepared from the following table:

Temperature °F	Density lb/ft ³	Viscosity lb/(hr)(ft)	Therm. Cond. Btu/(hr)(ft ²)(°F/ft)	Specific Heat Btu/(lb)(°F)
100	62.2	1.640	0.363	1.00
200	60.2	0.740	0.394	1.01
300	57.5	0.450	0.396	1.03
400	53.6	0.330	0.380	1.07
450	51.2	0.270	0.369	1.10
500	48.8	0.220	0.356	1.13

* Mean temperature of fuel solution in exchanger

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IV. THE TUBULAR ELEMENT - The analysis of the evaporator for both the mechanisms of nucleate boiling from a pool and heat transfer by forced convection are predicated on the arbitrary selection of 1/2 inch outside diameter tubes. Although the efficiency of a heat exchanger varies inversely as the diameter of tubes used, there is, apparently, a limit below which manufacturers of equipment do not consider it practical to go. The A. O. Smith Manufacturing concern found it very difficult to expand and weld 1/4 inch O.D. tubes into the tube-sheet of the heat exchanger for the HRE. The Lummus Company recommends the use of 1/2 inch O.D. tubes for the proposed large scale heat exchangers.

The thickness of the tube walls must be such as to withstand a working pressure of 1000 psi at not more than 10,000 psi stress. From Section VIII of the ASME Code for Unfired Pressure Vessels, the required wall thickness for ferrous tubes and pipe used as tubes shall be determined by:

$$t = \frac{D_o}{2.3S} (P + 0.036S)$$

where,   P = Maximum allowable working pressure, psi
                  t = Minimum wall thickness, inches
                  S = Maximum allowable working stress, psi
                  D_o = Outside diameter of tube, inches

Therefore, based on a maximum steady state working pressure of 1000 psi, the minimum wall thickness of the 1/2 inch O.D. tube must be 0.0296 inches. Since a corrosion allowance of ~0.030 inches is desired, the tube wall thickness used is 16 BWG (0.065 inches) which leaves a corrosion allowance of 0.035 inches.

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V. DESIGN STUDY OF A VERTICAL SUBMERGED TUBE EVAPORATOR WITH NUCLEATE

BOILING FROM THE OUTER TUBE SURFACES - Figure 1 shows schematically the arrangement of the shell and steam drum of a vertical tube evaporator to remove 50,000 KW of heat from fuel solution flowing through the tubes at 5000 GPM and an entrance temperature of 482°F by the mechanism of nucleate boiling from the outer tube surfaces.

A. Steam Generation - Assume that the steam produced is used to generate power in a turbo-generator exhausting to a condenser at 2 inches Hg. and is returned to the system at a temperature corresponding to this pressure less approximately 3°F hotwell depression or 98.2°F. Hence, the net steam generated is,

$$\frac{1.71 \times 10^8}{1199.8 - 66.2} = 1.51 \times 10^5 \text{ lb/hr}$$

B. Preheat of Feedwater - Feedwater is assumed fed to the evaporator shell at a saturation temperature corresponding to steam pressure of 215 psia or 387.9°F. It is assumed that the condensate return will be preheated in the steam drum by a portion of the gross steam generated condensing on preheater coils. At an average specific heat of 1.01 Btu/(lb)(°F), the preheat required is,

$$q_p = (1.51 \times 10^5)(1.01)(387.9 - 98.2) = 4.42 \times 10^7 \text{ Btu/hr}$$

At 215 psia, the heat of vaporization of steam is 837 Btu/lb, hence, the evaporator must vaporize 5.28×10^4 lb/hr in excess of the net steam generated, or a total of 2.04×10^5 lb/hr.

C. Evaporator Surface Area - In the process of evaporation from a pool of boiling water heated by submerged surfaces the heat transfer coefficient from the outside of the tubes is a function of the difference between the

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tube wall surface and the bulk temperature of the boiling water. When a hot fluid flows through the tubes and gives up its sensible heat, the temperature of the fluid and, hence, of the tube wall will diminish through the evaporator. Therefore, the over all coefficient of heat transfer will vary along the tube length and the ordinary method of calculating the required surface area by means of a constant coefficient and a logarithmic mean temperature difference will not suffice. However, since the terminal conditions are known, it is possible to solve for the required surface area by integration as explained in CF-49-8-231.

