DESIGN OF A
LITHIUM BROMIDE / WATER
ABSORPTION REFRIGERATION SYSTEM.

BY

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ABSTRACT

An absorption refrigeration system with a capacity of 50 000 Btu/hr (14.6 kJ/s) is designed. The use of lithium bromide as the absorbent necessitates having a heat source for maintaining the generator at a minimum temperature of 180°F (82.2°C). The lower temperature limit of the system is set by the evaporator requirements. A design value of 40°F (4.4°C) is used because water is the refrigerant. A mass and energy balance is made for a suitable range of generator temperatures. The thermal design as well as the sizing of heat exchangers is made to accomodate this temperature range.

ACKNOWLEDGEMENTS

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NOMENCLATURE

A Area, ft^2, (m^2)

a Area, in^2, (cm^2)

a_t Flow area of a single tube in^2 (cm^2)

c Specific Heat, Btu/lbm ^oF, (kJ/kg ^oC)

d Diameter, in, (cm)

F Temperature difference factor, t= F x LMTD, dimensionless

f Friction factor, ft^2/in^2, (cm^2/cm^2)

G Mass velocity, lbm/hr ft^2, (kg/s m^2)

G" Condensate loading for horizontal tubes, lbm/hr ft, (kg/s m)

h Film coefficient, Btu/ft^2hr ^oF, (kJ/m^2 s ^oC)

h Enthalpy, Btu/lbm, (kJ/kg)

h_{fi}' h_f' corrected to the inside surface and corrected for dirt factor, Btu/ft^2 hr ^oF, (kJ/m^2 s ^oC)

h_f' Heat transfer coefficient on fin side corrected for dirt factor, Btu/ft^2 hr ^oF, (kJ/s ^oC)

h_{io} Inside film coefficient referred to outside surface, $h_i \times A_i / A_o$, Btu/ft^2 hr ^oF, (kJ/m^2 s ^oC)

j_h Heat transfer factor, dimensionless

k Thermal conductivity, Btu/ft hr ^oF, (kJ/m s ^oC)

L Distance, ft, (m)

l Distance, in, (cm)

M Mass flow, lbm/hr, (kg/s)

m Mass flow ratio, lbm/lbm H_2O, (kg/kg H_2O)

N Number of tubes, dimensionless

P Pressure, lbf/in^2, (N/m^2)

p Pressure, mmHg, (N/m^2)

Q Heat flux, Btu/hr, (kJ/s)

R Fouling factor ft^2 hr ^oF/Btu, $(m^2 s\ ^oC / kJ)$

R Temperature ratio $(T_1 - T_2)/(t_2 - t_1)$, dimensionless

Re Reynolds number, dimensionless

r Radius, in, (cm)

S Temperature ratio $(t_2 - t_1)/(T_1 - t_2)$, dimensionless

T Temperature, (temperature of hot fluid in heat transfer calculation) ^oF, (^oC)

t Temperature of cold fluid in heat transfer calculations ^oF, (^oC)

U Coefficient of heat transfer, Btu/ft^2 hr ^oF, $(kJ/m^2\ s\ ^oC)$

V Velocity, ft/min, (m/s)

v Velocity, Ft/sec, (m/s)

v Specific volume, ft^3/lbm, $(m^3\ kg)$

X Lithium bromide weight fraction %

μ Dynamic viscosity, lbm/ft hr, (Ns/m^2)

Ω Fin efficiency, h_b / h_f, dimensionless

ρ Density, lbm/ft^3, (kg/m^3)

β Expansion coefficient, $1/^oF$, $(1/^oC)$

Subscripts

a absorber

b base

c condenser

e evaporator, equivalent, edge of fin

f fin, film

g generator

h liquid heat exchanger

i inside

o outside

s shell

t tube

w wall

1 in

2 out

LIST OF FIGURES AND TABLES

1 INTRODUCTION

Operation of the absorption refrigeration system is based on the principle that a chemical substance can attract large quantities of the vapor of another at relatively low temperatures. In the proposed system water is the refrigerant and lithium bromide is the absorbent or carrier.

The absorption system operates in a manner similar to the mechanical vapor compression cycle in its use of a condenser, throttling valve and evaporator with the exception that the mechanical vapor compression process is replaced by an absorber and generator.

In fig.1 the refrigerant at 8 is throttled to wet vapor and a lower pressure at 9 . Vaporization takes place in the evaporator to produce the refrigeration effect. The cool vapor at 10 is absorbed into the lithium bromide carrier in the absorber where the latent heat of the refrigerant vapor and heat of chemical reaction is removed by cooling water. High concentration lithium bromide enters the absorber at 3 to replenish the supply. The mixture of the carrier and refrigerant is

then pumped into the generator at a higher pressure at 6 . In the generator a high temperature source is necessary to separate the refrigerant from the carrier. Pure refrigerant vapor at 7 flows to the condenser where it becomes liquid at 8 while giving off its latent heat. The remaining carrier in the generator flows back into the absorber at 3 . A heat exchanger between the carrier and carrier-refrigerant serves to economize energy.

The refrigeration capacity of 50 000 Btu/hr (14.6 kJ/s) makes the system suitable for use as a large residential or small commercial air conditioning unit. The generator heat requirements may be satisfied by a large flat plate solar collector working under optimum conditions. The 40°F (4.4°C) evaporator temperature represents a practical lower limit imposed by the use of water as a refrigerant. The 90°F (32.2°C) absorber outlet temperature and the 110°F (43.3°C) condenser temperature represent practical design values for a system of this type. The latter determines the condenser and generator pressure. A 10°F (5.6°C) approach is used at the low end of the liquid heat exchanger.

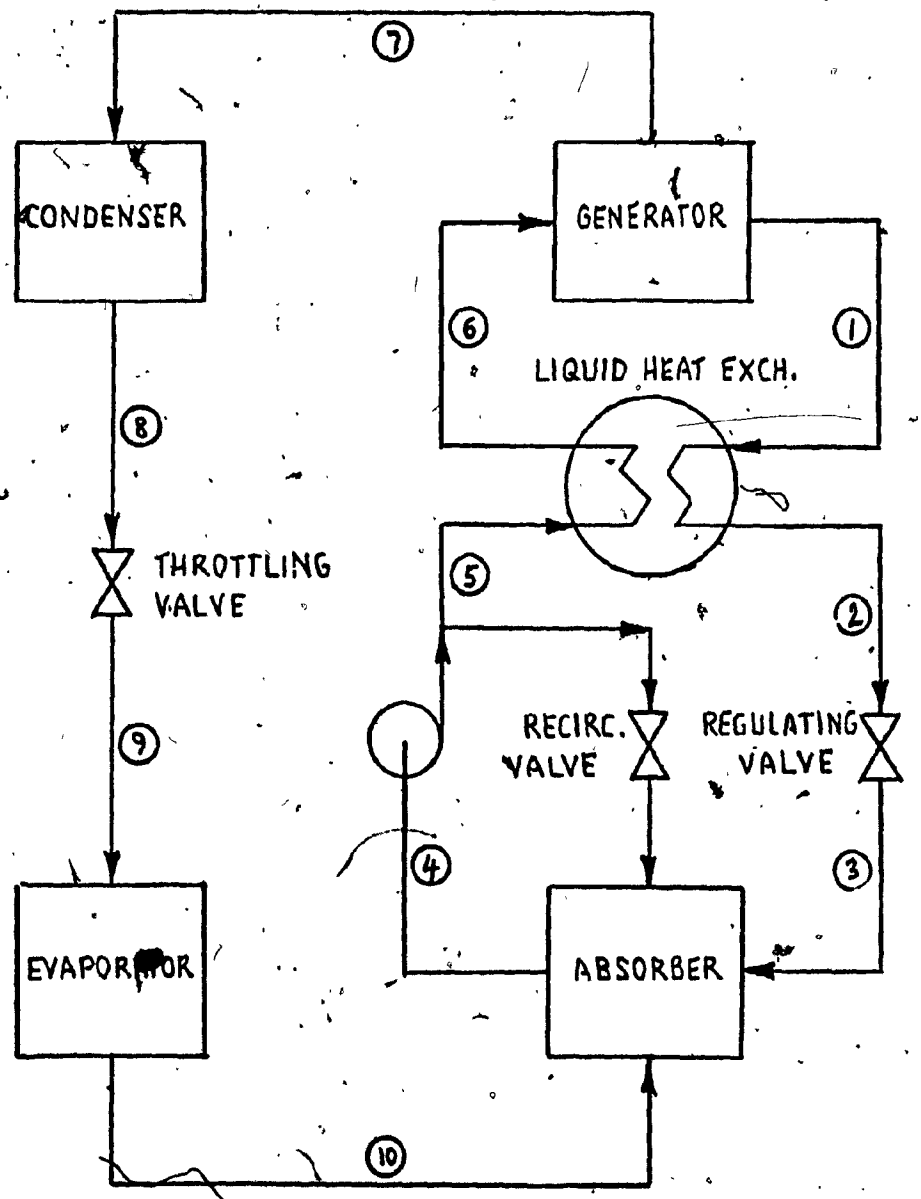

Fig. 1 Lithium Bromide water absorption system, Block diagram.

2 MASS AND ENERGY BALANCE

2.1 GENERAL REQUIREMENTS

Refrigeration capacity: 50 000 Btu/hr (14.6 kJ/s)

Evaporator temperature: 40°F (4.44°C)

Absorber outlet temperature: 90°F (32.2°C)

Condenser temperature: 110°F (43.3°C)

Generator temperature: 185°F (85°C)

2.2 ASSUMPTIONS

1. Refrigerant and absorbent phases are in equilibrium at the above points.
2. Pressure at the evaporator and condenser are equal and determined by the vapor pressure of water.
3. Pumping work is negligible.
4. Pressure losses across piping and heat exchangers are considered to be negligible for the mass and energy balance.
5. Temperature change across the regulating valve is negligible.
6. Heat transfer between equipment and surroundings is negligible.

2.3 THERMODYNAMIC PROPERTIES

With reference to fig. 1 , flow of refrigerant is assumed to be unity.

$m_7 = m_8 = m_9 = m_{10} = 1$ lb H_2O

Weight fraction of absorbent:

$X_7 = X_8 = X_9 = X_{10} = 0\%$

Evaporator conditions: (refrigerant)

$T_9 = T_{10} = 40^oF$ (4.44^oC)

vapor pressure $p_{10} = 6.290$ mmHg, (836 N/m^2)

(ASME steam tables p. 88)

$p_3 = p_4 = p_9 = p_{10}$

Enthalpy: $h_{10} = 1079$ Btu/lbm (2510 kJ/kg)

(ASME steam tables p. 88)

Absorber conditions: (Low concentration absorbent at 6.290 mmHg)

$T_4 = T_5 = 90^oF$ (32.2^oC)

Weight fraction $X_4 = 55.2\%$ (fig. 2)

$X_4 = X_5 = X_6$

Enthalpy: $h_4 = -75.5$ Btu/lb (-175.6 kJ/kg)

(fig. 2)

$h_4 = h_5$ (assumption 3)

Condenser conditions:

$T_8 = 110^oF$ (43.3^oC)

Vapor pressure: $p_8 = 65.94$ mmHg (8770 N/m^2)

(ASME steam tables p. 88)

$p_1 = p_2 = p_5 = p_6 = p_7 = p_8$ (Assumption 4)

Enthalpy : $h_8 = 77.98$ Btu/lbm (181kJ/kg)

(ASME steam tables p. 88)

$h_9 = h_8$

Generator conditions:

$T_1 = T_7 = 185^oF$ (85^oC)

Enthalpy of vapor: h_7 = 1140 Btu/lbm (2646 kJ/kg)

(ASME steam tables p. 86)

Weight fraction:(at 65.94 mmHg): X_1 = 58.7%

(fig. 2)

$X_1 = X_2 = X_3$

Enthalpy of mixture : h_1 = -31.5 Btu/lb (-73.2 kJ/kg)

(fig. 2)

Enthalpy at X_3 : h_3 = -70 Btu/lbm (-162.6 kJ/kg)

(fig. 2)

$h_2 = h_3$

Heat exchanger :

$T_2 = T_5 + 10 = 100^oF$ (37.8^oC)

$T_3 = T_2$ (assumption 5)

