DESIGN OF AN ABSORPTION REFRIGERATION UNIT FOR DOMESTIC AIR CONDITIONING SERVICE

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DESIGN OF AN ABSORPTION REFRIGERATION UNIT FOR DOMESTIC AIR CONDITIONING SERVICE

INTRODUCTION

The tremendous growth of the air conditioning field during the past several years has resulted in a demand for an economical heat-operated refrigerating unit.

Commercially available heat-operated refrigeration units may be divided into the following three broad classifications:

1. Steam-driven compression machines.
2. Steam jet machines.
3. Absorption machines.

Due to either high first cost or high operating cost, particularly at low or medium steam pressures, the first two of the above listed types do not meet the requirement of economy, and are therefore ruled out as possible units for air conditioning service. Nor is the old absorption unit using ammonia as the refrigerant and water as the absorbent suited for this type of service since it has similar handicaps which will be considered later. In addition, ammonia is not a suitable refrigerant for air conditioning since it is toxic and is inflammable.

In an effort to meet the demand, the refrigeration industry has developed several new absorption machines. Of commercial interest at the present time are the units

developed by Williams Oil-O-Matic Company which use methylene chloride as the refrigerant and dimethyl ether of triethylene glycol as the absorbent, and those developed by Servel, Inc., and Carrier Corporation which use water as the refrigerant and an aqueous solution of lithium bromide as the absorbent.

THE ABSORPTION REFRIGERATION UNIT

Principles of Absorption Refrigeration. The absorption cycle makes use of the affinity existing between certain substances which results in one substance being absorbed by the other. As an example of this phenomenon, consider two vessels, one containing a hygroscopic salt and the other containing saturated water. If these two vessels are closed and interconnected, the salt will absorb water vapor, thereby reducing the pressure over the water, causing it to boil. The heat required to produce this additional vapor is abstracted from the water itself or from its surroundings, thus producing a refrigerating effect. When sufficient water vapor has been absorbed by the salt, a liquid solution will result. This solution, however, may still retain its absorptive powers, depending upon the concentration of salt in the solution.

The principles of absorption as outlined above are directly comparable to those employed in the continuous absorption refrigerating system, shown schematically in Fig. 1. In the refrigeration cycle the source of water vapor is the evaporator, and the absorbing medium --a concentrated liquid solution of salt and water--is sprayed into the absorber. Here it absorbs the vapor flowing from the evaporator, thereby reducing the salt

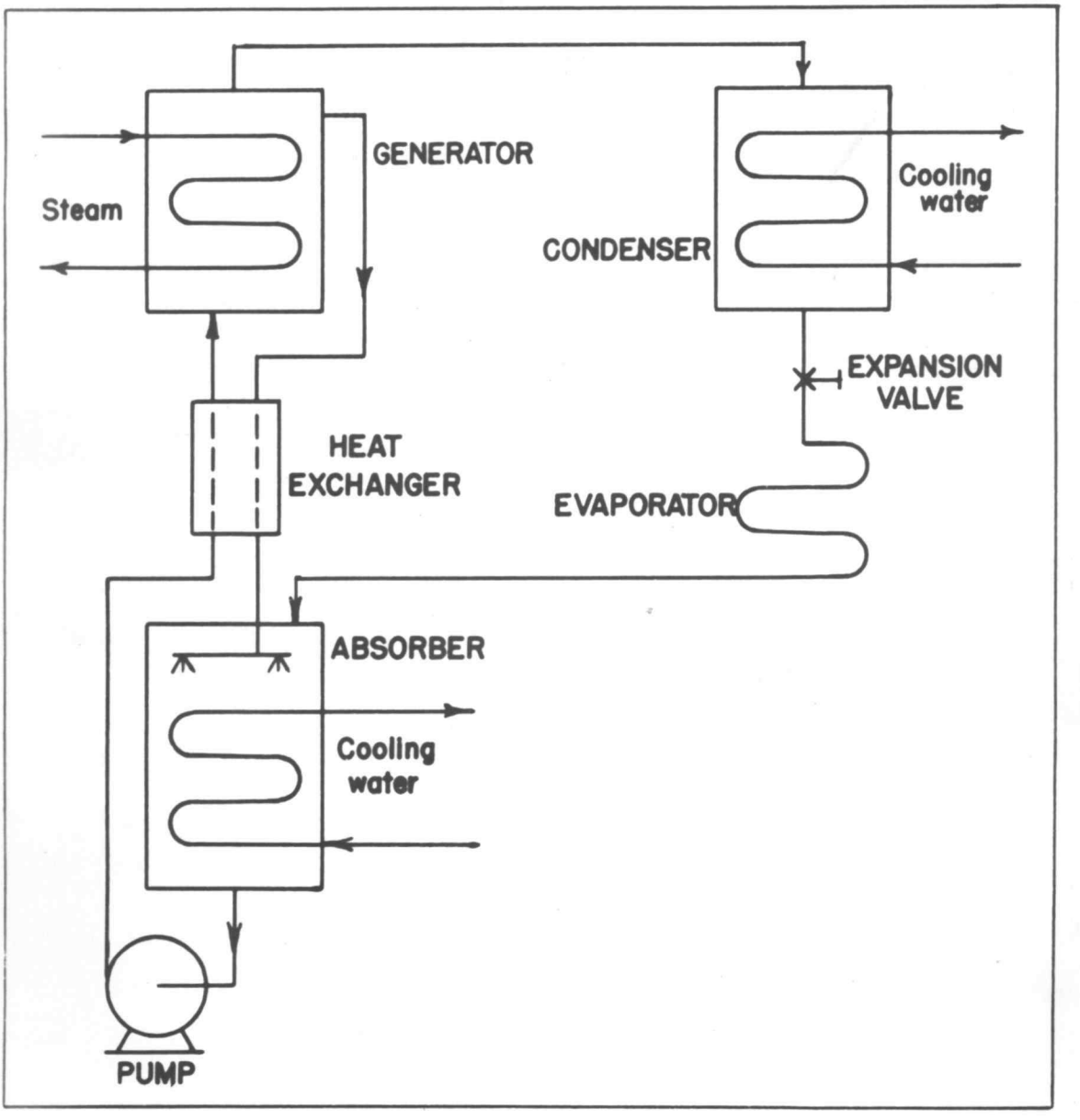


Fig. 1. Schematic diagram of simple absorption system apparatus.

concentration of the solution. This dilute solution then flows to the generator where heat is supplied which drives off the excess water vapor. The reconcentrated solution is then returned to the absorber, while the water vapor which has been driven off flows to the condenser where it is condensed and is then returned to the evaporator, thus completing the cycle. Cooling water is used in the condenser and absorber to remove the heat evolved by condensation at these points. The liquid-to-liquid heat exchanger shown in Fig. 1 serves a dual purpose: (1) it preheats the dilute solution entering the generator, thereby increasing the efficiency of the unit, and (2) it reduces the temperature of the reconcentrated solution entering the absorber to a value low enough to prevent "flashing" when the pressure is reduced to that maintained in the absorber.

The absorption cycle using ammonia as the refrigerant and aqua ammonia as the absorbent is necessarily more complicated than that described above, due to the fact that some water is boiled off with the ammonia even though the boiling point of water is higher than that of ammonia. This water vapor, after being liquified in the condenser, goes through the evaporator and on into the absorber as a liquid without producing refrigeration. This, of course, results in an increased load on the generator

and condenser with no increase in refrigerating effect, which reduces the efficiency of the cycle.