D. Coefficients of Heat Transfer

(a) Film coefficient in the tubes - This coefficient may be determined with sufficient accuracy by the equation

$$\frac{h_1 D}{k} = 0.023 \left(\frac{DG}{\mu} \right)^{0.8} \left(\frac{C_p \mu}{k} \right)^{0.3}$$

where, h_1 = Inside film coefficient, Btu/(hr)(ft²)(°F)
 k = Thermal conductivity of fluid, Btu/(hr)(ft²)(°F/ft)
 D = Inside tube diameter, Ft
 G = Mass velocity, Lb/(hr)(ft²)
 μ = Viscosity of fluid, Lb/(hr)(ft)
 C_p = Specific heat, Btu/(lb)(°F)

Using the physical properties of the reactor solution at the mean temperature of the film and since the tube diameter is constant, this equation reduces to

$$h_1 = 1.05 v^{0.8} \text{ Btu/(hr)(ft}^2\text{)(°F)}$$

(b) Scale coefficients

(1) Klotz has estimated the value of the scale deposit in the tubes as 4000 Btu/(hr)(ft²)(°F)

(2) McAdams recommends a value of 2000 Btu/(hr)(ft²)(°F) as the outer scale value to distilled water.

(c) Wall coefficient - The thermal conductivity of Type 347 stainless steel is taken as 130 Btu/(hr)(ft²)(°F/in) hence, for 16 BWG (0.065 in. w.t.) the wall coefficient is, $\frac{130}{0.065} = 2000 \text{ Btu/(hr)(ft}^2\text{)(°F)}$

(d) Coefficient from outer tube surface - Figure 2 is a reproduction of data from Figures 156 and 157 of Heat Transmission by McAdams for water boiling at 1 atmosphere in pools heated by vertical and horizontal surfaces. This figure also shows the data of Gammel and Adams for water at 212°F and atmospheric pressure flowing normal to a horizontal platinum wire at a velocity of 15.5 ft/sec. At a constant rate of heat transfer, this film coefficient increases as the 0.25 to 0.4 power of the absolute pressure (see Vol. II of Heat Transfer Lecture, p. 150, NEPA-979-IER-13). Assuming the lower value, the heat transfer coefficient at 215 psia may be determined from

$$h = h_{\text{atm}} \left(\frac{P}{P_{\text{atm}}} \right)^{0.250}$$

Applying this correction factor, the solid curve of Figure 6 for the heat transfer coefficient from a vertical tube to water boiling at 215 psia was obtained. Between a Δt of about 10°F to 40°F, the curve on logarithmic paper is linear and the coefficient may be calculated from the equation:

$$h_o = 3.80 \Delta t^{2.07}$$

E. Equation for Heating Surface - From this data, using the method of CF-49-8-231, the required evaporator surface may be calculated from

$$A = WC_p \left[0.001413 \ln \frac{(Q/A_o)_2}{(Q/A_o)_1} + \frac{1}{0.776V^{0.8}} \ln \frac{(Q/A_o)_2}{(Q/A_o)_1} - 0.314 \left\{ (Q/A_o)_2^{-0.674} - (Q/A_o)_1^{-0.674} \right\} \right]$$

where, A = Tube outer surface area, ft^2
               W = Weight rate of flow through tubes, lb/hr
               C_p = Specific heat of fluid in tubes, $\text{Btu}(\text{lb})(^\circ\text{F})$
               $(Q/A_o)_2$ = Heat flux at hot end, $\text{Btu}/(\text{hr})(\text{ft}^2)$
               $(Q/A_o)_1$ = Heat flux at cold end, $\text{Btu}/(\text{hr})(\text{ft}^2)$
               V = Velocity of fluid through tubes, Ft/hr

F. Required surface areas and pressure drops in a vertical submerged tube evaporator boiling water at 215 psia when dissipating 50,000 KW of power from the reactor solution at a constant flow of 5000 GPM through the tubes.

Flow Velocity thru tubes (Ft/sec)	Area ft^2	No. tubes	Av. length of tubes Ft.	Pressure drop thru tubes psi
10	8350	1490	42.8	18
15	7900	992	60.8	52
20	7820	745	80.3	114

VI. DESIGN STUDY OF A VERTICAL TUBE, FORCED CIRCULATION EVAPORATOR

Figure 3 is a schematic diagram of a vertical U-tube forced convection evaporator arrangement to remove 50,000 KW of heat from fuel solution flowing at 5,000 GPM through the tubes. The procedure used to arrive at the surface area, baffle spacing and shell dimensions as shown by Figure 4 is essentially that described in Chapter 15 of Process Heat Transfer by D. Q. Kern.