2.4 FLOW AND ENERGY REQUIREMENTS

Generator material balance

$$m_1 + m_7 = m_6$$

$$m_1 X_1 = m_6 X_6$$

$$m_1 X_1 = (m_1 + m_7) X_6$$

$$m_1 = m_7 \frac{(X_6)}{(X_1 - X_6)}$$

$$m_1 = \frac{1 \times 55.2}{58.7 - 55.2} = 15.77 \text{ lbm/lbm } H_2O \text{ (kg/kg } H_2O)$$

$$m_2 = m_3 = m_1$$

$$m_6 = m_1 + m_7 = 1 + 15.77 = 16.77 \text{ lbm/lbm } H_2O \text{ (kg/kg } H_2O)$$

$$m_4 = m_5 = m_6$$

Heat exchanger energy balance :

$$m_5 h_5 + m_1 h_1 = m_2 h_2 + m_6 h_6$$

$$h_6 = \frac{m_5 h_5}{m_6} + \frac{m_1 h_1}{m_6} + \frac{m_2 h_2}{m_6}$$

$m_1 = m_2$ and $m_5 = m_6$

$$h_6 = h_5 + \frac{m_1}{m_6}(h_1 - h_2)$$

$$= -75.5 + \frac{15.77}{16.77}(-3.15 + 70)$$

$$= -39.30 \text{ Btu/lbm } (-91.25 \text{ kJ/kg})$$

Temperature at h_6 and X_6 : $T_6 = 164^\circ F$ ($73.3^\circ C$) (fig. 2)

Flowrate of refrigerant (water):

$$m_{10} = \frac{q}{h_9 - h_{10}} = \frac{50\ 000}{1079 - 77.98} = 44.95 \text{ lbm/hr}$$

(20.39 kg/hr)

Flowrate of low concentration absorbent :

$$M_3 = 15.77 \times 49.95 = 787.7 \text{ lb/hr } (357.3 \text{ kg/hr})$$

Flowrate of high concentration absorbent :

$$M_4 = 16.77 \times 49.95 = 837.7 \text{ lb/hr } (380.0 \text{ kg/hr})$$

Generator energy requirements :

$$Q_g = M_7 h_7 + M_1 h_1 - M_6 h_6$$

$$= (49.95 \times 1140) + (787.7 \times -31.5) - (837.7 \times -39.3)$$

65 100 Btu/hr (19.03 kJ/s)

Condenser energy requirements:

$$Q_c = M_7 (h_8 - h_7)$$
$$= 49.95 (77.98 - 1140)$$
$$= - 53\ 000 \text{ Btu/hr } (-15.5 \text{ kJ/s})$$

Absorber energy requirements can be determined by assuming that the net energy input to the system must be zero.

$$Q_a = Q_c - Q_g - Q_e \qquad \text{(Assumption 6)}$$
$$= 53\ 000 - 65\ 100 - 50\ 000$$
$$= -62\ 100 \text{ Btu/hr } (-18.2 \text{ kJ/s})$$

Liquid heat exchanger energy transfer:

$$Q_h = M_1 (h_1 - h_2)$$
$$= 787.7 (-31.5 + 70)$$
$$= 30\ 300 \text{ Btu/hr } (8.86 \text{ kJ/s})$$

Coefficient of performance:

$$COP = \frac{Q_e}{Q_g} = \frac{50\ 000}{65\ 100} = 0.768$$

2.5 GENERATOR TEMPERATURE VARIATION

Results of mass and energy balance for five generator liquid temperatures varying from 175°F to 205°F (79.4°C to 96.1°C) are shown in tables 1 and 2 . The temperature of the

generator heat source is most easily subject to fluctuations especially if a solar collector is used as a source of heating water. The generator liquid temperature determines the mass fraction of the liquid leaving the generator and consists of high concentration absorbent ① ② ③ while the absorber outlet temperature, being kept constant, determines the weight fraction of the low concentration absorbent ④ ⑤ ⑥ . The lower the generator temperature the smaller the difference between the two weight fractions and therefore the higher the flowrate in the absorbent streams. With the data presented, an absolute limit would be 172°F at which an infinitely large amount of absorbent would have to be circulated. A practical design limit would be apprximately 180°F (82.2°C).

Varying the generator temperatures with all other variables being kept constant shows that heat transferred in the various components does not vary significantly except for the liquid heat exchanger where the heat transfer requirements increases greatly as the generator temperature approaches its critical low point.

Table 1 Thermodynamic properties of refrigerant and absorbent

COND.	CASE	TEMPERATURE T °F (°C)	PRESSURE P mmHg (N/m²)	LiBr X % LiBr	FLOW m lbm/lbm (kg/kg)	FLOW M lbm/hr (kg/s)	ENTHALPY h Btu/lbm (kJ/kg)
1	A	175 (79.4)	65.94 (119000)	56.0	69.0	3447 (0.435)	-34.5 (-19.2)
1	B	180 (82.2)	65.94 (119000)	57.3	26.3	1314 (0.166)	-33.5 (-18.6)
1	C	185 (85.0)	65.94 (119000)	58.7	15.77	738 (0.0994)	-31.5 (-17.5)
1	D	195 (90.6)	65.94 (119000)	61.5	8.76	437 (0.0551)	-28.5 (-15.8)
1	E	205 (96.1)	65.94 (119000)	63.8	6.42	321 (0.0405)	-24.5 (-13.6)
2	A	100 (37.8)	65.94 (119000)	56.0	69.0	3447 (0.435)	-74.5 (-13.6)
2	B	100 (37.8)	65.94 (119000)	57.3	26.3	1314 (0.166)	-72.5 (-40.3)
2	C	100 (37.8)	65.94 (119000)	58.7	15.77	788 (0.0994)	-70.0 (-38.9)
2	D	100 (37.8)	65.94 (119000)	61.5	8.76	437 (0.0551)	-64.5 (-35.8)
2	E	100 (37.8)	65.94 (119000)	63.8	6.42	321 (0.0405)	-59.5 (-33.1)
3	A	100 (37.8)	6.290 (11400)	56.0	69.0	3447 (0.435)	-74.5 (-41.4)
3	B	100 (37.8)	6.290 (11400)	57.3	26.3	1314 (0.166)	-72.5 (-40.3)
3	C	100 (37.8)	6.290 (11400)	58.7	15.77	788 (0.0994)	-70.0 (-38.9)
3	D	100 (37.8)	6.290 (11400)	61.5	8.76	437 (0.0551)	-64.5 (-35.8)
3	E	100 (37.8)	6.290 (11400)	63.8	6.42	321 (0.0405)	-59.5 (-33.1)
4	A	90 (32.2)	6.290 (11400)	55.2	70.0	3497 (0.441)	-75.5 (-41.9)
4	B	90 (32.2)	6.290 (11400)	55.2	27.3	1364 (0.170)	-75.5 (-41.9)
4	C	90 (32.2)	6.290 (11400)	55.2	16.77	838 (0.106)	-75.5 (-41.9)
4	D	90 (32.2)	6.290 (11400)	55.2	9.76	488 (0.0615)	-75.5 (-41.9)
4	E	90 (32.2)	6.290 (11400)	55.2	7.42	371 (0.0468)	-75.5 (-41.9)
5	A	90 (32.2)	65.94 (119000)	55.2	70.0	3497 (0.441)	-75.5 (-41.9)
5	B	90 (32.2)	65.94 (119000)	55.2	27.3	1364 (0.170)	-75.5 (-41.9)
5	C	90 (32.2)	65.94 (119000)	55.2	16.77	837 (0.106)	-75.5 (-41.9)
5	D	90 (32.2)	65.94 (119000)	55.2	9.76	488 (0.0615)	-75.5 (-41.9)
5	E	90 (32.2)	65.94 (119000)	55.2	7.42	371 (0.0468)	-75.5 (-41.9)

COND.	CASE	TEMPERATURE T °F (°C)	PRESSURE P mmHg (N/m²)	LiBr X % LiBr	FLOW m lbm/lbm kg/kg	FLOW M lbm/hr (kg/s)	ENTHALPY h Btu/lbm (kJ/kg)
6	A	169 (76.1)	65.94 (119000)	55.2	70.0	3497 (0.441)	-36.3 (-20.2)
6	B	166 (74.4)	65.94 (119000)	55.2	27.3	1364 (0.170)	-37.9 (-21.1)
6	C	164 (73.3)	65.94 (119000)	55.2	16.77	837 (0.106)	-39.3 (-21.8)
6	D	156 (68.9)	65.94 (119000)	55.2	9.76	488 (0.0615)	-43.2 (-24.0)
6	E	151 (66.1)	65.94 (119000)	55.2	7.42	371 (0.0468)	-45.2 (-25.1)
7	A	175 (79.4)	65.94 (119000)	0	1	49.95 (0.00630)	1136 (631)
7	B	180 (82.2)	65.94 (119000)	0	1	49.95 (0.00630)	1138 (632)
7	C	185 (85.0)	65.94 (119000)	0	1	49.95 (0.00630)	1140 (633)
7	D	195 (90.6)	65.94 (119000)	0	1	49.95 (0.00630)	1144 (635)
7	E	205 (96.1)	65.94 (119000)	0	1	49.95 (0.00630)	1148 (638)
8	A	110 (43.3)	65.94 (119000)	0	1	49.95 (0.00630)	77.98 (43.3)
8	B	110 (43.3)	65.94 (119000)	0	1	49.95 (0.00630)	77.98 (43.3)
8	C	110 (43.3)	65.94 (119000)	0	1	49.95 (0.00630)	77.98 (43.3)
8	D	110 (43.3)	65.94 (119000)	0	1	49.95 (0.00630)	77.98 (43.3)
8	E	110 (43.3)	65.94 (119000)	0	1	49.95 (0.00630)	77.98 (43.3)
9	A	40 (4.44)	6.290 (1400)	0	1	49.95 (0.00630)	77.98 (43.3)
9	B	40 (4.44)	6.290 (1400)	0	1	49.95 (0.00630)	77.98 (43.3)
9	C	40 (4.44)	6.290 (1400)	0	1	49.95 (0.00630)	77.98 (43.3)
9	D	40 (4.44)	6.290 (1400)	0	1	49.95 (0.00630)	77.98 (43.3)
9	E	40 (4.44)	6.290 (1400)	0	1	49.95 (0.00630)	77.98 (43.3)
10	A	40 (4.44)	6.290 (1400)	0	1	49.95 (0.00630)	1079 (599)
10	B	40 (4.44)	6.290 (1400)	0	1	49.95 (0.00630)	1079 (599)
10	C	40 (4.44)	6.290 (1400)	0	1	49.95 (0.00630)	1079 (599)
10	D	40 (4.44)	6.290 (1400)	0	1	49.95 (0.00630)	1079 (599)
10	E	40 (4.44)	6.290 (1400)	0	1	49.95 (0.00630)	1079 (599)

Table 2 **Heat requirements**

CASE	ABSORBER	GENERATOR	CONDENSER	EVAPORATOR	LIQUID HX
	Q_a	Q_g	Q_c	Q_e	Q_e
	Btu/hr (kJ/s)	Btu/hr (kJ/s)	Btu/hr (kJ/s)	Btu/hr (kJ/s)	Btu/hr (kJ/s)
A	-62000 (-18.1)	64800 (18.9)	-52800 (15.4)	50000 (14.6)	138000 (40.3)
B	-61600 (-18.0)	65400 (19.1)	-52900 (15.5)	50000 (14.6)	51200 (15.0)
C	-62100 (-18.2)	65100 (19.0)	-53000 (15.5)	50000 (14.6)	30300 (8.86)
D	-62600 (-18.3)	65800 (19.2)	-53200 (15.6)	50000 (14.6)	15700 (4.59)
E	-67200 (-19.6)	70600 (20.6)	-53400 (15.6)	50000 (14.6)	11200 (3.27)