Two additional pieces of equipment are usually included in the ammonia system to reduce the amount of water in the condenser and evaporator; an analyzer and a rectifier. The analyzer consists of a chamber through which the vapors leaving the generator pass in counterflow contact with the strong solution entering the generator. This reduces the temperature of the leaving vapor, thereby increasing its ammonia content, and at the same time raises the temperature of the solution entering the generator. The rectifier is actually the inlet part of the condenser arranged in such a manner that its condensate is returned directly to the analyzer or generator.

The addition of this equipment makes the machine more complicated and more expensive, and although the efficiency of the cycle is improved, it still remains comparatively low. As a result, the ammonia-water absorption machine is high in first cost and in operating cost, which makes its application for air conditioning uneconomical even if it were not already excluded due to the hazardous nature of ammonia.

Selection of the Absorbent-Refrigerant Combination. For reasons already given, steam-driven compression units, steam jet units, and absorption units employing the

ammonia-water combination are not suitable for air conditioning use. It was therefore necessary to investigate the possibility of developing an absorption system using refrigerant-absorbent combinations other than ammonia-water which would meet the requirements of this application. This investigation resolved itself into three steps: establishing the requirements for suitable combinations, screening of possible combinations to eliminate those which are worthless, and testing in detail those not eliminated by screening.

The basic requirements for successful absorbent-refrigerant combinations are:

1. The solubility of the refrigerant in the absorbent must be high at absorber temperature and at a pressure corresponding to evaporator temperature.

2. The absorption of the refrigerant vapor should be rapid, and the actual concentration of rich liquid should approach the equilibrium value.

3. The absorbent should be non-volatile or at least much less volatile than the refrigerant so that only simple distillation is required for driving off essentially pure refrigerant vapor.

4. The viscosities of the weak and rich solutions should be low to facilitate heat transfer, fluid flow, and mixing. High solubility favors low viscosity since

it further dilutes the more viscous fluid--the absorbent.

5. The freezing points of the liquids at compositions encountered at all points in the cycle must be lower than the lowest temperature to which they are subjected. If salt solutions are used as the absorbent, conditions under which salt would tend to crystallize from the solution must be avoided.

6. All components must be stable and resist all irreversible processes such as decomposition and corrosion.

The type and number of basic requirements naturally lead to a question of whether or not these requirements are in conflict with one another to such an extent that no combination could meet all of them. As a matter of fact the necessity of having an absorbent of low volatility and, at the same time, low viscosity are directly opposed since most fluids freeze or become viscous at temperatures within 200 F of their boiling points. Under point 4 it was stated that high solubility favors low viscosity. It therefore follows that this divergence of requirements is not serious in cases where solubility is high since high viscosity should not then be a problem even though the high volatility requirement is met.

Of all the requirements listed, by far the most important is the first; extraordinarily high solubility.

A method of comparison of the relative solubilities of various possible refrigerant-absorbent combinations has been devised based on Raoult's Law. This law, which applies to the major component or solvent, states that the partial vapor pressure of a component whose mol fraction in solution is sufficiently near unity equals the product of the mol fraction and the vapor pressure of the pure solvent. Expressed in the form of an equation, this becomes:

$$P = N P_o$$

where

P - Partial vapor pressure of major component.

P_o - Vapor pressure of pure solvent.

N - Mol fraction of major component in solution.

Perfect solutions--those for which Raoult's Law applies to all components over the entire range of compositions--result when the molecular forces surrounding the solvent and solute molecules resemble each other so closely that there are no forces of attraction or repulsion between solvent and solute molecules. If the partial vapor pressure of the solvent exceeds that predicted by Raoult's Law at a given concentration, the deviation is said to be positive and the solubility is less than it would be if the solution were perfect. Conversely, if the partial vapor pressure of the solvent is less than

that predicted by the law, the deviation is said to be negative and the component is more soluble than it would be for a perfect solution. In the interests of high solubility, therefore, it is essential that a combination be selected which has a high negative deviation.

Also, small and simple absorbent molecules have advantages over larger, more complex ones. Since they are small and simple they show less tendency to decompose and have low molal specific heats and relatively low viscosities. All refrigerants have comparatively small molecules since fluids having large molecules are not volatile enough. The two successful refrigerant-absorbent combinations, NH_3 - H_2O and H_2O - LiBr, are typical of combinations which have these advantages.

Since there is a large number of possible refrigerant-absorbent combinations which could be investigated, several screening tests have been proposed which are designed to separate those combinations which have desirable characteristics from those which do not. One method developed by Dr. R. M. Buffington is worthy of note. This method, called the temperature rise test, consists of mixing equal volumes of the pure liquids, refrigerant and absorbent, at room temperature and under pressure if necessary to prevent evaporation, and noting the practically instantaneous temperature rise. The two liquids

should dissolve in each other to form a single liquid phase, with no sign of another phase such as gum, solid, or vapor other than bubbles of pure refrigerant. Dr. Buffington recommends that those combinations which show a temperature rise of as much as 15 to 20 F and which do not form a second phase be further investigated, while those showing no temperature rise or which produce a second phase may be immediately eliminated (4, pp.384-386).

On the basis of the requirements outlined and by use of tests such as that mentioned, investigators have found that the combination of water as the refrigerant and a concentrated solution of lithium bromide and water as the absorbent is particularly well suited for high temperature refrigeration such as that encountered in air conditioning applications. Lithium bromide is practically non-volatile at temperatures encountered, therefore, no extra equipment such as analyzers and rectifiers are required. It has a high affinity for water, as indicated by a high negative deviation from the theoretical partial vapor pressure predicted by Raoult's Law, thus indicating an extraordinarily high solubility. Both components are stable and resist decomposition, and corrosive tendencies may be controlled by careful selection of materials and possible use of corrosion inhibitors.

Primary Design Considerations. What refrigerant-absorbent combination to use was the first decision to be made regarding the design of an absorption refrigerating unit for domestic air conditioning service. Several factors influenced the decision to use water as the refrigerant and an aqueous solution of lithium bromide as the absorbent, these factors being:

1. The cycle uses low temperature heat as an energy source, and is therefore particularly well suited to operation on low pressure steam.

2. Since the unit uses the latent heat of the steam, the steam rate is practically constant regardless of steam pressure.

3. It is conceivable that solar heat may be an energy source worthy of consideration.

4. The cycle operates at high efficiency over a wide range of load.

5. The cycle uses a cheap refrigerant (water) and the cost of lithium bromide is not prohibitive.

An estimate of the required capacity of the unit was based on the following reasoning: The capacity of the heating plant installed in the dwelling was assumed to be 100,000 BTU/hr and was assumed to have been based on design conditions of an outside temperature of 10 F and an inside temperature of 70 F. It was further assumed

that the maximum desired outdoor-indoor temperature difference for air conditioning would be 15 F. On this basis a sensible heat load of 25,000 BTU/hr would be imposed on the refrigerating unit. To allow for additional heat loads not included above, such as the latent heat of moisture condensed, additional heat load from cooking, etc., the estimated required capacity of the refrigerating unit was set at 3 tons (36,000 BTU/hr).