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With this arrangement the inside film coefficient, wall coefficient and scaling factors are treated as in the previous case of pool boiling. However, the isothermal boiling film coefficient is obtained by the equation of Colburn based on the premise that the heat must first be absorbed by the liquid by forced convection before passing into the vapor bubbles and that the liquid heating coefficient is the controlling coefficient in the sequence.

A. Heat Balance

(1) Tube side:

$$\text{Heat removed} = (5 \times 10^4)(3413) = 1.71 \times 10^8 \text{ Btu/hr}$$

$$\text{Exit temperature} = 482 - \frac{1.71 \times 10^8}{(3.01 \times 10^6)(0.84)} = 418.2^\circ\text{F}$$

(2) Shell side:

Steam is assumed to be separated in the drum at 215 psia ($t_{\text{Sat}} = 387.9^\circ\text{F}$) and subsequently used in a condensing turbine exhausting at 2 inches Hg. ($t_{\text{exh}} = 101.2^\circ\text{F}$). Allowing 3° hotwell depression, the temperature of condensate return will be 98.2°F . This must be preheated to saturation temperature of 387.9°F . The heating surface in the evaporator shell is, therefore, divided into a preheating zone and vaporizing zone.

$$\text{Steam generated} = \frac{1.71 \times 10^8}{(1199 - 66.2)} = \frac{1.51 \times 10^5 \text{ lb/hr}}{}$$

In order to obtain relatively high velocity of water over the tubes for good heat transfer coefficients and to prevent excessive fouling of the tubes, only 20% of the feed entering the evaporator will be assumed to be vaporized. Since the steam produced amounts to $1.51 \times 10^5 \text{ lb/hr}$, the feed entering the shell is,

$$\frac{1.51 \times 10^5}{0.20} = \frac{7.55 \times 10^5 \text{ lb/hr}}{}$$

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the difference,

$$(7.55 \times 10^5) - (1.51 \times 10^5) = 6.04 \times 10^5 \text{ lb/hr}$$

is supplied by recirculation. The temperature of the feed entering the evaporator will, therefore, be

$$q = \frac{(6.04 \times 10^5)(387.9 - 32) + (1.51 \times 10^5)(98.2 - 32)}{7.55 \times 10^5} = 299 \text{ Btu/lb}$$

$$\text{and, } t = 299 + 32 = 331^\circ\text{F}$$

B. Mean Temperature Difference (Subscripts p and v indicate preheating and vaporizing).

Figure 5 is the temperature - enthalpy diagram of the complete evaporator showing the temperatures of the tube-side and shell-side fluids respectively in the preheat and vaporization zones.

Figure 5

Enthalpy

$$q_p = 4.60 \times 10^7 \text{ Btu/hr}$$

$$q_v = 1.25 \times 10^8 \text{ Btu/hr}$$

$$Q = 1.71 \times 10^8 \text{ Btu/hr}$$

LMTD in Preheat Zone

$$\left\{ \begin{array}{l} \Delta t_p = (\text{LMTD})_p (F_t) \\ R = \frac{T_1 - T_2}{t_2 - t_1} = 0.302 \\ S = \frac{t_2 - t_1}{T_1 - t_1} = 0.377 \end{array} \right.$$

$$F_t = 0.99$$

$$(\text{LMTD})_p = \frac{(482-331) - (464.8-387.9)}{2.3 \log \frac{(482-331)}{(464.8-387.9)}} = 109^\circ\text{F}$$

$$\Delta t_p = (0.99)(109) = 108^\circ\text{F}$$

LMTD in vaporizer zone:

$$(\text{LMTD})_v = \frac{(464.8-387.9) - (418.2-387.9)}{2.3 \log \frac{(464.8-387.9)}{(418.2-387.9)}} = 50^\circ\text{F}$$

$$R = \frac{464.8-418.2}{387.9-387.9} = \infty$$

$$S = \frac{387.9-387.9}{464.8-387.9} = 0$$

$$F_t = 1$$

$$\Delta t_v = (1)(50) = 50^\circ\text{F}$$

Weighted Δt :