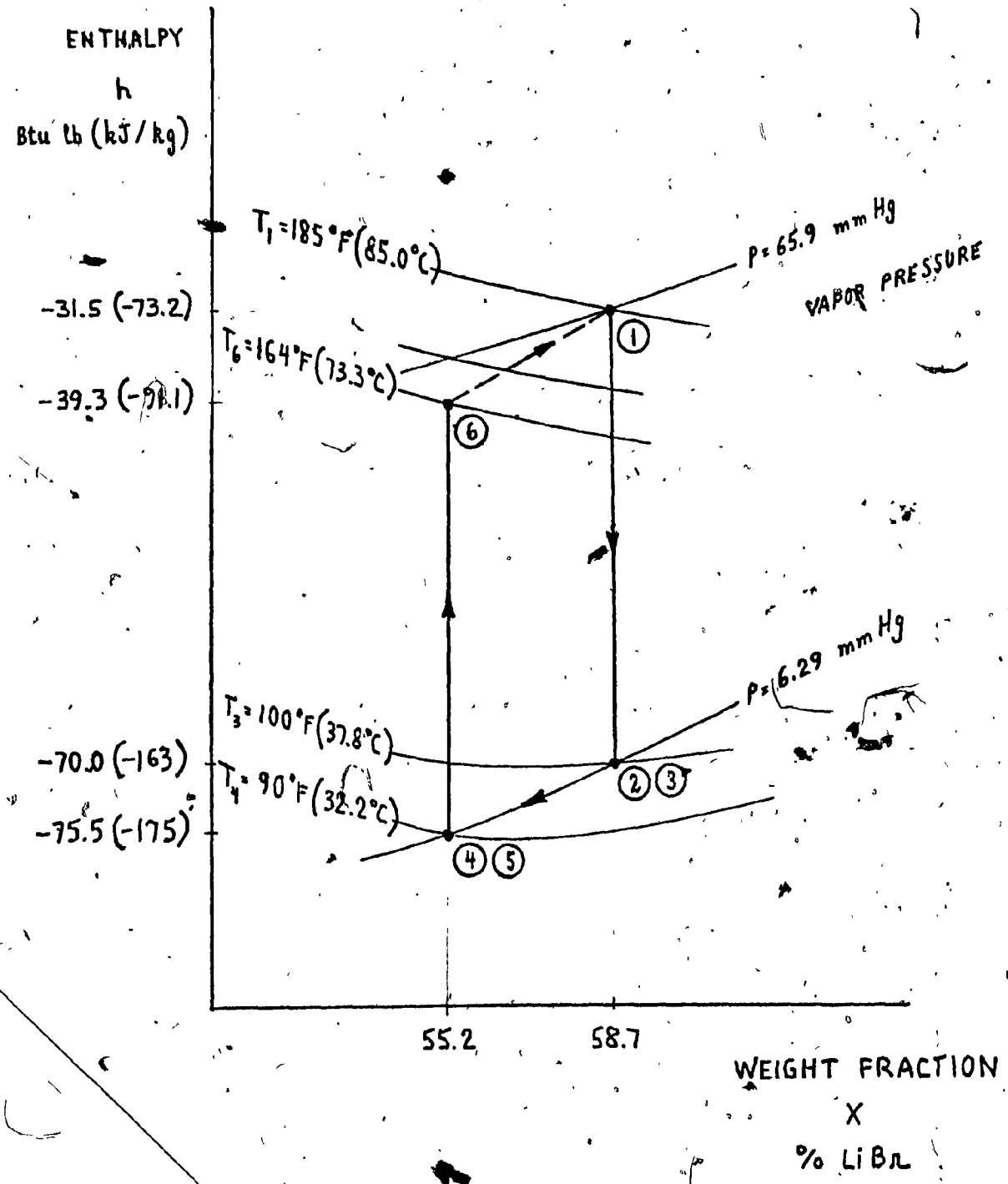

Fig.2 Enthalpy concentration diagram for case C

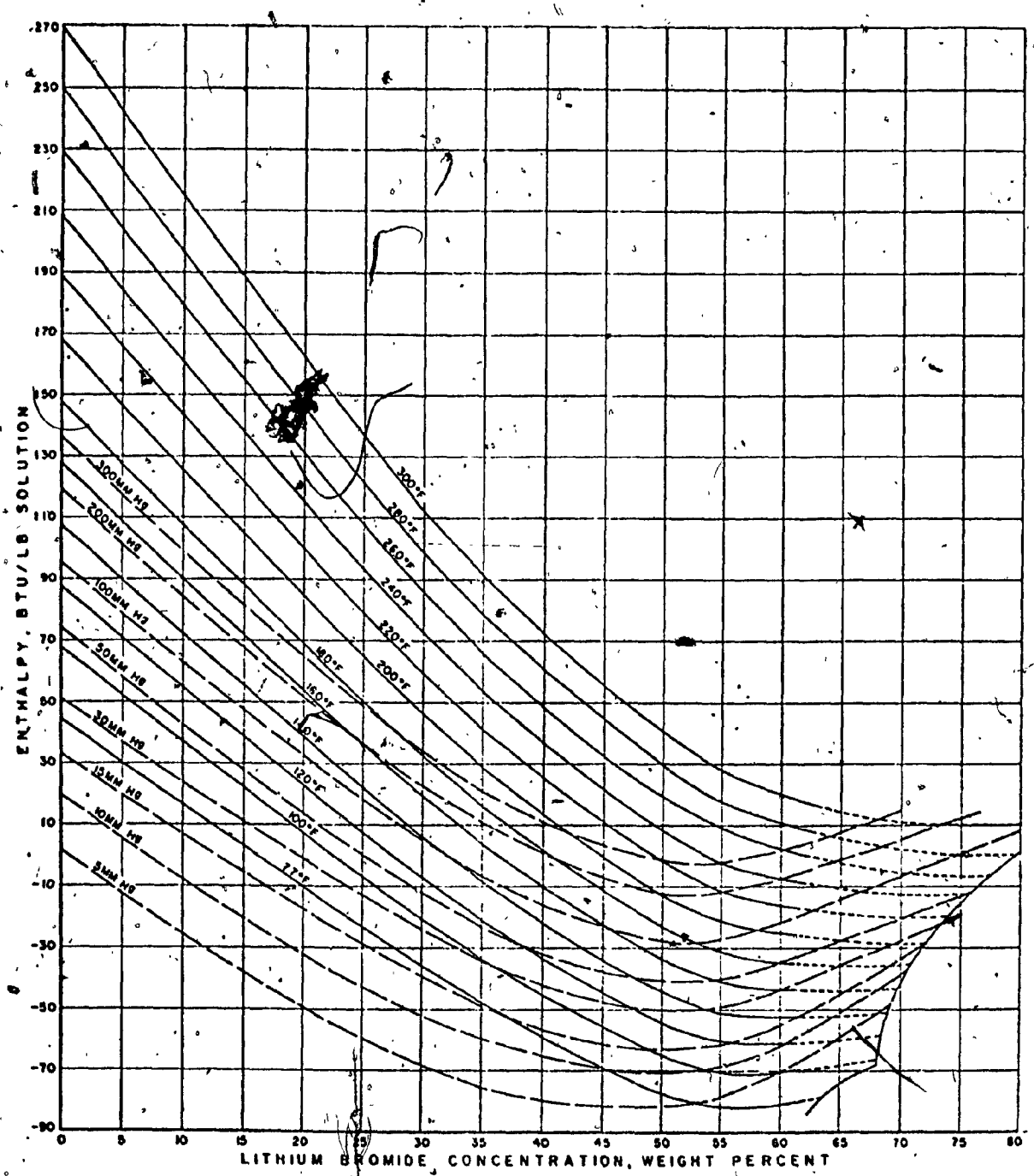

Fig.3 Enthalpy concentration diagram for LiBr-Water

3 MECHANICAL AND THERMAL DESIGN

3.1 GENERAL

Design will accomodate temperature and heat flow requirements for case B to case E as given in the mass and energy balance. Flow velocities are kept as low as possible for the purpose of reducing pumping requirements. Because both the evaporator and absorber have the same working pressure it is practical to combine their functions in a single vessel. In a similar manner both the condenser and generator are combined in one shell as shown in fig. 5.

Heat transfer between vessels and the ambient air is assumed negligible. The fouling factor used throughout will be 0.0010 ft^2 hr ^oF/Btu (0.176 m^2 s ^oC/kJ).

Properties of lithium bromide are taken from reference [6] when possible and estimated from the properties of sodium chloride which is similar chemically to LiBr. The properties required are; thermal conductivity, viscosity, specific heat and density.

3.1.1 Tubing

Design will be based on sched. 40 of a nominal pipe size of 1/8" with an I.D. of 0.269 in (0.6833 cm) and an O.D. of 0.405 in (1.029cm).

Tube pitch is square with a spacting of 1.5 x the outside diameter which gives a tube pitch of 0.6075 in. (1.543cm)

Equivalent diameter:

$$d_e = \frac{4 \times P_T^2 - \frac{d_o^2}{4}}{d_o} = \frac{4 \times 0.6075^2 - \times \frac{0.405^2}{4}}{\times 0.405}$$

$$= 0.755 \text{ in } (1.918 \text{ cm})$$

3.1.2 Finned tubing

As is the general design practice is to use finned tubing in the application of finned tubing to air or gas applications. Where necessary design will be based on 1/8 in sched. 40 IPS tubing with 1 in (2.54cm) diameter fins having a thickness of 0.035 in (0.0889cm). Spacing is at 8 fins per inch as shown below.

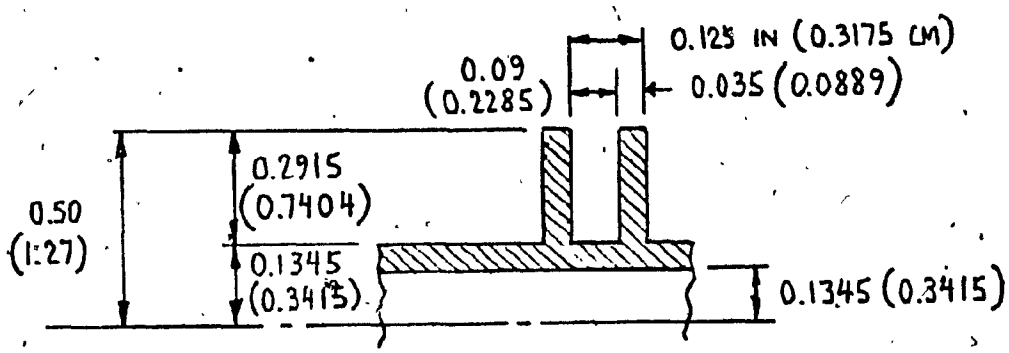

Fig. 4 Dimensions of finned tubing, in (cm)

Fin area: $\frac{\pi}{4}$ $(1^2 - 0.405^2) \times 2 \times 8 \times 12 = 126.1\ in^2/ft$

$= 2669\ cm^2/m$

Bare tube area : $\pi \times 0.405 \times 0.09 \times 8 \times 12 = 10.99\ in^2/ft$

$= 232.6\ cm^2/m$

Total projected perimeter:

$(0.2975 \times 2 \times 2 + 0.09 \times 2) \times 8 \times 12 = 131.5\ in/ft$

$= 1096\ cm/m$

Equivalent diameter:

$$d_e = \frac{2\ (\text{Fin area} + \text{Bare tube area})}{(\text{Projected perimeter}}$$

(Kern p. 554)

$$= \frac{2\ (126.1 + 10.99)}{131.5}$$

$= 0.6637\ in = (1.686\ cm)$

$= 0.0553\ ft = (0.0169\ m)$

3.1.3 Heat transfer

Forced convection film coefficients are obtained from a graphical representation of the sieder and tate correlations which give

$$\frac{h_i D}{k} = 1.86 \left[\left(\frac{DG}{\mu}\right)\left(\frac{c\,\mu}{k}\right)\left(\frac{D}{L}\right)\right]^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

for streamline flow.

For turbulent flow:

$$\frac{h_i D}{k} = 0.027 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c\,\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

These are combined on a single pair of coordinates with the ordinate

$$j_h = \left(\frac{h_i D}{k}\right) \left(\frac{c\,\mu}{k}\right)^{-1/3} \left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

plotted against values for $\left(\frac{DG}{\mu}\right)$

The film coefficient for free convection outside horizontal tubes is determined from the equation

$$h_c = 116\left[\left(\frac{k_f \rho_f c_f \beta}{\mu_f}\right)\left(\frac{\Delta t}{d}\right)\right]^{0.25}$$

(An alignment chart for this equation is available)

Although this equation is intented for a single tube it is assumed that the clearing between tubes of a half tube diameter still gives satisfactory results.