It was further assumed that steam at 5 psig (228 F) was available as an energy source for the unit, and that cooling water at 85 F was available for condenser and absorber cooling. The desired evaporator temperature was set at 45 F, which requires an absolute pressure of 0.3 in. Hg.

Information regarding the relationship of temperature, vapor pressure, and concentration for aqueous lithium bromide solutions was taken from the International Critical Tables and is given in Table I (13, 3:369). The maximum solubility of lithium bromide in water at various temperatures is given in Table II (16, 1:898). Information contained in these tables is shown graphically in Fig. 2. From this figure it can be seen that at 0.3 in. Hg the maximum concentration that could be used would be about 64% Li Br by weight. To insure against any possibility of crystallization, the maximum

TABLE I

VAPOR PRESSURE OF AQUEOUS LiBr SOLUTIONS

(Vapor Pressure in Inches Hg)

Conc. % LiBr by Wt.	Temperature, F					
	32	68	104	140	176	212
0	0.1803	0.6903	2.178	5.88	13.98	29.92
9.09	0.1733	0.652	2.085	5.645	13.43	28.7
16.67	0.1615	0.623	1.97	5.33	12.68	27.15
28.6	0.1340	0.52	1.65	4.5	10.78	23.25
37.5	0.0977	0.398	1.284	3.53	8.54	18.62
44.4	0.0709	0.2835	0.926	2.58	6.36	14.04
50	0.0473	0.193	0.638	1.808	4.51	10.09
54.6	0.0315	0.130	0.433	1.244	3.13	7.11
58.35	0.022	0.0906	0.303	0.870	2.21	5.08
61.5		0.063	0.2165	0.626	1.616	3.765
64.3			0.1575	0.468	1.228	2.935
66.7			0.122	0.366	0.988	2.44
68.8				0.295	0.808	2.03
70.6					0.684	1.744

TABLE II

SOLUBILITY OF LiBr IN WATER

Temperature, F	Maximum LiBr Dissolved	
	lb LiBr/100 lb H_2O	% Wt LiBr
32	118.5	54.3
50	137.7	57.9
68	146.7	59.5
86	158.3	61.3
104	170.0	63.0
122	177.4	64.0
140	185	64.9
176	203	67.0
212	220	68.8

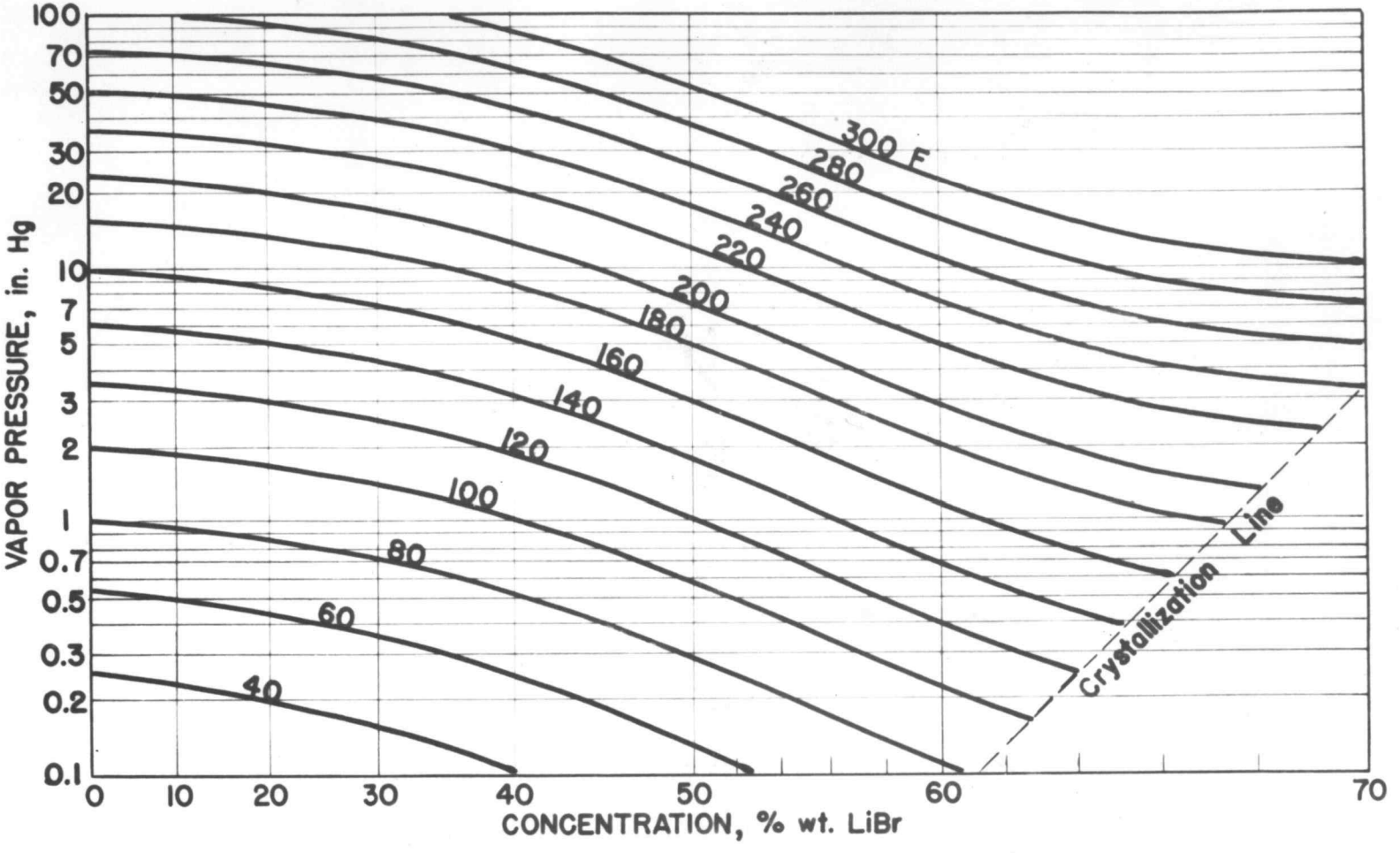


Fig.2. Relationship of temperature, vapor pressure, and concentration for LiBr solutions.

concentration to be used in the unit was set at 60%.

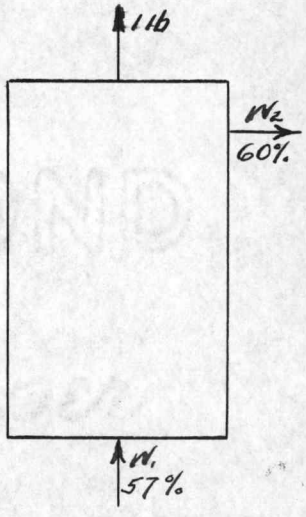
The concentration of the weak solution leaving the absorber was set at 57%. It is believed that this range of concentrations would provide rapid absorption of the refrigerant vapor without hindering the efficiency by requiring an excessive amount of solution to be heated in the generator per pound of refrigerant evaporated.

To maintain a high terminal difference in the generator the generator pressure was set at 2.6 in. Hg. From Fig. 2 it is evident that this pressure corresponds to a temperature of about 192 F at a concentration of 60%.

To determine the amount of solution flowing between the generator and the absorber, the following scheme was used:

Let: W_1 = Weight of 57% solution entering generator per lb of water evaporated

W_2 = Weight of 60% solution leaving generator per lb of water evaporated.



