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TO: H. E. Goeller and W. E. Unger
FROM: W. L. Carter

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0.0 SUMMARY

There are two condensers in the chemical plant low pressure system which are used to dissipate heat from radioactive decay of fission products. One condenser is associated with each of the decay-storage vessels. Attenuation of beta and gamma radiation in the stored slurry of fuel solution and solids causes boiling and decomposition of D_2O . These vapors are passed through a recombiner for D_2 and O_2 recovery and then into the condenser from which the condensate is refluxed to the decay-storage vessel.

The condenser is a conventional design of shell and tube heat exchanger. Some of the more pertinent data are summarized below.

Shell size	= 4" pipe, Sch. 40
Tube size	= 0.5" dia. x 0.049" wall x 3' lg.
No. tubes	= 19
Area (outside tubes)	= 7.46 ft ²
Condensing temperature	= 210°F
Condensing pressure	= 14 psia
Normal vapor rate	= 10.3 lb/hr
Maximum vapor rate	= 40 lb/hr
Vapor temperature entering condenser	= 293°F
Condensate temperature	= 140°F
Condenser duty (max rate)	= 40,205 BTU/hr
Cooling water required	= 2 gpm
Cooling water temperature (inlet)	= 70°F
Cooling water temperature (outlet)	= 110°F

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1.0 INTRODUCTION

In the low pressure system of the HRT chemical plant beta and gamma attenuation in the contents of the decay-storage vessels causes boiling and radiolytic decomposition of D_2O . A platinized alumina catalyst serves to recombine the decomposed D_2O in the presence of the excess steam, after which the vapor is condensed and refluxed. The condensers in this system therefore have the duty of removing heat from the decay of radioactivity. There are two condensers--one to serve each of the decay storage vessels.

The recombiner is immediately upstream of the condenser, and there is a requirement that the feed stream to the recombiner be superheated. This is to prevent condensation on the catalyst surface. Since the recombiner discharges directly into the condenser, one design criterion is that a portion of the condenser be a gas cooler or desuperheater. In addition it is desirable to subcool the condensate from the condensing temperature to $140^{\circ}F$.

The operating pressure of the unit is expected to be in the neighborhood of 14 psia. Contrary to the reactor low pressure system, there will be only minute volumes of fission product gases discharging from the chemical plant low pressure system. Therefore there will be no significant build up of pressure in the chemical plant to push non-condensables through the carbon beds. In fact it would not be surprising if the chemical plant condensers did not operate at a slight vacuum caused by the degassing effect of the boiling operation. For example, nitrogen, oxygen, xenon, krypton and any other non-condensable gases might be exhausted from the condenser vent by steam through the cold trap and into the carbon bed. All of these gases will be adsorbed to some degree by charcoal and not be available for the breathing action of the system as the vaporization rate fluctuates. It will be recalled that the chemical plant cannot breathe back through the carbon bed from the outside atmosphere because of the mercury seal between the stack and the carbon bed.

The tube side of the condenser will be required to be safe for an explosion of D_2 and O_2 which might occur if insufficient diluent steam is supplied. The general rule for this condition is that pressure surges of 10 to 20 times the operating pressure might result. This however presents no stringent design problems since 10 to 20 atmospheres pressure is a moderate design condition for equipment of this size. The shell side of the unit will be designed to hold cooling water at line pressure.

A schematic diagram of the chemical plant low pressure system is presented in Figure 1. This diagram shows the position of the recombiners with respect to other pertinent pieces of equipment. More details may be found on drawings E-20246 and E-24173 which are HRT Core Processing Engineering Flow Diagram-Sheet 1 and Sheet 2 respectively.

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FIGURE 1

SCHEMATIC DIAGRAM OF
HRT CHEMICAL PLANT LOW
PRESSURE SYSTEM

2.0 DESIGN BASIS

The condenser for the HRT chemical processing plant has been designed for the following conditions:

Comp.	Rate	Gm moles/min	lbs/hr
D ₂ O	2.9 - 17.85 ft ³ /min at 14 psia and 210°F	2.455 - 15.12	6.5 - 40

In addition to just D₂O the vapor stream will from time to time contain very small amounts of fission product gases. The amounts of these, however, are so small that they have been neglected.