Natural circulation film coefficients are obtained from a curve showing the relationship between h and $(\Delta t)_w$ from Kern [4, p. 474]. The film coefficient for boiling is considered to be independent of the velocity and dependent upon the temperature difference between the tube wall and the saturation temperature of the boiling fluid.

The condensation film coefficient for horizontal tubes is calculated using a correlation derived in Kern [4, p. 266]

$$h\left(\frac{\mu_f^2}{k_f^3 \rho_f^3 g}\right)^{1/3} = 1.5\left(\frac{4G''}{\mu_f}\right)$$

The value for condensate loading $G = \frac{W}{L(N)^{1/3}}$ is empirical and applicable, for horizontal tubes in bundles where the occurence of splashing of condensate over successive rows of tubes requires a different value for loading than for single tubes.

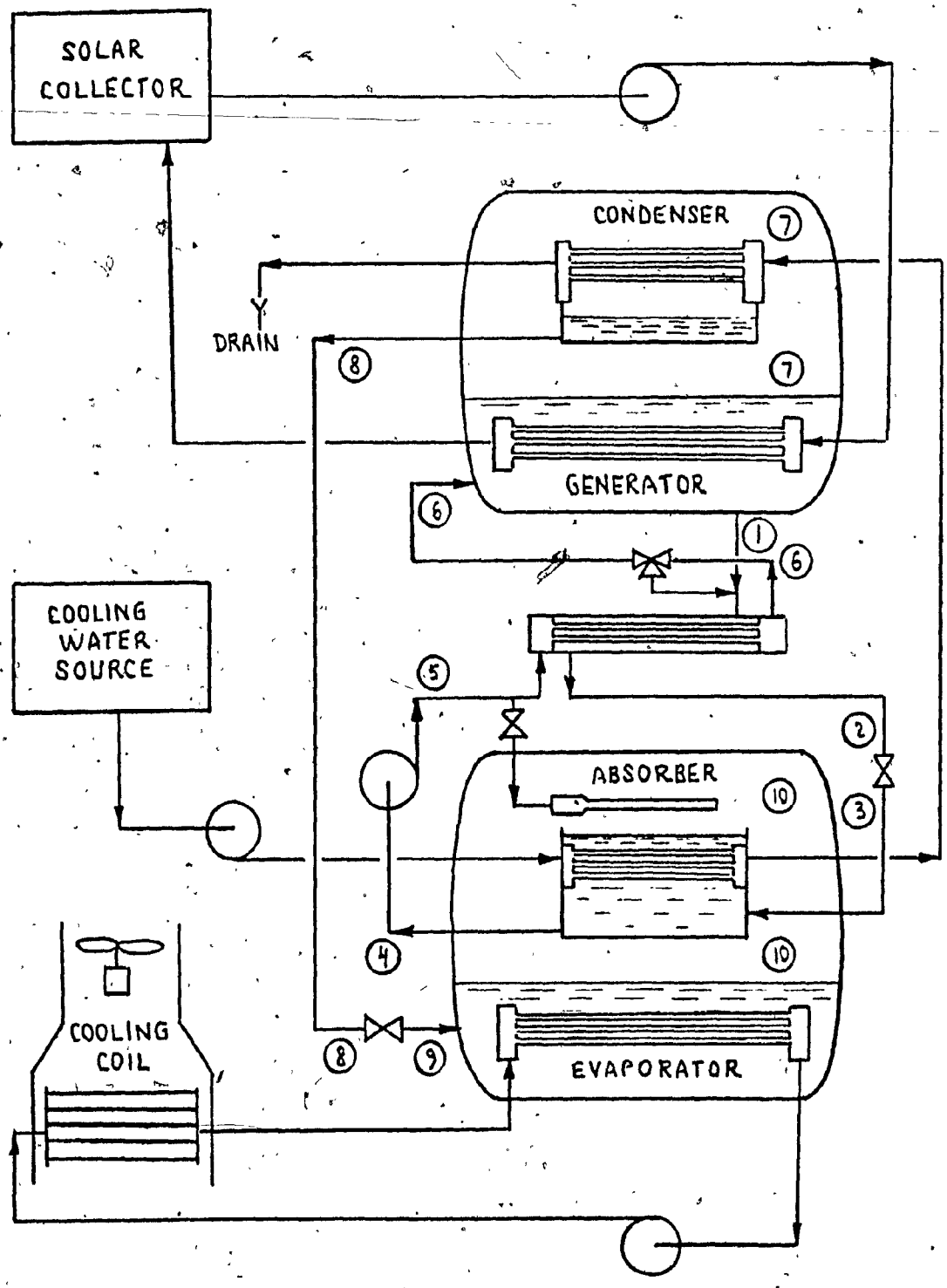

Fig. 5 Lithium bromide / water absorption system equipment and piping diagram

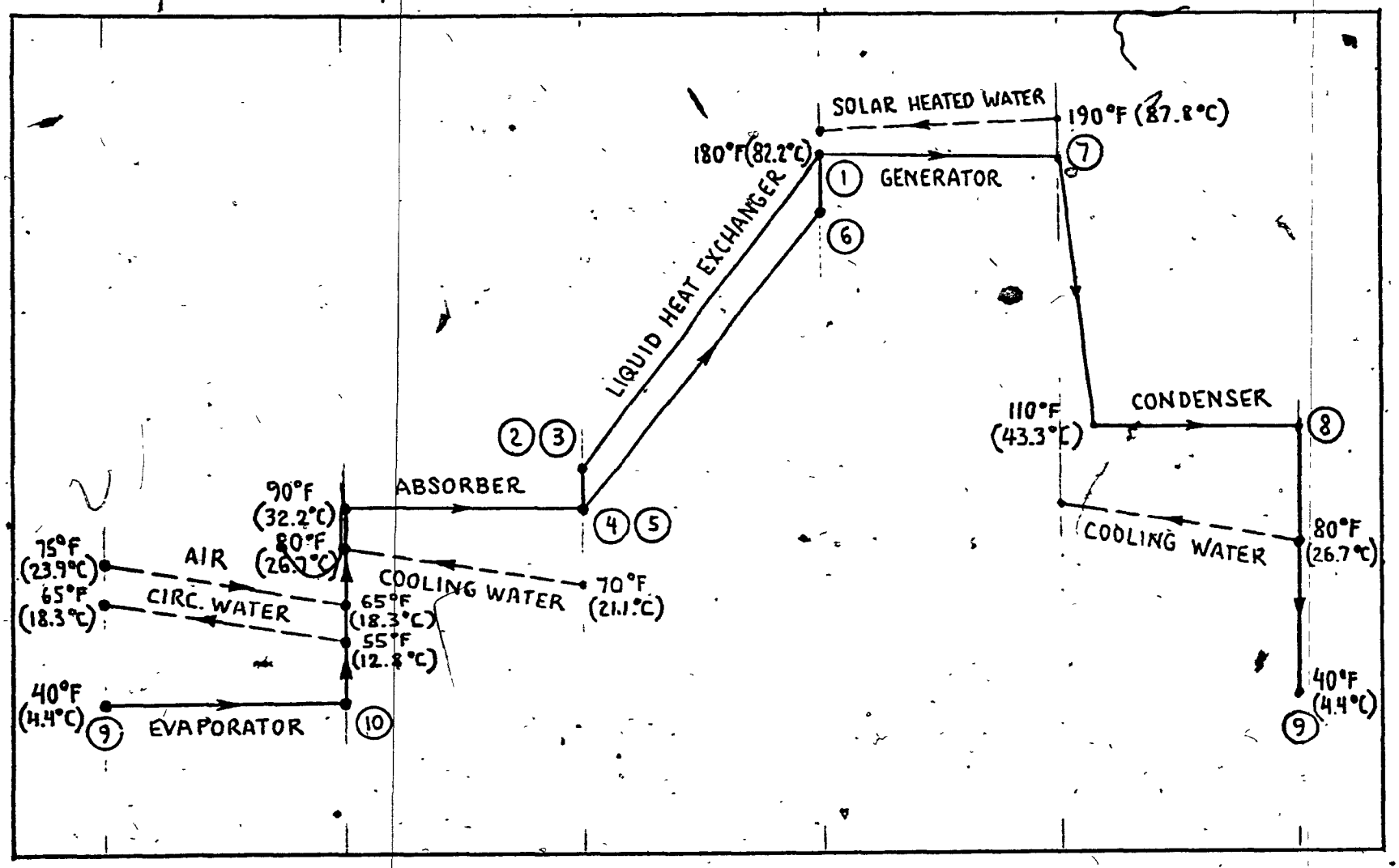

Fig. 6 Temperature chart for case B

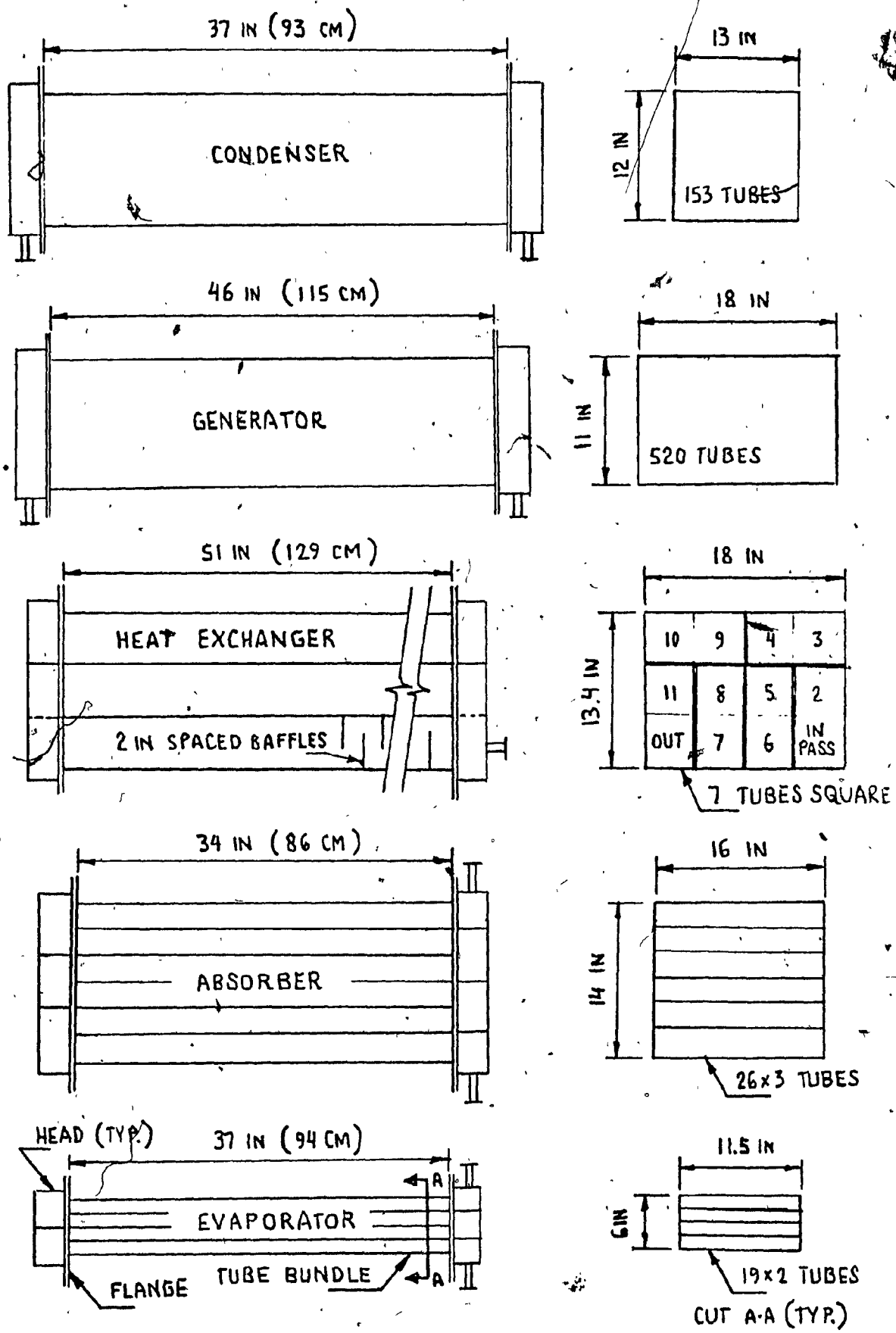

Fig. 7 Bundle and head configuration of heat exchangers

3.2 EVAPORATOR COOLING COIL

The cooling coil transfers heat from the ambient air to the evaporator to produce the air refrigeration effect. Direct expansion of the refrigerant in an air coil is avoided to allow the evaporator and absorber to be in the same vessel. In this design, water is circulated between the forced convection air cooled coil and the tube side of the evaporator, The coil will be of the finned type and will utilize a blower or fan. Coil sizing will be based on an air temperature drop from 75°F to 65°F (23.9°C to 18.3°C).