Then, the weight of fluid entering the generator is equal to the weight of the fluid leaving the generator:

$$W_1 = W_2 + 1,$$

and the weight of LiBr entering is equal to the weight of LiBr leaving:

$$0.57W_1 = 0.60W_2.$$

Simultaneous solution of these equations gives

$W_1 = 20$ lb

$W_2 = 19$ lb.

To determine the heat added or rejected in each unit of the cycle, it was necessary to establish the temperature of the fluid at various points in the system and to know the specific heat and the heat of solution at various concentrations. These properties are given in Table III and shown graphically in Figs. 3 and 4 (11, p.146-147).

TABLE III

HEAT OF SOLUTION AND SPECIFIC HEAT

OF AQUEOUS LiBr SOLUTIONS

Concentration	Heat of Solution	Specific Heat at 77 F
% LiBr by Wt.	BTU/lb LiBr	BTU/lb sol. F
0		1.000
30.25	0.496	0.678
32.5	0.493	0.659
34.7	0.489	0.639
36.7	0.486	0.620
38.6	0.482	0.604
40.3	0.478	0.588
42.0	0.473	0.574
45.1	0.464	0.549
47.9	0.453	0.528
50.4	0.441	0.510
52.6	0.428	0.494
54.7	0.414	0.478
56.6	0.399	0.467
58.3	0.384	0.455
59.9	0.369	0.444
60.9	0.359	0.438

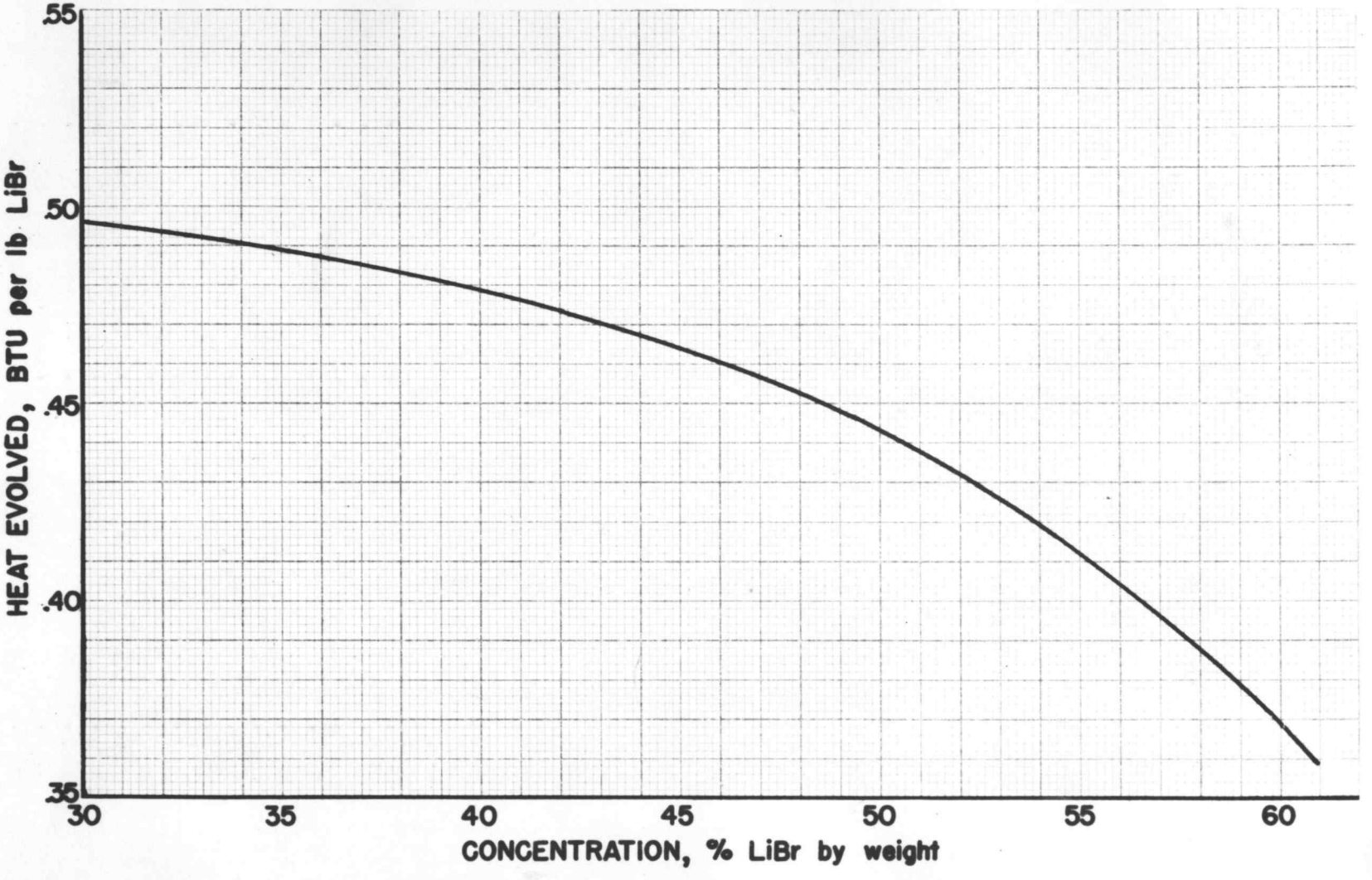


Fig. 3. Heat of solution of LiBr and water.