Temperature of gas entering condenser	= 146°C (293°F)
Temperature of condensate	= 45.8°C (110°F)
Viscosity of gas at 293°F	= 0.0127 cp
Thermal conductivity of gas at 293°F	= 0.0137 BTU-ft/hr-ft ² °F
Operating pressure	= 14 psia
Condensing temperature	= 210°F
Cooling water temperature in	= 70°F
Cooling water temperature out	= 110°F
Heat of reaction D ₂ + O ₂	= 197 BTU/hr
Design pressure (tube side)	= 500 psig
Design pressure (shell side)	= 75 psig

The maximum vapor rate of 40 lb/hr is fixed by the allowable pressure drop through the low pressure system, principally through the recombiner. The normal vapor rate will probably be 10.3 lb/hr; however, the condenser will be designed to take the maximum rate.

3.0 CONDENSER DUTY

The condenser duty is made up of the heat of condensation, desuperheating and condensate cooling. Condensing occurs at 14 psia and 210°F, whereas, the vapor enters the unit at 293°F. There are 83°F superheat to be removed before condensation occurs.

$$\text{Superheat in vapor} = 40 \text{ lb/hr} \times (1188-1150) \text{ BTU/lb} + 197 \text{ BTU/hr} \\ = 1717 \text{ BTU/hr}$$

The 197 BTU/hr is heat contributed by the exothermic reaction D₂ + O₂.

$$\text{Heat of condensation} = 40 \text{ lb/hr} \times (971.9)(0.919) \text{ BTU/lb} \\ = 35,700 \text{ BTU/hr}$$

Condensate is to be cooled from 210°F to 140°F. The heat to be removed in this operation

$$\begin{aligned} &= 40 \text{ lb/hr} \times (177.6 - 107.9) \text{ BTU/lb} \\ &= 2788 \text{ BTU/hr} \end{aligned}$$

$$\begin{aligned} \text{Total condenser duty} &= 1717 + 35,700 + 2788 \\ &= 40,205 \text{ BTU/hr} \end{aligned}$$

4.0 COOLING WATER REQUIREMENT

If it is assumed that cooling water is available at 70°F and that water leaves the condenser at 110°F, the water requirement

$$\begin{aligned} &= 40,205 / (110 - 70) \\ &= 1005 \text{ lb/hr} \\ &= 2.01 \text{ gpm} \end{aligned}$$

5.0 CONDENSER GEOMETRY

Coefficients of heat transfer depend upon the geometry of the condenser which must be assumed before calculations can be started. It will be assumed that the condenser is a shell and tube unit of all welded construction as shown in Figure 1. The shell will be made of two sections of 4-inch, Sch. 40 pipe between which will be an expansion joint. Six evenly spaced baffles will promote cross flow of cooling water.

Process vapors will be condensed inside 19 tubes which are 0.5 inches diameter and have 0.049-inch walls. The tube length will be 3 feet which makes a total heat transfer surface of 7.46 ft² based on the outside area of the tubes. In addition to the condensing load the condenser will be required to remove 83°F superheat and to subcool the condensate 70°F. The exit of the unit is vented to a cold trap and a charcoal adsorber.

Figure 2 presents the recombiner and condenser assembly. In elevation the recombiner is mounted 20 inches higher than the condenser to facilitate removal of the recombiner under hot operating conditions. Fabrication details of the unit are available on drawing D-20253, Revision 4, HRT Core Processing Recombiner-Condenser, H-1 and H-2.

6.0 HEAT TRANSFER COEFFICIENTS

The condenser can be visualized as being composed of three sections: (1) a desuperheating section, (2) a condensing section, and (3) a cooling section. The overall coefficient of heat transfer can be calculated for each section, and thereby the area for each section can be determined.

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FIGURE 2
RECOMBINER - CONDENSER ASSEMBLY

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6.1 Film Coefficient For Cooling Water

The shell side of the condenser will be baffled to promote cross flow of the cooling water stream. A method* of calculating film coefficients for liquids flowing normal to staggered banks of tubes is given by McAdams.

$$\left(\frac{hD_o}{k_f} \right) \left(\frac{c_p \mu_f}{k_f} \right)^{-1/3} = \phi \left(\frac{D_o G_{max}}{\mu_f} \right)$$

$\phi \left(\frac{D_o G_{max}}{\mu_f} \right)$ is obtained from Fig. 115 of Heat Transmission.

The terms of this equation have the following meanings and values.