3.2.1 Choice of cooling water temperatures

Heat transfer surface area varies inversely with the log mean temperature. The total average temperature difference available between the 40°F (4.4°C) evaporator temperature and the 70°F (21.1°C) average air temperature is 30°F (16.7°C). Assuming that a larger LMTD should be made available for heat transfer in the evaporator and that heat transfer surface on the cooling coil is half the cost of heat transfer surface inside the pressure vessel shell of the evaporator, the average circulating temperature would be:

$$t = 40 + \frac{2}{3}(30) = 60^{\circ}F\ (15.9^{\circ}C) \text{ with a } \Delta t \text{ of } 10^{\circ}F\ (5.5^{\circ}C)$$

for the cooling water.

Hot fluid		Cold fluid	
T_1 = 75°F (23.9°C)	High temp.	t_2 = 65°F (18.3°C)	10°F (5.5°C)
T_2 = 65°F (18.3°C)	Lower temp.	t_1 = 55°F (12.8°C)	10°F (5.5°C)
10°F (5.5°C)	Difference	10°F (5.5°C)	

LMTD = 10°F (5.5°C)

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{10}{10} = 1$$

$$S = \frac{t_2 - t_1}{T_1 - t_1} = \frac{10}{20} = 0.5$$

assuming that both fluids are unmixed because of fin arrangement

F = 0.91 (Kern, Fig.6.17a)

$$t = 10 \times 0.91 = 9.1^\circ F \ (5.05^\circ C)$$

3.2.2 Air requirements

$$M = \frac{Q}{c \quad t} = \frac{50\ 000}{0.25 \times 9.1} = 22\ 000 \text{ lbm/hr } (2.77 \text{ kg/s})$$

$$CFM = m \times c_a$$

$$= \frac{22\ 000 \times 13.3}{60} = 4880 \text{ ft}^3/\text{min } (2.3 \text{ m}^3/\text{s})$$

where specific volume of air is 13.3 ft^3/lbm (Jennings p. 88)

assuming a coil duct velocity of 500 ft/min (2.54 m/s) (Jennings p. 428)

Total face area: $A = \frac{CFM}{V} = \frac{4880}{500} = 9.76\ ft^2$

$(0.9067\ m^2)$

using a 3 ft x 3 ft (0.914 m x 0.914 m) duct for the coil would require a velocity of 540 ft/min (164.6 m/min) which is acceptable.

3.2.3 Shell side

area with 36 tubes spaced at 1 inch in 36 in x 36 in duct

$a_s = 36 \times 36 - 0.405 \times 36 \times 36 - 0.2975 \times 2 \times 0.035 \times 8 \times 36 \times 36 = 555.2\ in^2 = 3.86\ ft^2$ $(0.358\ m^2)$

$D_e = 0.0553$ ft (0.01686 m) (previous calculation)

Mass velocity:

$$G_s = \frac{M}{a_s} = \frac{22\ 000}{3.86} = 5700\ lbm/hr\ ft^2 \quad (7.74\ kg/s\ m^2)$$

Reynolds number:

$$Re_s = \frac{D_e\ G_s}{\mu} = \frac{0.0553 \times 5700}{0.0431} = 7310$$

where $\mu = 1.82 \times 10^{-6}\ Ns/m^2$ (0.0431 lb/ft hr) for air at 70°F (21.1°C) (Kern p. 825)

Heat transfer factor:

$j_h \approx 57$ (Kern p 555)

Prandtl number:

$$\frac{(C\ \mu)^{1/3}}{k} = \frac{(0.249 \times 0.0431)^{1/3}}{0.0149} = 0.896$$

where k = 0.0149 Btu/hr ft °F (2.57×10^{-5} kJ/m s °C) (Kern p. 801)

where c = 0.249 Btu/lb °F (1.04 kJ/kg °C) (Kern p. 805)

Coefficient of heat transfer:

$$h_f = j\left(\frac{k}{D_e}\right)\left(\frac{c}{k}\right)^{1/3}$$

$$= \frac{57 \times 0.0419 \times 0.896}{0.0553}$$

$$= 13.8 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F } (0.00782 \text{ kJ/m}^2\text{s } ^\circ\text{C})$$

Corrected heat transfer coefficient:

$$h_f' = \frac{1}{\frac{1}{13.8} + 0.0010}$$

$$= 13.6 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F } (0.0770 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

Corrected for fins (steel)

k = 26 Btu/ft hr °F (0.0449 kJ/m s °C) (Kern p. 542)

Calculation of fin efficiency:

$$(r_e - r_b)\sqrt{\frac{h}{r_b}} = \frac{(0.5 - 0.2025)}{12}\sqrt{\frac{13.6}{26 \times 0.00146}}$$

$$= 0.469$$

$$\frac{r_e}{r_b} = \frac{0.5}{0.2025} = 2.47$$

Fin efficiency:

$\Omega = 0.89$ (Kern p. 542)

$$h_{fi}' = (\Omega A_f + A_o)\frac{h_f'}{A_i}$$ (Kern p. 520)

$$= \left(\frac{0.89 \times 126.1}{144} + \frac{10.99}{144} \right) \frac{13.6}{0.0704}$$

$= 165$ Btu/ft^2 hr $^\circ$F (0.935 kJ/m^2 s$^\circ$C)

where $A_i = \frac{0.269 \times \pi}{12} = 0.0704$ ft^2/ft

3.2.4 Tube side

Water circulation requirement :

$$M = \frac{Q}{c \ \Delta t} = \frac{50\ 000}{1.00 \times 10} = 5\ 000 \text{ lbm/hr}$$

(0.631 kg/sec)

where c = 1.00 Btu/lb $^\circ$F (4.18 kJ/kg $^\circ$C)

(ASME steam tables p. 278)

Water can be carried by any number of banks. Utilization of all banks would result in an excessively low velocity with a correspondingly lower film coefficient. The pressure drop encountered with a single bank is acceptable.

With one bank of tubes:

$$a_t = \frac{\pi \times 0.269^2 \times 36}{4 \times 144} = 0.01421 \text{ ft}^2 \ (0.001320 \text{ m}^2)$$

velocity:

$$V = \frac{Mv}{A} = \frac{5000 \times 0{,}0160}{3600 \times 0.01421}$$

$= 1.6$ ft/sec (0.487 m/sec)

Mass velocity

$$G_t = \frac{w}{a_t} = \frac{5000}{0.01421} = 352\ 000 \text{ lb/hr ft}^2 \ (478 \text{ kg/s.m}^2)$$

Reynolds number

$$Re_t = \frac{DG_t}{\mu} = \frac{0.0224 \times 352\ 000}{2.84} = 2780$$

where μ = 2.84 lbm/ft hr (0.000120 Ns/m^2)

(ASME steam tables p. 280)

Heat transfer factor :

$j_h = 6.2$ (Kern p. 834)

Prandtl number at 60°F (15.6°C) :

$$\frac{(c\ \mu)^{1/3}}{k} = (8.33)^{1/3} = 2.02$$

(ASME steam tables p. 282)

Coefficient of heat transfer :

$$h_i = j_h\left(\frac{k}{D_e}\right)\left(\frac{c\ \mu}{k}\right)^{1/3}$$

$$= \frac{6.2 \times 0.344 \times 2.02}{0.0224}$$

= 192 Btu/ft^2 hr °F (1.09 kJ/m^2 s °C)

where k = 0.344 Btu/ft hr°F (0.000577 kJ/m s °C)

(ASME steam tables p. 281)

Corrected coefficient:

$$h_i' = \frac{1}{\frac{1}{h_i} + R} = \frac{1}{\frac{1}{192} + 0.0010} = 161 \text{Btu/ft}^2 \text{ hr } ^\circ\text{F}$$

(0.912 kJ/m^{2}s $^\circ$C)

3.2.5 Sizing

$$U = \frac{1}{\frac{1}{h_i'} + \frac{1}{h_{fi}}} = \frac{1}{\frac{1}{161} + \frac{1}{165}}$$

$= 81.5$ Btu/ft^2 hr $^\circ$F (0.484 kJ/m^2 s $^\circ$C)

inside surface required :

$$A = \frac{Q}{U \quad t} = \frac{50\,000}{81.5 \times 9.1} = 67.4 \text{ ft}^2 \ (6.26 \text{ m}^2)$$

inside surface per bank :

$$A = \frac{0.269 \times \pi \times 36 \times 36}{144} = 7.61 \text{ ft}^2 \ (0.707 \text{ m}^2)$$

number of banks :

$$N = \frac{67.4}{7.61} = 8.8 \text{ banks}$$

assume ten banks to allow the inlet and outlet to be on the same side.

3.2.6 Pressure drops

f = 0.000405 at Re 2780 (Kern p. 836)

$$P_t = \frac{fG_t^2 \, Ln}{5.22 \times 10^{10} \, D} \qquad \text{(Kern p. 148)}$$

$$= \frac{0.000405 \times 352\,000 \times 3 \times 10}{5.22 \times 10^{10} \times 0.0224}$$

$$= 1.29 \text{ lbf/in}^2 \quad (8880 \text{ N/m}^2)$$

where $D = \frac{0.269}{12} = 0.0224$ ft

3.3 EVAPORATOR

3.3.1 GENERAL

The evaporator is supplied with 5,000 lbm/hr (0.631 kg/s) of water from the cooling coil at 65°F (18.3°C). The water leaves the evaporator at 55°F (12.8°C).

Hot fluid		Cold fluid	
T_1 = 65°F (18.3°C)	Higher temp.	t_1 = 40°F (4.4°C)	25°F (13.9°C)
T_2 = 55°F (12.8°C)	Lower temp.	t_2 = 40°F (4.4°C)	15°F (8.3°C)
10°F (5.6°C)	Difference	0	10°F (5.6°C)

LMTD = 19.6°F (10.9°C)

3.3.2 SHELL SIDE

Assuming natural circulation boiling

T_i = 60°F (15.6°C) average

T_o = 40°F (4.4°C)

assuming h_o = 131 Btu/ft^2 hr °F (0.742 kJ/m^2 s °C)

h_{io} = 104.5 Btu/ft^2 hr °F (0.592 kJ/m^2 s °C)

$$\frac{Q}{A} = U\Delta t = \frac{19.6}{\frac{1}{131} + \frac{1}{104.5}} = 1139$$

Temperature difference between tube wall and outside temperature :

$$\Delta t = \frac{Q}{Ah} = \frac{1139}{131} = 8.7°F\ (4.9°C)$$

Assuming latent heat transfer only :

h_o = 130 Btu/ft^2 hr ^oF (0.735 kJ/m^2 s ^oC) (Kern p. 474)

which is close to what was assumed .