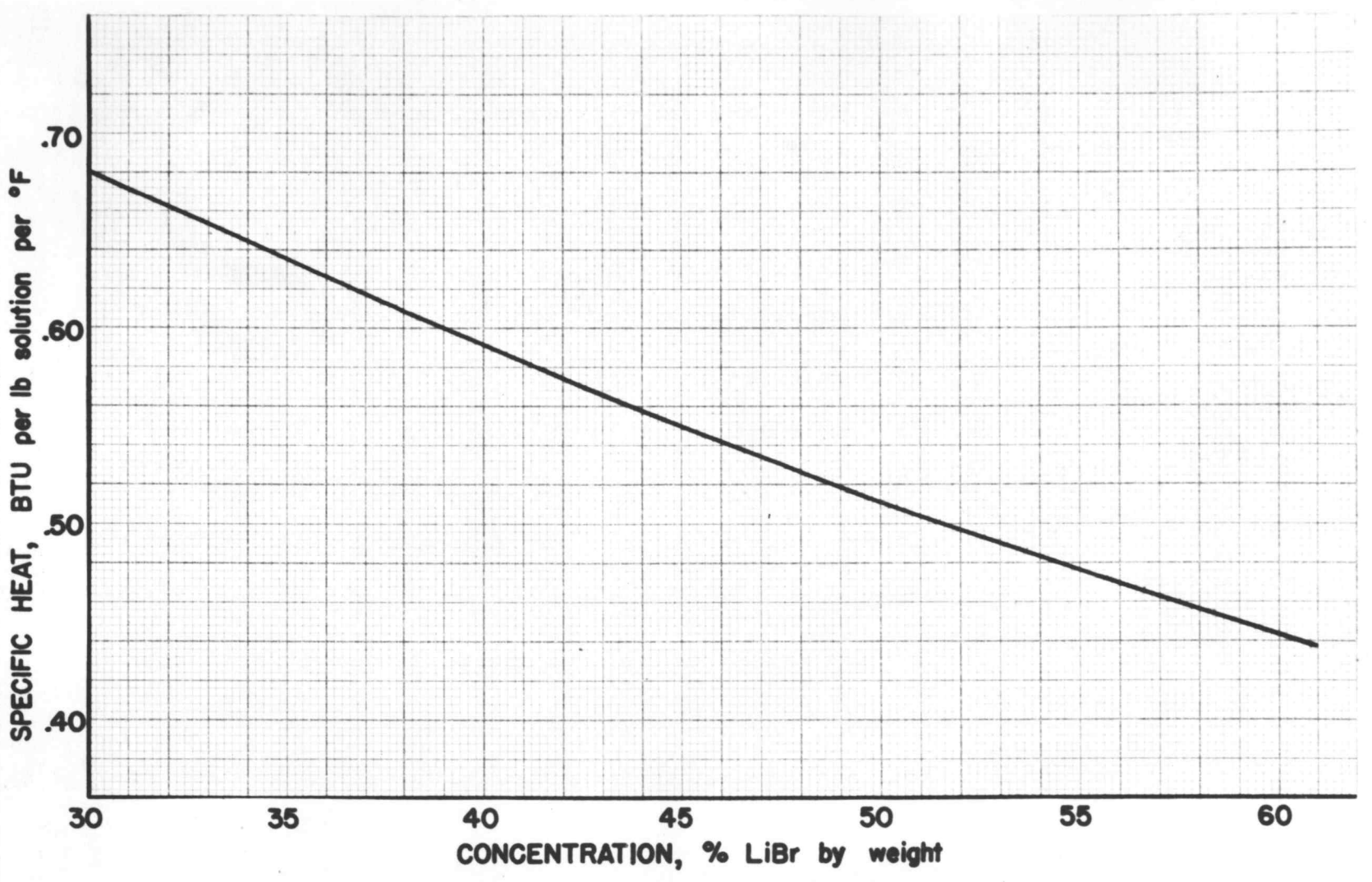
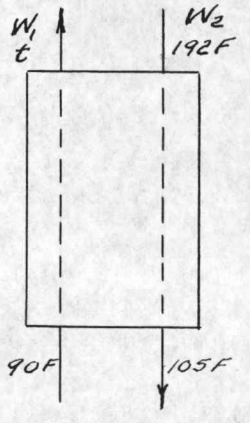


Fig. 4. Specific heat of aqueous solutions of LiBr at 77 °F

The temperature of the 60% solution leaving the generator has been set at 192 F. From Fig. 2 it is evident that the temperature of the solution entering the absorber (at 0.3 in. Hg) must be not greater than 110 F if flashing is to be prevented. This temperature was therefore set at 105 F, allowing a five degree margin of safety. In the same manner the temperature of the 57% solution leaving the absorber was set at 90 F.

In the heat exchanger, the heat given up by the hot solution is taken up by the cold solution, and the following equation may be written



$$W_1 C_1 \Delta t_1 = W_2 C_2 \Delta t_2.$$

Using specific heat values from Fig. 4, this equation becomes:

$$(20)(0.463)(t-90)=(19)(0.424)(192-105)$$

$$t = 165.8 \text{ F.}$$

With this, the temperature of the fluids entering and leaving each unit in the cycle was known and the heat added or rejected could be computed. This was done as follows, basing all calculations on a flow of one pound through the evaporator:

1. Heat removed in absorber, Q_a:

   (a) To condense vapor:

$$Q = h_{g45\text{ F}} - h_{f90\text{ F}}$$

$= 1081.5 - 58.0 = 1023.5$ BTU

(b) To cool solution:

$Q = WC\Delta t$

$= (19)(0.424)(105-90) = 126.7$ BTU

(c) Heat of solution:

$Q = (0.369)(0.60)(19) = 4.2$ BTU

Total heat removed in absorber, $Q_a = 1154.4$ BTU.

2. Heat removed in condenser, Q_c:

$Q_c = h_{g192\ F} - h_{f2.6\ in.\ Hg}$

$= 1146.7 - 77.9 = 1068.8$ BTU.

3. Heat added in evaporator, Q_e:

$Q_e = h_{g45\ F} - h_{f2.6\ in.\ Hg}$

$= 1081.5 - 77.9 = 1003.6$ BTU.

4. Heat added in generator, Q_g:

$Q_g + Q_e = Q_a + Q_c$

$Q_g = 1154.4 + 1068.8 - 1003.6$

$= \underline{1219.6}$ BTU

With absorption refrigeration the efficiency of the cycle is expressed as the "performance ratio", which is defined as the ratio of the useful refrigerating effect to the heat input. In the symbols used, this could be expressed:

$R = Q_e/Q_g$

$= 1003.6/1219.6$

$= \underline{82.4\%}$

DESIGN AND SELECTION OF EQUIPMENT

Theory. From the sketch of the absorption system apparatus it is obvious that the problem of design of a unit of this type is fundamentally an application of the theory of heat transfer involving the steady-state transfer of heat by conduction. This type of heat flow is expressed by the Fourier equation

$$Q = -kA \frac{dt}{dL}$$

where:

Q - Heat transferred by conduction per unit of time

A- Surface area

t - Temperature

L - Thickness of the material

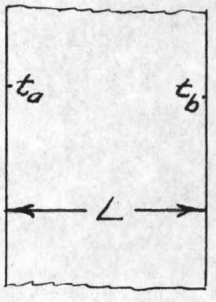
k - Constant of proportionality, called the thermal conductivity coefficient.

For a plane wall of a single homogeneous material, the heat transferred by conduction per unit of time may be obtained by separating variables and integrating between the limits of L = 0 to L = L, as follows:

$$Q = -kA \frac{dt}{dL}$$

$$Q \int_0^L dL = -kA \int_{t_a}^{t_b} dt$$

$$Q = \frac{kA(t_a - t_b)}{L}$$



On each side of the wall just considered there is a fluid. Adjacent to the surfaces of the wall this fluid creates a thin film which provides a barrier to heat flow. The quantity of heat which may be transferred through this film may be expressed by the expression

$$Q = hA\Delta t$$

where:

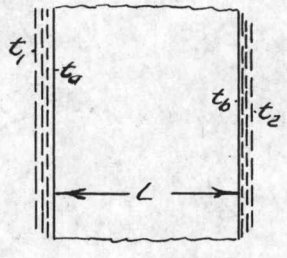
Q - Heat transferred through the film per unit of time

h - Film heat transfer coefficient

A - Surface area

Δt - Change in temperature across the section of the film.

Including the effect of the films, the equations of heat transfer through a plane wall are:



$$Q = h_1A(t_1 - t_a)$$

$$Q = \frac{kA(t_a - t_b)}{L}$$

$$Q = h_2A(t_b - t_2).$$

Rearranging terms, these equations become:

$$t_1 - t_a = \frac{Q}{h_1A}$$

$$t_a - t_b = \frac{QL}{kA}$$

$$t_b - t_2 = \frac{Q}{h_2A}$$

By adding these three equations together and noting that

the quantity of heat transferred through each section, Q, is identical in each expression, the following result is obtained:

$$t_1 - t_2 = \frac{Q}{A}\left(\frac{1}{h_1} + \frac{L}{k} + \frac{1}{h_2}\right)$$

$$Q = \frac{A(t_1 - t_2)}{\left(\frac{1}{h_1} + \frac{L}{k} + \frac{1}{h_2}\right)}$$

If, in the above equation, we let:

$$U = \frac{1}{\left(\frac{1}{h_1} + \frac{L}{k} + \frac{1}{h_2}\right)},$$

it would become:

$$Q = UA(t_1 - t_2).$$

The symbol "U" is the overall heat transfer coefficient and is defined by the equation given.