- h = coefficient of heat transfer, BTU/hr-ft²-°F
- D_o = outside diameter of tubes = 0.0417 ft
- c_p = specific heat of fluid = 1 BTU/lb-°F
- G_{max} = mass velocity, lb/hr-ft²
- μ_f = viscosity, lb/ft-hr (at film temperature t_f)
- k_f = thermal conductivity, BTU-ft/hr-ft²-°F (at film temperature t_f)

* McAdams, W. H., Heat Transmission, page 230, Fig. 115, 2nd edition, McGraw-Hill Book Co., Inc., New York, 1942

In order to evaluate the viscosity and thermal conductivity, it is first necessary to determine the film temperature. By definition the film temperature is

$$t_f = t_s - 0.5 (t_s - t)_m$$

The surface temperature, t_s , will vary along the length of the condenser tube. If it is assumed that the hot end of the condenser will have a tube surface at 230°F and that the cold end will be at 130°F, the average value of

$$t_s (\text{avg}) = 180^\circ\text{F}$$

Using these two values for temperature and the inlet and outlet water temperatures given in the above diagram, the mean temperature difference can be calculated.

$$\begin{aligned} (t_s - t)_m &= \frac{(230-110) - (130-70)}{\ln \frac{120}{60}} \\ &= 86.6^\circ\text{F} \end{aligned}$$

$$\begin{aligned} \text{Therefore } t_f &= 180 - 0.5 \times 86.6 \\ &= 136.7^\circ\text{F} \end{aligned}$$

At this temperature

$$\begin{aligned} \mu_f &= 1.18 \text{ lb/ft-hr} \\ k_f &= 0.375 \text{ BTU-ft/hr-ft}^2\text{-}^\circ\text{F}. \end{aligned}$$

There are 19 tubes, 0.5" diameter, inside the 4", Sch. 40, pipe shell. Therefore the cross section effective for cooling water flow is

$$\begin{aligned} \text{Area} &= (\text{area of shell}) - (\text{area of tubes}) \\ &= 12.73 - (19)(0.5)^2\pi/4 \\ &= 12.73 - 3.73 \\ &= 9.0 \text{ in}^2 \end{aligned}$$

The mass rate of flow of cooling water can be calculated.

$$\begin{aligned} G_{\text{max}} &= 1005 \text{ lb/hr} \times 144/9 \text{ ft}^2 \\ &= 16,080 \text{ lb/hr-ft}^2 \end{aligned}$$

The Reynold's number for this situation is

$$\begin{aligned} D_o G_{\text{max}}/\mu_f &= (0.0417)(16,080)/1.18 \\ &= 568. \end{aligned}$$

Reading from Fig. 115 of Heat Transmission,

$$\left(\frac{hD_o}{k_f} \right) \left(\frac{c_p \mu_f}{k_f} \right)^{-1/3} = \phi \left(\frac{D_o G_{\text{max}}}{\mu_f} \right) = 18.$$

Substituting known values

$$\left(\frac{0.0417 \text{ h}}{0.375} \right) \left(\frac{1 \times 1.18}{0.375} \right)^{-1/3} = 18$$

$$(0.1112 \text{ h})/1.466 = 18$$

$$h = 237 \text{ BTU/hr-ft}^2\text{-oF}$$

6.2 Fouling Factor

$$h_{\text{foul}} = 1000 \text{ BTU/hr-ft}^2\text{-oF}$$

6.3 Wall Coefficient

$$\text{wall thickness} = x = 0.049 \text{ in.}$$

$$\text{conductivity} = 10 \text{ BTU-ft/hr-ft}^2\text{-oF}$$

$$K_w = k/x = (10)(12)/0.049 = 2350 \text{ BTU/hr-ft}^2\text{-oF}$$

6.4 Film Coefficient Inside Tubes for Desuperheating Section

Film coefficients for the flow of gases inside tubes can be correlated with properties of the system by equations and curves given in Heat Transmission. It is first necessary to determine whether the flow is turbulent or streamline. This is a function of the Reynold's No. which is

$$Re = DG/\mu$$

$$D = 0.402 \text{ in.} = 0.0335 \text{ ft}$$

$$\mu = 0.0127 \text{ c}_p$$

$$\text{Tube cross section} = 0.000878 \text{ ft}^2$$

$$Re = (0.0335)(40)/(0.0127)(2.42)(0.000878)(19) = 2610$$

Flow is in the turbulent region; therefore, the correlation of Eqn. 4c, page 168, of Heat Transmission will be used.

$$\frac{hD}{k} = 0.023 \left(\frac{DG}{\mu} \right)^{0.8} \left(\frac{c_p \mu}{k} \right)^{0.4}$$

$$\frac{0.0335 h}{0.0137} = 0.023 (2610)^{0.8} \left(\frac{0.48 \times 0.0127 \times 2.42}{0.0137} \right)^{0.4}$$

$$h = (0.023)(540)(1.01)(0.0137)/0.0335 = 5.1 \text{ BTU/hr-ft}^2\text{-oF}$$

6.5 Film Coefficient Inside Tubes For Condensing Section

For film type condensation inside horizontal tubes Eqn. 18, page 268, of Heat Transmission will be used.