3.3.3 TUBE SIDE

using a velocity of 1.5 ft/sec

$$A = \frac{Mv}{V} = \frac{5000 \times 0.0160}{1.5 \times 3 \times 3600} = 0.0148 \text{ ft}^2 \ (0.00137 \text{ m}^2)$$

Number of tubes required :

$$N = \frac{0.0148}{0.000395} = 38$$

where tube area $A = \frac{\pi}{4} \left(\frac{0.269}{12}\right)^2 = 0.000395 \text{ ft}^2$

area taken up by one tube :

$A = (0.405 \times 1.5)^2 = 0.369 \text{ in}^2$ (2.38cm^2)

total outside area of bundle :

$0.369 \times 38 = 14 \text{ in}^2$ (90.3 cm^2)

Mass velocity :

$$G_t = \frac{W}{a_t} = \frac{5\ 000}{0.0148} = 338\ 000 \text{ lbm/hr ft}^2 \ (459 \text{ kg/s m}^2)$$

Reynolds number :

$$R_e = \frac{DG_t}{\mu} = \frac{0.0224 \times 338\ 000}{2.99} = 2530$$

where μ = 2.99 lbm/ft hr (0.000126 Ns/m^2)

(ASME steam tables p. 280)

$j_h = 5$ (Kern p. 834)

Prandtl number at 55°F (12.8°C)

$$\frac{(c \quad \mu)^{1/3}}{k} = (8.80)^{1/3} = 2.06 \text{ (ASME steam tables p. 282)}$$

Heat transfer coefficient :

$$h_i = \dot{J}_h \frac{k}{D} \left(\frac{c \quad \mu}{k}\right)^{1/3}$$

$$= \frac{5 \times 0.342 \times 2.06}{0.0224}$$

$$= 157.3 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F } (0.891 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

where $k = 0.342$ at 55°F (12.8°C)

(ASME steam tables p. 281

$$h_{io} = 157.3 \left(\frac{0.269}{0.405}\right) = 104.5 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F } (0.592 \text{ kJ/m}^2\text{s } ^\circ\text{C})$$

3.3.4 <u>Sizing</u>

$$U = \frac{1}{\frac{1}{h_o} + 2R + \frac{1}{h_{io}}}$$

$$U = \frac{1}{\frac{1}{130} + 0.0002 + \frac{1}{104.5}} = 51.9 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F}$$

$$(0.293 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

Heat transfer area :

$$A = \frac{Q}{U \quad \Delta t} = \frac{50\ 000}{51.9 \times 19.6} = 49.1 \text{ ft}^2 \ (4.56 \text{ m}^2)$$

Tube area available :

$$A = \frac{0.405 \times \pi \times 12}{144} = 0.106 \ ft^2/ft$$

Total lenght of tube required :

$$\frac{49.1}{0.106} = 463 \ ft \ (141 \ m)$$

Lenght of bundle :

$$L = \frac{463}{38} = 12.2 \ ft \ (3.72 \ m)$$

at four passes :

L = 3.04 ft = 37 in (94 cm)

3.3.5 Pressure drops

The pressure drop on the shell side is negligible because pool boiling takes place.

Pressure drop on the tube side :

f= 0.00042 ft^2/in^2 (0.0605 cm^2/cm^2) (Kern p. 836)

at Re=2530

$$P_t = \frac{f \ G_t^2 \ Ln}{5.22 \times 10^{10} \ De}$$

$$= \frac{0.00042 \times 338\ 000^2 \times 2.77 \times 4}{5.22 \times 10^{10} \times 0.0224}$$

$$= 0.457 \ lbf/in^2 \ (3150 \ N/m^2)$$

3.4 ABSORBER

3.4.1 GENERAL

In the absorber, steam at 40°F (4.4°C) is absorbed into the circulating lithium bromide solution. Design is based on case E because this offers the highest heat requirement while the design LMTD is kept constant in the absorber for all cases. Heat is carried away by cooling water which is assumed to be available at a temperature below 70°F (21.1°C). This cooling water is pumped through the absorber and condenser in series. The quantity to be pumped gives a cooling water temperature rise of 10°F (5.6°C) in the absorber allowing the condenser to be designed for incoming cooling water at 80°F (26.7°C). Absorption will take place while the lithium bromide is being sprayed on the absorber. In this way a large liquid area can be made available for the absorption process during spraying. Design of the absorber is made on the basis of the tubes being submerged in lithium bromide solution. Operational adjustments should permit operation with the tubes exposed to possibly allow a higher rate of heat transfer in operation.

Cooling water requirements :

$$w = \frac{Q}{c \ \Delta t} = \frac{67\ 200}{0.999 \times 10} = 6730 \text{ lbm/hr } (0.849 \text{ kg/s})$$

Hot fluid		Cold fluid	
T_1 = 90°F (32.2°C)	Higher temp.	t_2 = 80°F (26.7°C)	10°F (5.6°C)
T_2 = 90°F (32.2°C)	Lower temp.	t_1 = 70°F (21.1°C)	20°F (11.1°C)
0	Difference	10°F (5.6°C)	10°F (5.6°C)

LMTD = 14.4°F (8.0°C)

3.4.2 TUBE SIDE

Assuming a velocity of 1 ft/sec (0.305 m/sec) for the cooling water, the total flow area required is :

$$A_t = \frac{W}{V} = \frac{6730}{1 \times 62.3 \times 3600} = 0.0300 ft^2 (\ 0.00279\ m^2)$$

Number of tubes required:

$$N = \frac{0.0300 \times 144}{0.0568} = 76$$

$$a_t' = \frac{\pi}{4}(.269)^2 = 0.0568\ in^2\ (\ .366\ cm^2)$$

$$a_t = \frac{N_t a_t'}{144} = \frac{76 \times 0.0568}{144} = 0.0300\ ft^2\ (\ 0.0028\ m^2)$$

$$G_t = \frac{w}{a_t}$$

$$= \frac{6730}{0.0300} = 224\ 000\ lbm\ /\ hr\ ft^2\ (\ 303\ kg/s\ m^2)$$

$$Re_t = \frac{DG_t}{\mu} = \frac{0.0224 \times 224\ 000}{2.39} = 2100$$

where μ = 2.39 lb/ft hr (0.000101 Ns/m^2) at 75°F (23.9°C)

(ASME steam tables p. 280)

Heat transfer coefficient :

assuming a ratio of lenght to diameter

$$\frac{L}{D} = 200$$

$j_h = 4.0$ (Kern p. 874)

Prandtl number at 75°F (23.9°C)

$$\left(\frac{c\ \mu}{k}\right)^{1/3} = (6.9)^{1/3} = 1.90$$ (ASME steam tables p. 282

Coefficient of heat transfer :

$$h_i = j_h \left(\frac{k}{D_e}\right)\left(\frac{c\ \mu}{k}\right)^{1/3}$$

$$= \frac{4.0 \times 0.351 \times 1.90}{0.0224} = 119 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F } (0.674 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

where k = 0.351 Btu/ft hr °F (0.000606 kJ/m s °C) at 75°F (23.9°C) (ASME steam tables p. 281)

$$h_{io} = 119 \times \frac{\text{I.D.}}{\text{O.D.}} = 119 \times \frac{0.269}{0.405} = 79.0 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F}$$

$$= (0.448 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

3.4.3 <u>SHELL SIDE</u>

assume h_o = 80 Btu/ft² hr °F (0.452 kJ / m² s °C) for estimation of wall temperature

$$\frac{Q}{A} = U\Delta t = \frac{15.0}{\frac{1}{80} + \frac{1}{75.1}} = 581$$

$$\Delta t = \frac{Q}{Ah} = \frac{581}{80} = 7.3\ ^{o}F\ (4.1\ ^{o}C)$$

$t_w = 90 - 7.3 = 82.7\ ^{o}F\ (28.2\ ^{o}C)$

Film temperature

$$t_f = \frac{T_a + t_w}{2} = \frac{90 + 82.7}{2} = 86.4\ ^{o}F\ (30.2\ ^{o}C)$$

The following properties are required for a LiBr-water mixture at $X = 55.2\%$ and $T = 86.4^{o}F\ (30.2^{o}C)$

Coefficient of thermal expansion:

assuming that it can be taken as being the same as for water at $86.4^{o}F\ (30.2^{o}C)$.

between $86^{o}F$ and $87^{o}F$:

$\rho_1 = 62.150\ lb/ft^3$ $\qquad$ $\rho_2 = 62.162\ lb/ft^3$

$$\beta = \frac{\rho_1^2 - \rho_2^2}{2(t_2 - t_1)\rho_1\rho_2} = 0.000186\ 1/^{o}F\ (\ 0.000335\ 1/^{o}C)$$

thermal conductivity:

using values for calcium chloride brine or sodium chloride brine

$k = 0.31$ Btu/ft hr ^{o}F (0.46 kJ /m hr ^{o}C)

(Kern p. 800)

Density:

$\rho = 62.2 \times 1.61 = 100.1\ lb/ft^3\ (1600\ kg/m^3)$ (Ref 6 p. 10)

Specific heat:

using values for sodium chloride brine

$c = 0.8$ Btu/lb ^{o}F (3.34 kJ/kg ^{o}C) (Kern p. 804)

Viscosity:

using values ~~for sodium~~ chloride brine (Kern p. 804)

μ = 1.9cp (0.000194 Ns/m^2)

For free convection

$$\frac{k^3 \rho^2 c \beta}{\mu_f} = \frac{0.31 \times 100.1 \times 0.80 \times 0.000186}{1.9}$$

$$= 0.023$$

$$\frac{\Delta t}{d_o} = \frac{90 - 82.7}{0.405} = 19.0$$

h_o = 87 Btu/ft^2 hr ^{o}F (0.493 kJ/m^2 s ^{o}C) (Kern p. 216)

Adding 25% factor of safety because of the uncertainty involved in the estimation of lithium bromide properties

$$h_o = \frac{87}{1.25} = 69.6 \text{ Btu/ft}^2 \text{ hr } ^o\text{F } (0.394 \text{ kJ/m}^2 \text{ s } ^o\text{C})$$

3.4.4 SIZING

Overall heat transfer coefficient

$$U = \frac{1}{\frac{1}{h_{io}} + 2R + \frac{1}{h_o}}$$

$$U = \frac{1}{\frac{1}{79.0} + 0.002 + \frac{1}{69.6}} = 34.5 \text{ Btu/ft}^2 \text{ hr } ^o\text{F}$$

(0.196 kJ/m^2 s ^{o}C)

$$\text{Area} = \frac{Q}{U \ \Delta t} = \frac{67\ 200}{34.5 \times 14.4} = 135 \text{ ft}^2 \ (12.5 \text{ m}^2)$$

Total length of tube :

$$\frac{135}{0.106} = 1270 \text{ ft } (388 \text{ m})$$

Length of straight bundle :

$$L = \frac{1270}{76} = 17 \text{ ft } (5.2\text{m})$$

at 6 passes :

L = 2.79 ft (0.86m) = 34 in (86cm)

3.4.5 PRESSURE DROPS

The pressure drop on the shell side is negligible.

Pressure drop on the tube side :

$f = 0.00044 \text{ ft}^2/\text{in}^2 \; (0.063 \text{ cm}^2/\text{cm}^2)$

at Re=2100 (Kern p. 836)

$$P_t = \frac{fG_t^2 \, Ln}{5.22 \times 10^{10} \, D}$$

$$= \frac{0.00044 \times 224\,000^2 \times 2.86 \times 6}{5.22 \times 10^{10} \times 0.0224}$$

$$= 0.32 \text{ psi} \quad (2210 \text{ N/m}^2)$$

3.5 LIQUID HEAT EXCHANGER

3.5.1 GENERAL

The liquid heat exchanger economizes energy by using the hot stream of high concentration absorbent to heat up the incoming low concentration absorbent.

In the estimation of LiBr properties both the high temperature stream at an average temperature of 140°F (60°C) with X = 57.3% and the low temperature stream at 128°F (53.3°C) with X = 55.2% are assumed to be similar enough to be given the same properties.