The foregoing development was based on the assumption that the temperature of the fluid on either side of the wall remains constant; the hot fluid at temperature t_1 and the cold fluid at temperature t_2. Although this may be the case under some circumstances, there are many instances where either the temperature of the hot or cold fluid, or both, changes as the fluid moves along the wall. In these cases the above equation is written:

$$Q = UA\Delta t_m$$

where Δt_m is the mean temperature difference for the entire surface under consideration.

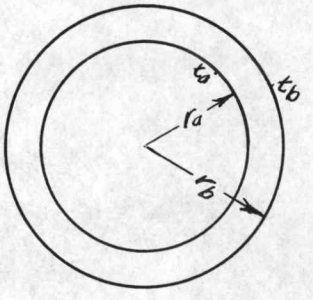
In the case of heat transfer through a cylindrical wall of unit length, such as the tube of a heat exchanger, the Fourier equation may be modified and solved as follows:

$$Q = -kA \frac{dt}{dL}$$

$$= -k(2\pi r) \frac{dt}{dr}$$

$$Q \int_{r_a}^{r_b} \frac{dr}{r} = -k(2\pi) \int_{t_a}^{t_b} dt$$

$$Q \ln \frac{r_b}{r_a} = 2\pi k(t_a - t_b).$$



The expressions for the effect of the surface films are:

$$Q = hA\Delta t$$

$$Q = h_1(2\pi r_a)(t_1 - t_a)$$

$$Q = h_2(2\pi r_b)(t_b - t_2).$$

By solving each equation for the temperature difference, adding, and solving the resulting equation for Q, the following is obtained:

$$Q = \frac{2\pi(t_1 - t_2)}{\frac{1}{h_1 r_a} + \frac{\ln r_b/r_a}{k} + \frac{1}{h_2 r_b}}$$

$$= \frac{2\pi r_b (t_1 - t_2)}{(\frac{r_b/r_a}{h_1} + \frac{r_b \ln r_b/r_a}{k} + \frac{1}{h_2})}.$$

For a unit length of cylinder the term $2\pi r_b$ represents the outside surface area of the cylinder. Using this, and defining the overall heat transfer coefficient by the following expression:

$$U = \frac{1}{\frac{r_b/r_a}{h_1} + \frac{r_b \ln r_b/r_a}{k} + \frac{1}{h_2}},$$

the expression for the quantity of heat transferred through a cylindrical wall per unit of time becomes:

$$Q = UA_b(t_1 - t_2).$$

As in the case previously considered, this development is based on the assumption that temperatures t_1 and t_2 remain constant. By reasoning similar to that previously used the foregoing equation is modified and becomes:

$$Q = UA_b\Delta t_m.$$

The equations defining the overall heat transfer coefficient, U, show that it is a function of the thermal conductivity of the metallic wall, k, and a function of the film coefficients, h_1 and h_2. The thermal conductivity coefficient, k, is a property of the specific metal used, and remains essentially constant over a wide range of temperature. Values of "k" for various metals and alloys have been determined experimentally and may be obtained from various references. The film coefficients, h_1 and h_2, are functions of certain properties of the flowing fluids (such as viscosity, density and specific heat) and are functions of the velocities of flow. They are also dependent on other factors such as tube diameter, whether the fluids are being heated or cooled, etc.

Through the correlation of experimental data by use of dimensional analysis, various empirical formulas have been developed for evaluating "h" and are available from literature pertaining to heat transfer.

Materials. Since various parts of the refrigerating unit will be in contact with air, water, steam, and the neutral lithium bromide salt solution, the prime requisite of the materials selected must be their ability to resist the corrosive tendencies of the fluids with which they will be in contact. Other properties of the materials under consideration which should be taken into account include: workability, strength and ductility, thermal conductivity, and cost.

The ability of a material to resist corrosion is dependent on many variables, and in general can only be evaluated by subjecting the material to actual service conditions. Since this was not possible, a survey of the published data regarding corrosion resistance characteristics of several "corrosion-resistant" metals and alloys was made. Conclusions drawn from this survey indicate that all parts of the unit which are in direct contact with the lithium bromide solutions should be made of monel, or if monel is not available, one of the high brasses such as Admiralty metal. "Stainless steel", often considered a cure-all for corrosion problems, was

found to be unsatisfactory for this application due to its tendency to pit when in contact with the salt solutions.

The heat transfer coefficient for monel is quite low as compared with that for any of the pure metals or the copper alloys. In this particular application, the effect of a change in the thermal conductivity coefficient on the value of U, the overall heat transfer coefficient, would be small; in most cases negligible.

Another consideration which should not be overlooked is the possibility of using corrosion inhibitors such as potassium chromate to reduce the corrosive action of the salt solution. Inhibitors are frequently used for reducing the corrosive action of brines employed in the ice-making industry.

Maintenance of the Vacuum. At design conditions the low-side pressure (in the evaporator and absorber) is to be maintained at 0.3 in. Hg and the high-side pressure (in the generator and condenser) at 2.6 in. Hg. The problem of maintaining these low pressures and of purging air and other non-condensible gases from the system is one of major importance.

Large commercial units of this type manufactured by Carrier Corporation employ a 2-stage purge system consisting of steam and water jets with an interstage

purge condenser. Apparently this method is practical on units of such a size as to warrant constant supervision. Due to the fact that the ejector may fail to start and would fail in operation if the pressure in the purge condenser rose above a critical value, this method is not considered satisfactory for the unit being designed since it is intended to be automatic in operation.

Mechanical purge systems such as centrifugal or reciprocating pumps were not considered practical due to high cost. An aspirator using a water jet was not feasible due to the large quantity of water which would be required (6, pp.5-8).

By eliminating all other possibilities it appeared that the only remaining solution was to construct the entire unit pressure tight, and to evacuate the system after assembly. It is possible that the unit would have to be hermetically sealed to guarantee freedom from air leakage. To facilitate the problem of design of the unit, it was therefore assumed that this method would be used; however, further investigation and experimentation would be required to be assured that this is the most practical approach.

Properties of the Flowing Fluid. In addition to those properties of the lithium bromide solutions already presented, there are other properties which must be known

in order to evaluate the overall heat transfer coefficients for those pieces of apparatus in which these solutions flow. These properties are the density, thermal conductivity, and viscosity of the solutions.

Table IV provides data from which the density of the solutions may be determined (13, 3:77). Since no information regarding the thermal conductivity of these solutions was available from the literature, it was assumed that the thermal conductivity of the solution was equal to 95% of that of water at the same temperature. The value of 95% represents an average value of thermal conductivities of other brines used in refrigeration applications.