$$\frac{q_p}{(\Delta t_p)} = \frac{4.60 \times 10^7}{108} = 4.27 \times 10^5 \text{ Btu/(hr)}(^{\circ}\text{F})$$

$$\frac{q_v}{(\Delta t_v)} = \frac{1.25 \times 10^8}{50} = 2.50 \times 10^6 \text{ Btu/(hr)}(^{\circ}\text{F})$$

$$\sum \frac{q}{\Delta t} = 2.927 \times 10^6 \text{ Btu/(hr)}(^{\circ}\text{F})$$

$$\Delta t \text{ (weighted)} = \frac{1.71 \times 10^8}{2.927 \times 10^6} = \underline{58.4^{\circ}\text{F}}$$

C. Coefficients of Heat Transfer:

- (1) Film coefficient in tubes

$$h_i = 1.05 v^{0.8} \text{ Btu/(hr)}(\text{ft}^2)(^{\circ}\text{F})$$

(2) Fouling coefficient. The scaling factors on the inside and outside of the tubes are lumped into one resistance equal to $R = 0.001$.
 $(^{\circ}\text{F})(\text{ft}^2)(\text{hr})/\text{Btu}$

- (3) Wall coefficient

$$h_w = 2000 \text{ Btu/(hr)}(\text{ft}^2)(^{\circ}\text{F})$$

(4) Outside tube surface coefficient. This coefficient is evaluated from the equation of Colburn.

$$\frac{h_o D_o}{k_f} = 0.33 \left[\frac{(C_p \mu_f)^{1/3}}{k_f} \right] \left[\frac{(D_o G)^{0.6}}{\mu_f} \right]$$

To allow for the effect of the leakage that occurs around the baffles in cross-baffled exchangers, one assumes no leakage, computes the corresponding coefficient h_o from the above equation, and multiplies h_o by the factor 0.6.

Coefficients calculated from this equation for tube side velocities ranging from 1 to 15 ft/sec are plotted on Figure 6 with the velocity as the parameter and showing the outside forced convection film coefficient as a function of Δt .

These curves show that below a Δt at which nucleate boiling from the tube surface predominates, the effect of Δt on the isothermal boiling coefficient at given velocity is negligible. Above this Δt , the outside heat transfer coefficient must be obtained as in pool boiling from a submerged surface. Gamel and Adams have substantiated this conclusion experimentally as shown by their data on Figure 2. The two large solid dots on this curve were calculated by the Colburn equation and indicate excellent correlation with the experiment. The slight slope of the curve in the forced convection region is explained by the difference in physical characteristics of the film at the different Δt 's. Based on this data, the clean overall coefficients for preheating and vaporizing are given in the following table:

Clean Overall H.T. Coefficients

Velocity over tubes Ft/sec	U_c in Preheat Zone			U_c in Vaporizer Zone		
	Velocity thru tubes Ft/sec			Velocity thru tubes Ft/sec		
	10	15	20	10	15	20
2	577	604	621	577	604	621
4	687	727	750	693	733	756
6	749	797	823	754	803	831
8	790	843	873	797	850	882
10	817	873	906	828	885	922

$$U = \text{Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$$

The effect of an assumed fouling factor = 0.001 is to reduce the values of the clean overall coefficients to the values given by the following table which have been used to arrive at the required surface area of the evaporator.

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Fouled Overall H.T. Coefficients

Velocity over tubes Ft/sec	U _o in Preheat Zone			U _o in Vaporizer Zone		
	Velocity thru tubes - ft/sec			Velocity thru tubes - ft/sec		
	10	15	20	10	15	20
2	366	376	398	366	377	383
4	408	422	430	410	423	430
6	428	443	451	430	446	455
8	441	457	467	443	460	469
10	451	465	476	453	470	481

D. Surface Areas - Based on the equation:

$$A = \frac{Q}{U \Delta t} \quad (\text{Ft}^2)$$

Clean Surface Area Required - Ft²

Velocity over tubes ft/sec	Clean preheat and vaporizer surface						Total clean surface		
	Velocity thru tubes ft/sec						Vel. thru tubes ft/sec		
	10		15		20		10	15	20
	Ap	Av	Ap	Av	Ap	Av			
2	740	4330	707	4140	688	4030	5070	4847	4718
4	623	3610	587	3420	570	3310	4233	4007	3880
6	570	3320	536	3120	519	3010	3890	3656	3529
8	541	3140	507	2940	489	2835	3681	3447	3324
10	523	3020	489	2830	472	2710	3543	3319	3182