Specific gravity = 1.62 (Ref. 6 p. 10)

The following are estimated from sodium chloride brine

μ = 1.9 cp (0.000194 Ns/m^2) (Kern p. 823)

k = 0.31 Btu/ft hr °F (0.000534 kJ/m s °C)

(Kern p. 800)

c = 0.8 Btu/lb °F (3.34 kJ/kg °C) (Kern p. 804)

Hot fluid		Cold fluid	
T_1 = 180°F (82.2°C)	Higher temp.	t_2 = 166°F (74.4°C)	14°F (7.8°C)
T_2 = 100°F (37.8°C)	Lower temp.	t_1 = 90°F (32.2°C)	10°F (5.6°C)
80°F (44.4°C)	Difference	76°F (42.2°C)	4°F (2.2°C)

LMTD = 11.9 °F (6.6°C)

heat transferred 51 200 Btu/hr (15.0 kJ/s) (table 2)

3.5.2 TUBE SIDE

assuming 1364 lb/hr (0.172 kg/s) in the tube side with a velocity of 0.2 ft/sec (0.061 m/sec)

cross sectional area :

$$A_t = \frac{W}{V} = \frac{1364}{0.2 \times 62.4 \times 1.62 \times 3600} = 0.0187 \text{ ft}^2 \ (0.0017 \text{ m}^2)$$

Number of tubes required :

$$N_t = \frac{0.0187 \times 144}{0.0568} = 47.5$$

assume 49 tubes (7 x 7)

Mass velocity :

$$G_t = \frac{W}{a_t} = \frac{1364}{0.0187} = 72\ 900 \text{ lb/hr ft}^2 \ (99.0 \text{ kg /s m}^2)$$

Reynolds number :

$$Re = \frac{D\ G_t}{\mu} = \frac{0.0224 \times 72\ 900}{4.6} = 355$$

$j = 1.6$ (Kern p. 834)

Film coefficient

$$h_i = j \frac{k}{D} \left(\frac{c\,\mu}{k}\right)^{1/3}$$

$$= 1.6 \left(\frac{0.31}{0.0224}\right) \left(\frac{0.8 \times 1.9}{0.31}\right)^{1/3}$$

$$= 37.6 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F } (0.212 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

$$h_{io} = h_i \times \frac{I.D.}{O.D.} = 37.6 \times \frac{0.269}{0.405} = 25.0 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F}$$

$$(0.142 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

3.5.3 SHELL SIDE

assume a 2 inch baffle spacing with a square bundle of 7 tubes x 7 tubes

$$\text{shell area} = \frac{7 \times 0.2025 \times 2}{144} = 0.0197 \text{ ft}^2 \ (0.00183 \text{m}^2)$$

$$G_s = \frac{W}{a_s} = \frac{1314}{0.0197} = 66\ 700 \text{ lbm/hr ft}^2 \ (90.4 \text{ kg/s m}^2)$$

$$Re = \frac{D_e G_s}{\mu} = \frac{0.0629 \times 66\ 700}{4.6} = 913$$

$$\text{where } D_e = \frac{0.755}{12} = 0.0629 \text{ ft}$$

j = 16, (Kern p. 838)

film coefficient :

$$h_o = j \frac{k}{D} \left(\frac{c\ \mu}{k} \right)^{1/3}$$

$$= 16 \left(\frac{0.31}{0.0629} \right) \left(\frac{0.8 \times 1.9}{0.31} \right)^{1/3}$$

$$= 134 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F } (0.759 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

Heat transfer coefficient :

$$U = \frac{1}{\frac{1}{h_o} + 2R + \frac{1}{h_{io}}}$$

$$= \frac{1}{\frac{1}{134} + 0.002 + \frac{1}{25.0}}$$

$$= 20.2 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F } (0.112 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

adding 25% factor for uncertainty involved in estimation of LiBr properties

$U = 16.2\ Btu/ft^2\ hr\ ^oF\ (0.0916\ kJ/m^2\ s\ ^oC)$

3.5.4 SIZING

area :

$$A = \frac{Q}{U\ \Delta t} = \frac{51\ 200}{16.2 \times 11.9} = 266\ ft^2\ (24.7\ m^2)$$

tube area :

$$a_t = \frac{0.405 \times \times 12}{144} = 0.106\ ft^2/ft\ (0.0323\ m^2/m)$$

total length of tube

$$\frac{266}{0.106} = 2500\ ft\ (762\ m)$$

Total length of bundle

$$L = \frac{2500}{49} = 51\ ft\ (15.6\ m)$$

at 12 bends

$L = 4.25\ ft\ (1.29m) = 51\ in\ (129\ cm)$

3.5.5 PRESSURE DROPS

shell side :

number of baffles

$$N = \frac{51 \times 12}{2} = 306$$

$f = 0.0034\ ft^2/in^2\ (0.490\ cm^2/cm^2)$

at Re=913 (Kern p. 839)

shell diameter is equivalent to 7 tubes plus spaces

$$D_s = 0.405 \times 1.5 \times 7 + \frac{0.405}{2} = 4.45 \text{ in} = 0.371 \text{ ft } (0.113 \text{ m})$$

$$P_s = \frac{f\, G_s^2\, D_s\, (N+1)}{5.22 \times 10^{10}\, De}$$

$$= \frac{0.0034 \times 66700 \times 0.371 \times 307}{5.22 \times 10^{10} \times 0.0629}$$

$$= 0.525 \text{ lb/in}^2 \quad (3620 \text{ N/m}^2)$$

which can be tolerated on the shell side because the liquid is flowing from a high to a low pressure region.

The total head available is

$(65.94 \text{ mmHg} - 6.29 \text{ mmHg})\ 0.0193 = 1.15 \text{ lb/in} \quad (7930 \text{ N/m}^2)$

tube side :

$f = 0.00145 \text{ ft}^2/\text{in}^2 \ (0.209 \text{ cm}^2/\text{cm}^2)$ at Re=355 (kern p. 836)

$$P_t = \frac{f\, G_t^2\, L\, n}{5.22 \times 10^{10} D}$$

$$= \frac{0.00145 \times 72\,900^2 \times 4.25 \times 12}{5.22 \times 10^{10} \times 0.0224}$$

$$= 0.336 \text{ lb/in}^2 \quad (2320 \text{ N/m}^2)$$

3.6 GENERATOR

3.6.1 CHOICE OF TEMPERATURE

It is desirable to be able to work with as low a high temperature source as possible. The lowest practical limit for the generator is taken to be 180°F (82.2°C). Assuming that the heating water enters at 190°F (87.8°C) and that the equivalent of a 3 in. sched. 40 is used to transport the water at a velocity of 2.0 ft/sec (0.610 m/sec).

$$Q = VA$$

$$= 2.0 \times \frac{\pi}{4} (3.068)^2 \times \frac{3600}{144 \times 0.0166}$$

$$= 22\ 300 \text{ lb/hr} \quad (2.81 \text{ kg/s})$$

Temperature drop of heating water

$$\Delta t = \frac{Q}{Wc}$$

$$= \frac{65\ 400}{22\ 300 \times 1.004}$$

$$= 2.92\ ^\circ F\ (1.62^\circ C)$$

Hot fluid		Cold fluid	
T_1 = 190°F (87.8°C)	Higher temp.	t_1 = 180°F (82.2°C)	10°F (5.6°
T_2 = 187°F (86.1°C)	Lower temp.	t_2 = 180°F (82.2°C)	7°F (3.9°C
3°F (1.6°C)	Difference	0	3°F (1.6°C

LMTD = 8.4°F (4.7°C)

3.6.2 SHELL SIDE

Assuming natural circulation boiling

$T_i = 188.5^oF\ (86.9^oC)$

$T_o = 180^oF\ (82.2^oC)$

$h_o = 56\ Btu/ft^2\ hr\ ^oF\ (0.827\ kJ/m^2\ s\ ^oC)$

$$\frac{Q}{A} = U\,\Delta t = \frac{8.4}{\frac{1}{56} + \frac{1}{146}} = 340$$

Between tube wall and inside temperature

$$\Delta t = \frac{Q}{Ah} = \frac{340}{56} = 6.1\ ^oF\ (3.4^oC)$$

$h_o = 56\ Btu/ft^2\ hr\ ^oF$ (Kern p. 474)

$(273\ kcal/m^2\ hr\ ^oC)$

which is the same as what was assumed.

3.6.3 TUBE SIDE

Flow area for 0.5 ft/sec (0.122 m/s)

$$A = \frac{W}{V} = \frac{22\ 300}{0.5 \times 60.5 \times 3\ 600} = 0.205\ ft^2\ (0.0190\ m^2)$$

Number of tubes required:

$$N = \frac{0.205 \times 144}{0.0568} = 520$$

Area taken up by one tube = 2.38 cm^2 with tubes spaced at 1.5 x outside dia.

Total outside area of bundle = 191.9 in^2 = 14.8 in x 14.8 in

Mass velocity :

$$G_t = \frac{W}{a_t} = \frac{22\ 300}{0.205} = 109\ 000 \text{ lbm /hr ft}^2 \ (148 \text{ kg/s m}^2)$$

Reynolds number :

$$R_e = \frac{DG_t}{\mu} = \frac{0.269 \times 109\ 000}{12 \times 0.795} = 3070$$

where μ = 0.795 lb/ft hr (0.328cp)

(ASME steam tables p. 280)

and $$\frac{L}{D} = \frac{27}{0.269} = 100$$

j_h = 10 (Kern p. 834)

Prandtl number :

$$\left(\frac{c\ \mu}{k}\right)^{1/3} = (2.06)^{1/3} = 1.27$$

(ASME steam tables p. 282)

Heat transfer coefficient

$$h_i = j_h\left(\frac{K}{D}\right)\left(\frac{c\mu}{k}\right)^{1/3}$$

$$= \frac{10 \times 0.389 \times 12 \times 1.27}{0.269}$$

= 220 Btu/ft hr °F (1.24 kJ/m^2 s °C)

where k = 0. 389 Btu/ft hr°F (0.000672 kJ/m s °C)

(ASME steam tables p. 281)

$$h_{io} = h_i \times \frac{ID}{OD} = 220 \times \frac{.269}{.405} = 146 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F}$$

(0.827 kJ/m^2 s $^\circ$C)

3.6.4 SIZING

Overall heat transfer coefficient :

$$U = \frac{1}{\frac{1}{h_{io}} + 2R + \frac{1}{h_i}}$$

$$= \frac{1}{\frac{1}{146} + 0.002 + \frac{1}{56}}$$

= 37.4 Btu/ft^2 hr $^\circ$F (0.212 kJ/m^2 s $^\circ$C)

Area :

$$A = \frac{Q}{U \ \Delta t} = \frac{65\ 400}{37.4 \times 8.4} = 208 \text{ ft}^2 \ (19.3 \text{ m}^2)$$

$$\frac{.405 \times \pi \times 12}{144} = 0.106 \text{ ft}^2/\text{ft}$$

Total lentgh of tube :

$$L = \frac{208 \text{ ft}^2}{0.106 \text{ ft}^2/\text{ft}} = 1960 \text{ ft } (597 \text{ m})$$

Lenght of bundle :

$$L = \frac{1960 \text{ ft}}{520 \text{ tubes}}$$

= 3.77 ft (1.15m)

= 45.3 in (115cm)

3.6.5 PRESSURE DROPS

The pressure drop on the shell side is negligible because pool boiling is taking place.

Pressure drop on the tube side:

$f = 0.0004\ ft^2/in^2$ ($0.0576\ cm^2/cm^2$) at Re 3070 (Kern p. 836)

$$P_t = \frac{f\ G_t^2\ Ln}{5.22 \times 10^{10}\ D}$$

$$= \frac{0.0004 \times 10\ 9000^2 \times 3.77 \times 1}{5.22 \times 10^{10} \times 0.0224}$$

$$= 0.0153\ lb/in^2\ (105\ N/m^2)$$

3.7 CONDENSER

3.7.1 GENERAL

The condenser will be contained in a vessel which also includes the generator. Condensation of pure steam takes place outside horizontal tubes. The condensate drips through the tubes and is collected by a condensate collecting pan. Desuperheating of steam is accomplished by additional tube surface on the same bundle.