Only one value of viscosity of an aqueous solution of lithium bromide was found in the references; however, more data were available for aqueous solutions of lithium chloride. Data on the viscosity of these solutions are given in Table V, and are shown graphically in Fig. 5 (13, 5:15). From Table V it can be seen that the viscosities of LiBr and LiCl solutions are within 0.1% at the one condition of temperature and concentration available for comparison. On this basis it was assumed that the viscosities of LiBr and LiCl solutions could be considered to be approximately equal under like conditions of temperature and concentration. Since for any given

TABLE IV

SPECIFIC GRAVITY OF AQUEOUS LiBr SOLUTIONS

(Referred to Water at 39.2 F)

Conc. % LiBr by Wt.	Temperature, F					
	32	68	104	140	176	212
1	1.00731	1.00547	0.99939	0.9904	0.9790	0.9656
2	1.01484	1.01280	1.00663	0.9976	0.9862	0.9729
4	1.03012	1.02769	1.02135	1.0123	1.0009	0.9877
6	1.04574	1.04293	1.03642	1.0273	1.0159	1.0029
8	1.06173	1.05854	1.05186	1.0427	1.0313	1.0184
10	1.07812	1.07455	1.06770	1.0585	1.0471	1.0344
12	1.09493	1.09099	1.08397	1.0747	1.0634	1.0508
14	1.11219	1.10789	1.10069	1.0914	1.0801	1.0676
16	1.12993	1.12528	1.11789	1.1086	1.0972	1.0849
18	1.14817	1.14318	1.13560	1.1262	1.1149	1.1027
20	1.16695	1.16162	1.15384	1.1444	1.1331	1.1210
22	1.18631	1.18063	1.17264	1.1632	1.1519	1.1399
24	1.20628	1.20024	1.19204	1.1826	1.1713	1.1593
26	1.22689	1.22047	1.21207	1.2026	1.1913	1.1794
28	1.24817	1.24135	1.23277	1.2232	1.2119	1.2001
30	1.27017	1.26292	1.25417	1.2445	1.2333	1.2215
35	1.32850	1.32040	1.31115	1.3012	1.2899	1.2782
40	1.39233	1.38360	1.37376	1.3635	1.3520	1.3404
45	1.46259	1.45354	1.44299	1.4323	1.4207	1.4092
50			1.52010	1.5089	1.4971	
55			1.60673	1.5948	1.5829	
60			1.70499	1.6923	1.6800	
65			1.81756	1.8038	1.7913	

TABLE V

VISCOSITY OF AQUEOUS LiBr AND LiCl SOLUTIONS

Concentration, F	Ratio of viscosity of solution to that of water at same temperature, R	
Formula wt. salt per 1,000 g H_2O	$R_{64.4\ F}$	$R_{77\ F}$
LiBr:		
0.1		1.015
LiCl:		
0.1	1.016	1.014
0.25	1.038	1.035
0.5	1.074	1.069
1	1.147	1.142
2	1.298	1.302
3	1.473	1.479
4	1.669	1.673
5	1.891	1.895
6	2.14	2.15
7	2.44	2.45
8	2.80	2.81
9	3.23	3.23
10	3.76	3.73
11	4.40	4.33
12	5.14	5.06
13	6.03	5.94
14	7.14	6.99
15	8.47	8.23
16	9.99	9.60
17	11.8	11.12

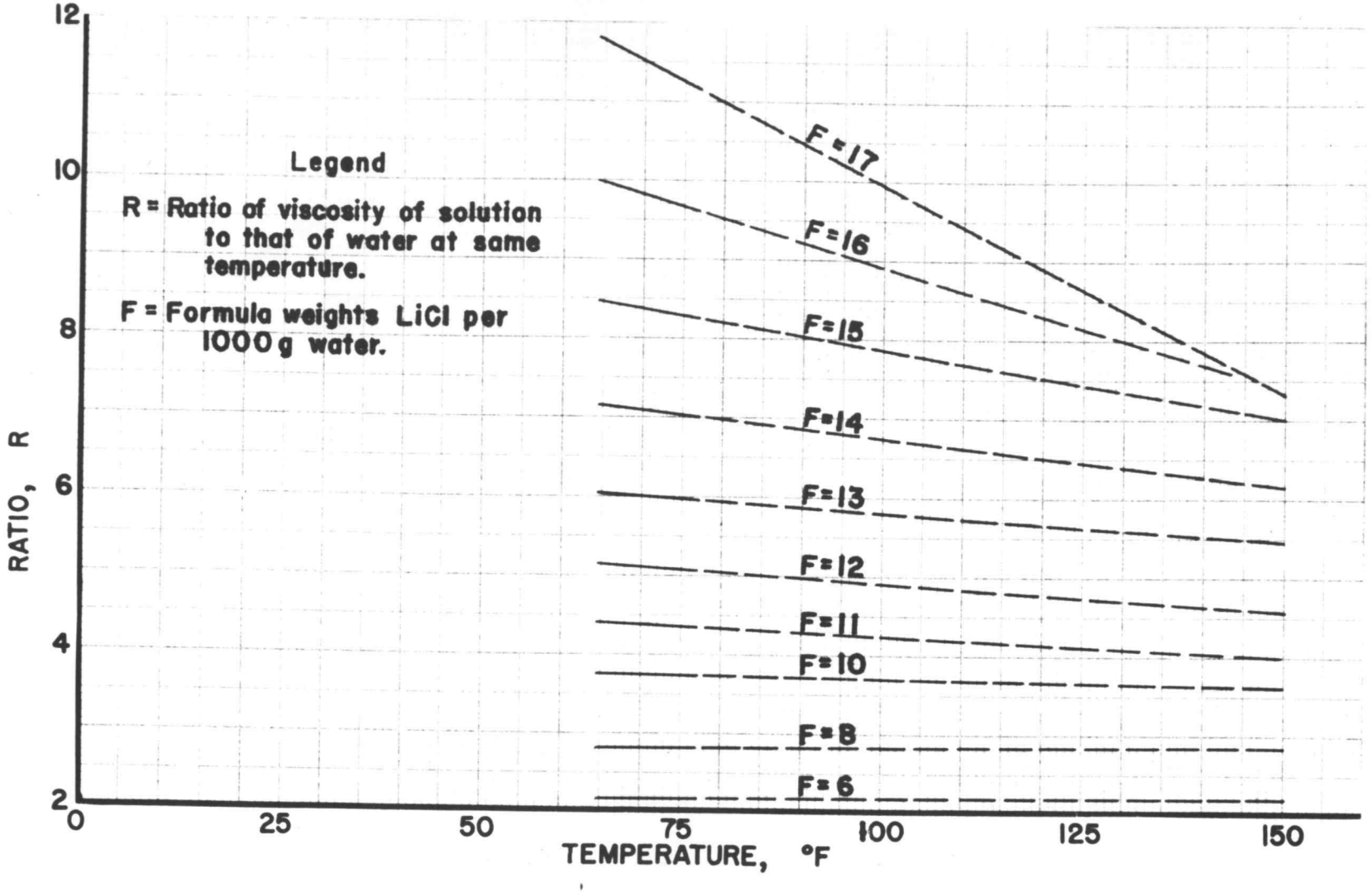


Fig. 5. Viscosity of aqueous LiCl solutions.

concentration the values of viscosity were given for only two temperatures, it was necessary to extrapolate the curves of Fig. 5 to determine a value of viscosity at a temperature beyond the range of the data. Obviously this is poor practice and an effort was made to secure a small amount of lithium bromide salt so that viscosity tests of sample solutions at design conditions of concentration and temperature could be made. Due to government restrictions on this material, however, the lithium bromide was not available so it was necessary to assume that information obtained from Fig. 5 was correct.