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Fouled Surface Area Required - Ft²

Velocity over tubes ft/sec	Fouled preheat and vaporizer surface Velocity thru tubes ft/sec						Total fouled surface Vel. thru tubes - ft/sec		
	10		15		20		10	15	20
	Ap	Av	Ap	Av	Ap	Av			
2	1165	6830	1135	6630	1070	6530	7995	7765	7600
4	1045	6100	1010	5910	993	5820	7145	6920	6813
6	997	5820	964	5610	948	5500	6817	6574	6448
8	968	5640	934	5430	913	5330	6608	6364	6243
10	947	5520	918	5320	898	5200	6467	6238	6098

E. Pressure Drop

1. Tube side - The pressure drop thru the inside of the tubes is calculated from the Fanning equation:

$$\Delta P = \frac{4f\rho V^2 L}{(144)(2)gD}$$

where,

ΔP = Pressure drop, psi

f = Friction factor

ρ = Density, lb/ft³

L = Length of tube, ft

D = Inside diameter of tube, ft

V = Velocity in tubes, ft/sec

g = Acceleration of gravity, ft/sec²

Velocity thru shell ft/sec	10 ft/sec. thru tubes			15 ft/sec thru tubes			20 ft/sec thru tubes		
	No. of tubes	Av. length Ft.	ΔP psi	No. of tubes	Av. length Ft.	ΔP psi	No. of tubes	Av. length Ft.	ΔP psi
2	1490	41	17.2	993	59.7	52.3	745	78.0	112
4	1490	36.7	15.4	993	53.2	46.6	745	69.8	100
6	1490	35.3	14.8	993	50.6	44.3	745	66.1	95.2
8	1490	33.9	14.2	993	48.9	42.8	745	64.1	92.2
10	1490	33.2	13.9	993	48.0	42.0	745	62.6	90.0

2. Shell Side - Kern, in Process Heat Transfer, states that the pressure drop through the shell of an exchanger is proportional to the number of times the fluid crosses the bundle between baffles, and to the distance across the bundle each time it is crossed. Equation (7.44) p. 147 in Process Heat Transfer gives the pressure drop of a fluid being heated or cooled in cross-flow through an exchanger, including entrance and exit losses;

$$\Delta P_s = \frac{f G_s^2 D_s (N+1)}{2 g \rho D_e \phi_s}$$

where,

ΔP_s = Pressure drop, psi

f = Shell side friction factor, ft^2/in^2 (Fig. 29 p. 839 of Process Heat Transfer)

G_s = Mass velocity, $\text{lb}/(\text{hr})(\text{ft}^2)$

D_s = Inside diameter of shell, ft

$N+1$ = Number of times fluid crosses bundle from inlet to outlet

g = Acceleration of gravity, $4.17 \times 10^8 \text{ ft/hr}^2$

ρ = Density, lb/ft^3

D_e = Equivalent diameter, Ft.

$\phi_s = (\mu/\mu_w)^{0.14} \approx 1$

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a) At 10 ft/sec & 5000 GPM thru 1/2 in. O.D. - 16 BWG Tubes

Number of tubes = 1490

Minimum inside diameter of shell = 43.7 in.

Assumed inside diameter of shell, D_s = 48 in.

	Shell Side Velocity, ft./sec.				
	2	4	6	8	10
Mass velocity, lb/(hr)(ft ²)	389,000	778,000	1,165,000		
Reynold's number	68,000	136,000	204,000		
Friction factor, ft ² /in ²	0.00148	0.0013	0.0012		
No. of bundle crosses	13	23	33		
Flow area across bundle, ft ²	1.95	0.972	0.649		
Baffle spacing, in.	17.5	8.75	5.83		
Shell side pressure drop, psi	4.2	26.3	78.0		

b) At 15 ft/sec. & 5000 GPM thru 1/2 in. O.D. - 16 BWG Tubes

Number of tubes = 993

Minimum inside diameter of shell = 35.7 in.

Assumed inside diameter of shell = 40 in.