$$h = 0.95 \left(\frac{k_f^3 \rho_f g L_H}{\mu_f w} \right)^{1/3}$$

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The subscript, f , means that these values are taken at film temperature, t_f , which is defined as

$$t_f = t_{sv} - 0.75\Delta t$$

$$t_{sv} = \text{saturation temperature of vapor} = 210^\circ\text{F}$$

$$t_s = \text{condensing surface temperature} = 200^\circ\text{F (assumed)}$$

$$\Delta t = t_{sv} - t_s = 10^\circ\text{F}$$

$$\begin{aligned}\text{Therefore } t_f &= 210 - 0.75(10) \\ &= 202.5^\circ\text{F}\end{aligned}$$

At this surface temperature

$$\mu_f = 0.34 \text{ c}_p = 0.823 \text{ lb/ft-hr}$$

$$k_f = 0.389 \text{ BTU-ft/hr-ft}^2\text{-}^\circ\text{F}$$

$$\rho_f = 66.8 \text{ lb/ft}^3$$

$$g = \text{acceleration due to gravity} = 4.17 \times 10^8 \text{ ft/hr-hr}$$

$$L_H = \text{length of tube} = 1 \text{ ft (assumed condensing length)}$$

$$\begin{aligned}w &= \text{condensate flow rate per tube} = 40/19 \\ &= 2.1 \text{ lb/hr}\end{aligned}$$

When these values are substituted in the above equation, there results

$$\begin{aligned}h &= 0.95 \left[\frac{(0.389)^3 (66.8)^2 (4.17 \times 10^8) (1)}{(0.823)(2.1)} \right]^{1/3} \\ &= 0.95 (63.2 \times 10^9)^{1/3} \\ &= 3780 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}\end{aligned}$$

It is apparent that this film will offer negligible resistance to heat transfer.

6.6 Film Coefficient Inside Tubes For Subcooling Section

For this section the coefficient of heat transfer is calculated by means of Eqn. 30, page 205, of Heat Transmission.

$$h (k^3 \rho^2 g / \mu_f^2)^{0.333} = 0.67 \left(\frac{c_{\mu_f}^{1.667}}{k_L \rho^{0.667} g^{0.333}} \right)^{0.333} \left(\frac{4T}{\mu_f} \right)^{0.111}$$

$$\mu_f = \text{film viscosity} = 0.47 \text{ c}_p = 1.14 \text{ lb/ft-hr}$$

$$k = 0.377 \text{ BTU-ft/hr-ft}^2\text{-}^\circ\text{F}$$

$$\rho = 67 \text{ lb/ft}^3$$

$$c = 1 \text{ BTU/lb-}^\circ\text{F}$$

$$g = 4.17 \times 10^8 \text{ ft/hr-hr}$$

$$L = 2 \text{ ft (assumed length of subcooler)}$$

$$\begin{aligned}T &= \text{flow rate per unit breadth} = w/2L \\ &= 40/(19)(2)(2) \\ &= 0.526 \text{ lb/hr-ft}\end{aligned}$$

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Viscosity was evaluated at 140°F; other terms were evaluated at an average temperature of 170°F.

Substituting in the above equation,

$$h \left[\frac{(0.377)^3 (67)^2 (4.17 \times 10^8)^{-0.333}}{(1.14)^2} \right]^{0.333}$$

$$= 0.67 \left[\frac{(1.14)^{1.667}}{(0.377)(2)(67)^{0.667} (4.17 \times 10^8)^{0.333}} \right]^{0.333} \left[\frac{(4)(0.526)}{1.14} \right]^{0.111}$$

$$h/4260 = (0.67) \left(\frac{1.075}{21} \right) (1.071)$$

$$h = 156 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$$

7.0 HEAT TRANSFER AREAS

7.1 Desuperheating Section

The surface area required to desuperheat the incoming gas is

$$A = q/U\Delta T_m$$

where the terms have the usual significance. The film and wall resistances may be added to give the overall heat transfer coefficient U. Basing the resistance on the water side film, there results

$$\frac{1}{U} = \frac{1}{h_{H_2O}} + \frac{A_{H_2O}}{h_{wall} A_{wall}} + \frac{A_{H_2O}}{h_g A_g} + \frac{1}{h_{foul}}$$

$$A_{H_2O} = \text{water side surface area per foot} = 0.131 \text{ ft}^2/\text{ft}$$

$$A_{wall} = \text{average wall surface area per foot} = 0.118 \text{ ft}^2/\text{ft}$$

$$A_g = \text{gas side surface area per foot} = 0.105 \text{ ft}^2/\text{ft}$$

Substituting numerical values

$$1/U = 1/237 = 0.131/(2450)(0.118) + 0.131/(5.1)(0.105) + 1/1000$$

$$U = 3.9 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$$

The total quantity of heat transferred in the desuperheating section is 1717 BTU/hr. The temperature rise in the cooling water as a result of this heat absorption will be 1.7°F; whereas, the gas will cool from 293°F to 210°F. Since there is countercurrent flow and the exit water temperature is 110°F, the cooling water entering the desuperheating section must be at 108.3°F.