Assuming that cooling water comes in at 80°F (26.7°C)

$$t = \frac{Q}{cW} = \frac{53\,400}{0.998 \times 6730} = 7.95\ ^{\circ}F\ (4.4^{\circ}C)$$

where c = 0.998 Btu/lb °F (4.17 kJ/kg °C) at 85°F (29.4°C)

(ASME steam tables p. 278)

Desuperheating

Hot fluid		Cold fluid	
T_1 = 180°F (82.2°C)	higher temp.	t_2 = 87.95°F (31.1°C)	92°F (51.1
T_2 = 110°F (43.3°C)	lower temp.	t_1 = 80°F (26.7°C)	30°F (16.7
70°F (38.9°C)	Difference	7.95°F (4.6°C)	62°F (34.4

LMTD = 55.3°F (30.7°C)

Latent heat transfer

Hot fluid		Cold fluid	
T_1 = 110°F (43.3°C)	Higher temp.	t_2 = 87.95 (31.1°C)	22.1°F (12.3
T_2 = 110°F (43.3°C)	Lower tmep.	t_1 = 80°F (26.7°C)	30°F (16.7°C
0	Difference	7.95°F (4.6°C)	7.95°F(4.4°C

LMTD = 25.8°F (14.3°C)

Enthalpy of steam at 110°F (43.3°C)

h = 1109 Btu/lbm (2570 kJ/kg)

Heat transferred during desuperheating process

$Q = W (h_7 - h)$

$= 49.95 (1138 - 1109)$

$= 1400$ Btu/hr (0.409 kJ/s)

Latent heat

$Q = W (h - h_8)$

$= 49.95 (1109 - 77.98)$

$= 51\ 500$ Btu/hr (1.51 kJ/s)

Total heat

Q = 52 900 Btu/hr (15.4 kJ/s)

Percentage sensible heat : 2.6%

3.7.2 LATENT HEAT AREA

Assuming a condensing film coefficient of 4320 Btu/ft^2 hr °F (24.5 kJ/m^2 s °C) and a tube side coefficient of 84.4 Btu/ft^2 hr °F (0.478 kJ/m^2 °C)

$$U = \frac{1}{\frac{1}{4320} + 0.002 + \frac{1}{75.1}} = 64.3 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F} \quad (0.364 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

$$A = \frac{Q}{U_D \Delta t} = \frac{51\ 500}{64.3 \times 25.8} = 31.0 \text{ ft}^2 \ (0.00300 \text{m}^2)$$

3.7.3 SENSIBLE HEAT AREA

Assuming a shell side film coefficient of I[illegible]2 Btu/ft^2 hr ^oF (0.00804 kJ/m^2 s ^oC)

$$U_o = \frac{1}{\frac{1}{1.42} + 0.002 + \frac{1}{75.1}} = 1.39 \text{ Btu/ft}^2 \text{ hr } ^oF \quad (0.00786 \text{ kJ/m}^2 \text{ s } ^oC)$$

$$\text{Area : } A = \frac{Q}{U_D \Delta t} = \frac{1400}{1.39 \times 55.3} = 18.2 \text{ ft}^2 \ (1.69 \text{m}^2)$$

Total area

$A = 31.0 + 18.2 = 49.2$ ft^2 (4.57m^2)

Percentage area for sensible heat :

$$\frac{18.2 \times 100}{49.2} = 37\%$$

3.7.4 BUNDLE SIZE CALCULATION

Assuming a cooling water of 0.5 ft/sec (0.1520 m/s) velocity, flow area required:

$$a_t = \frac{W}{V} = \frac{6730 \times 0.0161}{0.5 \times 3600} = 0.0602 \text{ ft}^2 \ (0.00559 \text{ m}^2)$$

where v = 0.0161 ft^3/lbm (0.00100 m^3/kg) at 84°F (28.9^{o}C)

(ASME steam tables p. 88)

Area per tube :

$$A = \frac{\pi}{4} \quad \frac{0.269^2}{12} = 0.000395 \text{ ft}^2 \ (0.0000367 \text{ m}^2)$$

Number of tubes:

$$N = \frac{0.0602}{0.000395} = 152.4$$

Assuming that 63% of these are for latent heat

$N_\ell = 96$

Sensible and latent area per foot of bundle

$$A = \frac{0.405 \times \pi \times 152.4}{12} = 16.1 \text{ ft}^2 \ (1.49\text{m}^2)$$

Length of bundle:

$$L = \frac{49.2}{16.1} = 3.05 \text{ ft } (0.930 \text{ m}) = 36.7 \text{ in } (93.0\text{cm})$$

Average area taken up by tubes:

$$A = 152.4\ (1.5 \times 0.405)^2 = 56.2 \text{ in}^2\ (363 \text{ cm}^2)$$

Assuming a square bundle:

$l = 7.50$ in (19.1cm)

Allowing an extra 1/8 in at each side

$l = 7.75$ in (19.7 cm)

3.7.5 <u>TUBE SIDE</u>

Flow area at 0.5 ft/sec (0.152 m/s)

$a_t = 0.0602 \text{ ft}^2\ (0.00559 \text{ m}^2)$

Mass velocity :

$$G_t = \frac{w}{a_t} = \frac{6730}{0.0602} = 112\ 000 \text{ lb/hr ft}^2$$

$$(152 \text{ kg/s m}^2)$$

Reynolds number

$$Re_t = \frac{DG_t}{\mu} = \frac{0.0224 \times 112\ 000}{2.12} = 1180$$

where μ = 2.12 lb/ft hr (89.4 x 10^{-6} Ns/m^2)
at 84°F (28.9°C) (ASME steam tables p. 286)

and $D = \frac{0.269}{12} = 0.0224$ ft (0.0068 m)

Heat transfer factor where $\frac{L}{D} = \frac{37.3}{0.269} = 139$

j = 3.9 (Kern p. 834)

Prandtl number at 84°F (28.9°C)

$$\frac{(c\ \mu)^{1/3}}{k} = (6.04)^{1/3} = 1.82$$

(ASME steam tables p. 282)

Coefficient of heat transfer

$$h_i = j_h\left(\frac{k}{D_e}\right)\left(\frac{c\ \mu}{k}\right)^{1/3}$$

$$= \frac{3.9 \times 0.356 \times 1.82}{0.0224} = 113 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F } (0.369 \text{ kJ/m}^2 \text{ s } ^\circ\text{C})$$

where k = 0.356 Btu/ft hr °F (0.000615 kJ/m s °C)
at 84°F (28.9°C) (ASME steam tables p. 281)

$$h_{io} = 113 \times \frac{I.D.}{O.D.} = \frac{113 \times 0.269}{0.405} = 75.1 \text{ Btu/ft}^2 \text{ hr } ^\circ\text{F}$$

(0.425 kJ/m^2 s °C)

which is the same as what was assumed for area calculation

ORIGINALS MARKED

3.7.6 SHELL SIDE CONDENSATION

Loading :

$$G'' = \frac{M}{LN_{t_1}^{2/3}} = \frac{44.95}{3.05 \times 96^{2/3}} = 0.703$$

(Kern EQ 12.43 p. 266)

to determine film temperature

assume h_o = 4320 Btu/ft^2 hr ^oF (24.5 kJ/m^2 s ^oC)

h_{io} = 75.1 Btu/ft^2 hr ^oF (0.425 kJ/m^2 s ^oC)

$$\frac{Q}{A} = U \Delta t = \frac{26}{\frac{1}{4320} + \frac{1}{75.1}} = 1919$$

$$\Delta t = \frac{Q}{Ah} = \frac{1919}{4320} = 0.4$$

t_w = 110 - 0.4 = 109.6 ^oF (43.1^oC)

$$t_f = \frac{T_v + t_w}{2} = \frac{110 + 109.6}{2} = 110^oF \ (43.3^oC)$$

where k_f = 0.367 Btu/ft hr ^oF (0.000634 kJ/m s ^oC)
at 110^oF (43.3 ^oC) (ASME steam tables p. 281)
μ_l = 1.52 lb/ft hr (.0000643 Ns/m^2)
(ASME steam tables p. 280)
ρ_l = 61.9 lbm/ft^3 (992 kg/m^3) (ASME steam tables p. 87)

$$h\left(\frac{\mu_l^2}{k_l \rho_l^2 g}\right)^{1/3} = 1.5\left(\frac{4G''}{\mu_l}\right)^{-1/3}$$ (Kern EQ 12.42 p. 266)

$$h\left(\frac{0.398}{0.0494 \times 3832 \times 417\ 000\ 000}\right)^{1/3} = 1.5\left(\frac{4 \times .703}{0.631}\right)^{-1/3}$$

h = 5356 where g = (32.2 × 3600 × 3600) ft/hr

which is very close to assumption of 4320 because the difference in heat transfer rate is negligible.

3.7.7 SHELL SIDE DESUPERHEATING

Shell area: a_s = shell I.D. x tube spacing x length / tube pitch

$$= \frac{7.75 \times 0.2025 \times 38.7}{144 \times 0.6075} = 0.658 ft^2 \ (0.0611 m^2)$$

Mass velocity:

$$G_s = \frac{W}{a_s} = \frac{49.95}{0.658} = 75.9 \ lb/hr \ ft^2$$

(0.103 kg / s m^2)

Mean temperature for desuperheating

$$t = \frac{180 + 110}{2} = 145^oF \ (62.8 \ ^oC)$$

where μ = 0.0260 lbm/ft hr (1.09 x 10^{-6} Ns/m^2) at 145 ^oF (62.8^oC)

(ASME steam tables p. 280)

$$D_e = \frac{0.755}{12} = 0.0629 \ ft \ (0.0192 \ m)$$

$$Re_s = \frac{D_e G_s}{\mu} = \frac{0.0629 \times 75.9}{0.0260} = 183$$

j_h = 7.0 (Kern fig. 28 p. 838)

Prandtl number at 145^oF (62.8^oC)

$$\left(\frac{c \ \mu}{k}\right)^{1/3} = (0.93)^{1/3} = 0.976$$

(ASME steam tables p. 282)

Outside heat transfer coefficient :

$$h_o = j_h \left(\frac{k}{De}\right)\left(\frac{c \ \mu}{k}\right)^{1/3} = \frac{7.0 \times 0.0127 \times 0.976}{0.0629}$$

$= 1.38$ Btu/ft^2 hr ^oF (0.00781 kJ/m^2 s ^oC)

where $k = 0.0127$ Btu/ft hr ^oF (0.0000219 kJ/m s ^oC)
at 145^oF (62.8 ^oC) (ASME steam tables p. 281)

which is similar to the heat transfer coefficient assumed

<u>Note</u>: recalculation of film coefficient using Fig 10.4 of Kern yields similar results.

3.7.9 PRESUURE DROPS

The pressure drop on the shell side is negligible.

Pressure drop on the tube side:

$f = 0.00052$ ft^2/in^2 (0.075 cm^2/ cm^2) at Re=1180

(Kern p. 836)

$$P_t = \frac{f G_t^2 L n}{5.22 \times 10^{10} D}$$

$$= \frac{0.00052 \times 112\,000^2 \times 3.05 \times 1}{5.22 \times 10^{10} \times 0.0224}$$

$= 0.017$ lbf/in^2 (118 N/m^2)

CONCLUSIONS AND SUGGESTIONS

The system is designed to have an evaporator capacity of 50 000 Btu/hr (14.6 kJ/s) . This size requirement determines the use of shell and tube equipment rather than double pipe exchangers. It is general practice in units such as this to combine the evaporator and absorber in a single vessel as well as the generator and condenser. This constraint dictates the use of circulating water to carry heat energy to and from the unit.

Construction of the unit would be in accordance with fig. 5 . The length of the individual heat exchangers was kept under 4.5 ft (1.4 m) to avoid excessive unit length. Exchanger pressure drops were kept as low as possible to allow pumps of low power consumption to be used.

Redesign of the unit for the lowest possible cooling temperature would necessitate a complete modification of the evaporator in order to accomodate direct expansion. This would require the evaporator to be separate from the absorber. Assuming a Δt of $10^{\circ}F$ ($5.6^{\circ}C$) a space could not however be cooled to much less than $50^{\circ}F$ ($10^{\circ}C$).

Changing the generator heat source from a solar energy source to steam would considerably reduce the size of the generator heat transfer surface and would necessitate a redesign of the generator as well as a revision of the mass and energy balance. A direct fired generator would require further modification with the likelihood that the generator would have to be independent of the condenser.

REFERENCES

(1) A.S.H.R.A.E.
Handbook of Fundamentals
New York: A.S.H.R.A.E. , 1972

(2) A.S.M.E.
1967 ASME steam tables
New York, 1967

(3) Jennings, B.H.
Environmental Engineering
Scranton, Penn.: International Textbook Company, 1970

(4) Kern, D.Q.
Process Heat Transfer
New York: McGraw-Hill, 1950

(5) Kreider, J.F. and Kreith, F.
Solar Heating and Cooling
New York: McGraw-Hill, 1975

(6) Weil, S.A.
Correlation of the LiSCN-LiBr-H_2O Thermodynamic Properties
A.G.A. Symposium on Absorption Air-conditioning Systems, Chicago Feb. 6 , 1968

(7) Wood, B.D.
Applications of Thermodynamics
Reading, Mass.: Addison-Wesley, 1969