	2	4	6	8	10
Mass Velocity, lb/(hr)(ft ²)	389,000	778,000	1,165,000		
Reynold's number	68,000	136,000	204,000		
Friction factor, ft ² /in ²	0.00148	0.0013	0.0012		
No. of bundle crosses	16	29	41		
Flow area across bundle, ft ²	1.95	0.972	0.649		
Baffle spacing, in.	21.1	10.5	7.0		
Shell side pressure drop, psi	4.32	27.6	80.8		

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c) At 20 ft/sec & 5000 GPM thru 1/2 in. O.D. - 16 BWG Tubes

Number of tubes = 745

Minimum inside diameter of shell = 31.1 in.

Assumed inside diameter of shell = 36 in.

	2	4	6	8	10
Mass velocity, lb/(hr)(ft ²)	389,000	778,000	1,165,000		
Reynold's number	68,000	136,000	204,000		
Friction factor, ft ² /in ²	0.00148	0.0013	0.0012		
No. of bundle crosses	19.5	35	49		
Flow area across bundle, ft ²	1.95	0.972	0.648		
Baffle spacing, in.	23.4	11.7	7.77		
Shell side pressure drop, psi	4.73	29.9	86.8		

d) Summary of pressure drop thru tubes & shell:

Velocity thru shell Ft/sec	Velocity thru tubes, Ft/sec					
	10		15		20	
	ΔP_T	ΔP_S	ΔP_T	ΔP_S	ΔP_T	ΔP_S
2	17.2	4.2	52.3	4.32	112	4.73
4	15.4	26.3	46.6	27.6	100	29.9
6	14.8	78.0	44.3	80.8	95.2	86.8
8	14.2		42.8		92.2	
10	13.9		42.0		90.0	

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CONCLUSIONS:

- (1) To meet the condition that the pressure drop through the tube side of the evaporator should not be greater than 20 psi., the velocity of fuel solution through the tubes should not exceed 10 ft/sec. At 10 ft/sec the tube side pressure drop ranges from 13.9 psi to 17.2 psi depending on the shell side velocity. At 15 ft/sec, the pressure drop jumps to a range of 42.0 psi to 52.3 psi.
  - (2) The allowable pressure drop through the shell side of the evaporator is unspecified. However, the study shows that the shell side pressure drop will range from around 4 psi at 2 ft/sec through the shell to around 80 psi at 6 ft/sec. At 4 ft/sec., the pressure drop is approximately 26 to 29 psi which is deemed to be a reasonable value.
  - (3) At 10 ft/sec through the tubes and 4 ft/sec over the tubes, the required surface area to remove 50,000 kw of heat at 5000 GPM flow is 7145 ft² of which 1045 ft² will be preheat area and the remainder of 6100 ft² will be vaporizing area. With these conditions, it is expected that the evaporator will generate 1.51×10^5 lb/hr of 215 psia saturated steam.
  - (4) A comparable evaporator to remove 50,000 kw of heat by the process of nucleate boiling from the tube surface at 10 ft/sec through the tubes requires 8350 ft² of surface with a pressure drop through the tubes of 18 psi. Thus, the forced circulation type of evaporator will effect a savings of 1205 ft² or 14.5% in surface area and, hence, in hold-up of fuel solution.
- VI. The suggested design shown by Figure 4 apparently fulfills the requirements of a vertical tube heat exchanger for the ISHR as listed in Section I of this memorandum. Pertinent information required for the detailed design is listed in the accompanying table of specifications.

The baffle spacing shown by this Figure is staggered to allow for the decrease in density of the water vapor mixture as it flows through the shell. The pressure drop calculations previously shown are for uniformly spaced baffles calculated on the basis of the entrance fluid density. However, for a fixed rate of flow, as the density of the mixture decreases, the velocity and hence the pressure drop will increase for a uniform baffle spacing. Therefore, the baffle spacing through the vaporizing section is increased uniformly to a maximum of 37 1/2 in. This arrangement permits a more or less uniform shell side velocity and, hence, a lower pressure drop.