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The mean temperature difference is

$$\Delta T_m = \frac{(293-110)-(210-108.3)}{\ln \frac{183}{101.7}} = \frac{81.3}{\ln 1.799} = 138.5^\circ\text{F}$$

Heat transfer is calculated by the familiar equation

$$A = q/U\Delta T_m = 1717/(3.9)(138.5) = 3.2 \text{ ft}^2$$

7.2 Condensing Section

The overall heat transfer coefficient of the condensing section is found in the same manner as above.

$$\begin{aligned} \frac{1}{U} &= \frac{1}{h_{H_2O}} + \frac{A_{H_2O}}{h_{wall}A_{wall}} + \frac{A_{H_2O}}{h_c A_c} + \frac{1}{h_{foul}} \\ &= 1/237 + 0.131/(2450)(0.118) + 0.131/(3780 \times 0.105) + 1/1000 \\ U &= 167 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F} \end{aligned}$$

The quantity of heat absorbed in the condensing section is 35,700 BTU/hr, and since this is absorbed by 1005 lb/hr of cooling water, the temperature rise of the water will be 35.5°F. At the exit of the condensing section--entrance to desuperheating section--the cooling water is at 108.3°F. Therefore it must be at 72.8°F when it enters the condensing section. For a condensing temperature of 210°F,

$$\Delta T_m = \frac{(210-72.8)-(210-108.3)}{\ln \frac{137.2}{101.7}} = \frac{35.5}{\ln 1.349} = 118.5^\circ\text{F}$$

The area required for the condensing section is

$$A = q/U\Delta T_m = 35,700/(167)(118.5) = 1.8 \text{ ft}^2$$

7.3 Subcooling Section

Using the same procedure of the two preceding calculations,

$$\begin{aligned} \frac{1}{U} &= \frac{1}{h_{H_2O}} + \frac{A_{H_2O}}{h_{wall}A_{wall}} + \frac{A_{H_2O}}{h_{sc}A_{sc}} + \frac{1}{h_{foul}} \\ &= 1/237 + 0.131/(2450)(0.118) + 0.131/(156)(0.105) + 1/1000 \\ U &= 73.1 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F} \end{aligned}$$

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Through this section the temperature rise of the cooling water is 2.8°F; the condensate cools from 210°F to 140°F.

$$\Delta T_m = \frac{(210-72.8)-(140-70)}{\ln \frac{137.2}{70}} = \frac{67.2}{\ln 1.96}$$

$$= 100^\circ\text{F}$$

The heat to be transferred in subcooling the condensate is 2788 BTU/hr.

$$A = q/U\Delta T_m = 2788/(73.1)(100)$$

$$= 0.38 \text{ ft}^2$$

7.4 Total Area Of Condenser

By adding the required areas for the three sections of the unit, the total calculated area can be found.

$$\text{Total area} = \text{desuperheating area} + \text{condensing area} + \text{subcooling area}$$

$$= 3.2 + 1.8 + 0.38$$

$$= 5.38 \text{ ft}^2$$

The condenser selected for these calculations is that described in paragraph 5.0 and contains 7.46 ft² of tube outside surface. This means that the condenser is about 39% oversized. Since these calculations were made for the condenser operating under maximum load, there is even more excess capacity for normal operation. Maximum operating load is 40 lb/hr; whereas, the normal load is 10.3 lb/hr.

8.0 THERMAL EXPANSION OF TUBES

It might be assumed that the most stringent condition for stressing the juncture of the tube sheet and shell occurs when the maximum temperature difference exists between tubes and shell. If steam were admitted to the condenser with cooling water shut off, the tubes could be at 293°F and the shell at 100°F. The linear coefficient of expansion of stainless steel is 9.6 x 10⁻⁶ in./in.-°F; therefore

$$\Delta L = 9.6 \times 10^{-6} \text{ in./in.}^\circ\text{F} \times 36 \text{ in.} \times (293-100)^\circ\text{F}$$

$$= 0.0667 \text{ in.}$$

This amount of expansion is probably insufficient to overstress the weld between tube sheet and shell; however, the shell is designed with an expansion joint to avoid all complications that might result from such expansion.

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