Average shell-side fluid density in vaporizer section

- (1) Density of exit mixture:

$$\rho = \frac{(1.51 \times 10^5) + (6.04 \times 10^5)}{(2.13)(1.51 \times 10^5) + (0.0185)(6.04 \times 10^5)} = 2.28 \text{ lb/ft}^3$$

- (2) Average density

$$\rho_{\text{avg}} = \frac{54.1 + 2.28}{2} = 28.19 \text{ lb/ft}^3$$

Corrected shell-side pressure drop: The pressure drop is computed in this case by using the mass velocity and Reynold's number based on the liquid properties at the vaporizer inlet, and the mean density between the inlet and outlet.

Pressure drop thru preheat zone = 4.68 psi

Pressure drop thru vaporizer zone = 15.70 psi

Total corrected pressure drop = 20.38 psi

EVAPORATOR SPECIFICATION SHEET

Performance Data

	Shell side	Tube side
Fluid circulated	Distilled water	Solution
Total fluid entering	7.55×10^5 lb/hr	5000 GPM (3.01×10^6 lb/hr)
Fluid vaporized	1.51×10^5 lb/hr	
Fluid properties		
Density	54.1 lb/ft ³ *	75 lb/ft ³
Viscosity	0.35 lb/(hr)(ft)*	0.42 lb/(hr)(ft)
Specific heat	1.06 Btu/(lb)(°F) *	0.84 Btu/(lb)(°F)
Thermal Conductivity	0.383 Btu/(hr)(ft ²)(°F/ft)*	0.35 Btu/(hr)(ft ²)(°F/ft)
Temperature in	482 °F	331 °F
Temperature out	418.2 °F	387.9°F
Operating pressure	1000 psi	215 psia
Number of passes	1	2
Velocity	4 ft/sec	10 ft/sec
Pressure drop	20.4 psi	15.4 psi
Fouling resistance	0.0005	0.0005
Heat exchanged	50,000 kw (1.71×10^8 Btu/hr)	
Transfer rate (weighted)	410 Btu/(hr)(ft ²)(°F)	59.3% of clean rate
Weighted Δt	58.4 °F	
Heat transfer area	7145 ft ²	
Max. heat flux in vaporizer zone	20,500 Btu/(hr)(ft ²) Less than 30,000	

* Corresponds to liquid properties at 387.9°F

CONSTRUCTION DATA

	Shell side	Tube side
Design pressure	600 psi	1000 psi
Test pressure	900 psi	2000 psi
Design temperature	500°F	550°F
<u>Tubes:</u> 347 stainless No. 1490 O.D. 1/2 in. BWG - 16 Av. length 36.7 ft Pitch - 3/4 in. triangular		
<u>Shell:</u> Welded steel A-212-B SR & XR I.D. 46" O.D. 48" Thickness 1"		
<u>Shell cover:</u> Steel A-212-B, welded to shell		
<u>Bonnet:</u> 347 stainless steel SR & XR		
<u>Tube sheet:</u> 347 stainless steel 9 in. minimum thickness reinforced for 2980 - 1/2" holes drilled thru		
<u>Baffles:</u> Segmental, 8 3/4 to 37 1/2 in. staggered pitch, 75% cut, cross flow		
<u>Connections:</u> <u>Shell in:</u> 10 inch - sch. 40 <u>out:</u> 20 inch - sch. 40 <u>Channel in:</u> 12 inch - sch. 80 <u>out:</u> 12 inch - sch. 80		
<u>Corrosion allowance:</u> Shell side, 1/16" Tube side, 0.035"		
<u>Code requirements:</u> ASME 1950 on shell side; 1949 Par. U-68 on tube side except that maximum allowable design stress shall not exceed 10,000 psi		

(SR indicates stress relieving & XR indicates radiographing)

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SCHEMATIC ARRANGEMENT OF 50,000 KW VERTICAL
TUBE EVAPORATOR - NUCLEATE BOILING FROM TUBES

FIG. 1

FIG. 2 Heat Transfer Coefficients for Water Boiling at One Atmosphere From Horizontal and Vertical Tubes.

SCHEMATIC DIAGRAM - FORCED CIRCULATION EVAPORATOR

FIG. 3

Drawing #16772

Scale $\frac{1}{2}'' = 1'-0''$

FIG. 4

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FIG. 6 - Shell Side Heat Transfer Coefficients for Forced Convection of Water at 387.9 °F and 215 psia Over the Outside Surfaces of 1/2 Inch O.D. Vertical Tubes in a 25% Segmentally Baffled Heat Exchanger.

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