Properties of water and water vapor are available in several of the references and are not included in this work.

Design Procedure. For each piece of equipment in the unit the heat transfer surface area required was determined by use of the equation

$$Q = UA\Delta t_m.$$

In each instance the value of Q was known and that of Δt_m could be calculated. To get an approximate value of the surface area, A, an average value of U was assumed and A could then be computed.

By knowing the approximate value of the surface area, A, and the volume of fluid flowing, it was possible to determine the diameter, length, and number of tubes

required for the unit in question. With these values tentatively set, the theoretical value of U could be computed and compared to that originally assumed. When the computed value of U did not agree closely with that assumed, that part of the unit was redesigned using the computed value of U as an improved approximation. To illustrate this method, the computations required for the design of the condenser have been included in the section covering the design of that unit.

The above procedure was used in the design of each piece of heat transfer equipment in the unit except the generator, for reasons that will be discussed in the following section.

Design of the Generator-condenser. Since the specific volume of the superheated steam (the refrigerant) leaving the generator is large, a line of large diameter would be required to transmit this vapor from the generator to the condenser at a reasonable velocity. To eliminate this problem, the generator and condenser were combined in a single drum. This had no effect on the determination of heat transfer surface required for either the generator or the condenser but did eliminate the problem of connecting the two separate pieces of apparatus.

In the generator the heat transfer problem is one in which steam is condensing on one surface and fluid

is boiling on the other. Information regarding heat transfer to boiling liquids is quite meager and contradictory. It is generally agreed, however, that the heat transfer rate is dependent on the ability of the liquid to wet the metal surface, and that aqueous salt solutions generally have lower coefficients than water at the same conditions of pressure and temperature differential. No definite agreement has been reached on what, if any, effect the condition of the heat transfer surface (whether rough or smooth) has on the overall heat transfer coefficient, nor have empirical relationships been obtained which express the film transfer coefficients in terms of the properties of the fluids used (10, pp.294-310).

Experiments to determine the overall heat transfer coefficients from condensing steam to boiling liquids have been performed by several different investigators and results obtained have been compiled and published (10, pp.304-305). Based on these data a reasonable value for the overall heat transfer coefficient of the generator was set at 250 BTU/ft^2-hr-F. The required heat transfer surface was then calculated and found to be 3.67 sq.ft. Based on a maximum steam velocity through the tubes of about 15 ft/sec, the following requirements were established:

No. tubes per pass: 7

No. passes:	1
Tube size:	3/4 in.
Wall thickness:	No. 18 BWG
Length:	33 in.
Material:	Monel.

The refrigerating unit was designed so that the cooling water would flow in series, first through the absorber tubes and then through the condenser tubes, so that a maximum increase in cooling water temperature could be had, thus reducing the required flow of cooling water to a minimum. Cooling water was assumed to enter the absorber at 85 F and leave the condenser at 105 F. Since the specific heat of water throughout this range of temperatures is essentially constant at 1 BTU/lb F, the increase in temperature of the cooling water flowing through each of these units is proportional to the heat removed by it in that unit. On this basis the temperature of the cooling water entering the generator could be determined and was found to be 95.37 F.

The vapor being driven out of solution in the generator enters the condenser at a temperature of 192 F and a pressure of 2.6 in. Hg and is therefore superheated. Since the amount of heat to be extracted from the vapor to reduce it to a saturated state is small when compared to its heat of vaporization, the temperature of the vapor

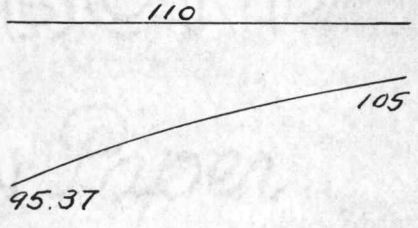
may be taken as the temperature of saturated steam at 2.6 in. Hg for the determination of the mean temperature difference. Any error involved in this approximation is small and tends to further reduce the value of the mean temperature difference.

Calculation of the logarithmic mean temperature difference was performed as follows:

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\ln \frac{\Delta t_1}{\Delta t_2}}$$

$$= \frac{14.63 - 5}{\ln \frac{14.63}{5}}$$

$$= 8.95 \text{ F}$$



The approximate heat transfer surface was then determined as follows:

$$Q = UA\Delta t_m$$

$$A = \frac{Q}{U\Delta t_m}$$

$$= \frac{(1068.8 \text{ BTU/lb})(35.9 \text{ lb/hr})}{(400 \text{ BTU/ft}^2\text{-hr-F})(8.95 \text{ F})}$$

$$= 10.62 \text{ sq. ft.}$$

Next, the weight of the cooling water flowing through the tubes was calculated, as shown below:

$$Q = WC\ t$$

$$W = \frac{Q}{C\Delta t}$$

$$= \frac{(1068.8 \text{ BTU/lb})(35.9 \text{ lb/hr})}{(3600 \text{ sec/hr})(1 \text{ BTU/lb F})(9.63 \text{ F})}$$

$$= 1.108 \text{ lb/sec.}$$

The desired velocity of the cooling water was assumed to be 3 ft/sec. The required cross-sectional area of the tubes was computed by using the continuity equation:

$$Q = AV$$

$$A = \frac{Q}{V}$$

$$= \frac{(1.108 \text{ lb/sec})(144 \text{ in}^2/\text{ft}^2)}{(62.0 \text{ lb/ft}^3)(3 \text{ ft/sec})}$$

$$= 0.856 \text{ sq. in.}$$

If 3/8" No. 20 BWG tubing is used, the number of tubes required to supply this area is 12. The outside surface area of this tubing is 0.0798 sq. ft/lin. ft; therefore, the required length per tube is:

$$L = \frac{10.62 \text{ ft}^2}{(0.0798 \text{ sq.ft/lin. ft})(12)}$$

$$= 11.1 \text{ ft.}$$

To use the same shell for both the generator and condenser would therefore require that the condenser be 4-pass.

The assumed value of U was checked by using empirical relationships for determining the film coefficients and computing a theoretical U value by use of the equation:

$$U = \frac{1}{\frac{r_b/r_a}{h_1} + \frac{r_b \ln r_b/r_a}{k} + \frac{1}{h_2}}.$$

The theoretical U value thus obtained was 520 BTU/ft^2-hr-F. Since the originally assumed U value of 400 was somewhat lower than the computed value, the assumed value of 400

was considered sufficiently accurate for design purposes.

From these computations the following requirements for the condenser were established:

No. tubes per pass:	12
No. passes:	4
Tube size:	3/8 in.
Wall thickness:	No. 20 BWG
Length per pass:	33 in.
Material:	Admiralty metal.

Sectional views of the proposed generator-condenser assembly are shown in Fig. 6. This unit is designed to operate in the following manner: Heat supplied to the generator tubes drives refrigerant vapor from the solution in this section of drum. This vapor rises and is condensed on the condenser tubes above the boiling solution. The condensate thus obtained drips from the condenser tubes into the tray below and flows from the tray into the line leading to the evaporator.

Design of the Heat Exchanger. An estimate of the heat transfer surface required for the heat exchanger was calculated after assuming an initial U value for this unit of 60 BTU/ft^2-hr-F. To obtain a velocity sufficiently high to assure turbulent flow of the solution through the tube, the unit could have only one ½" tube per pass. The required total length of this tube was found to be

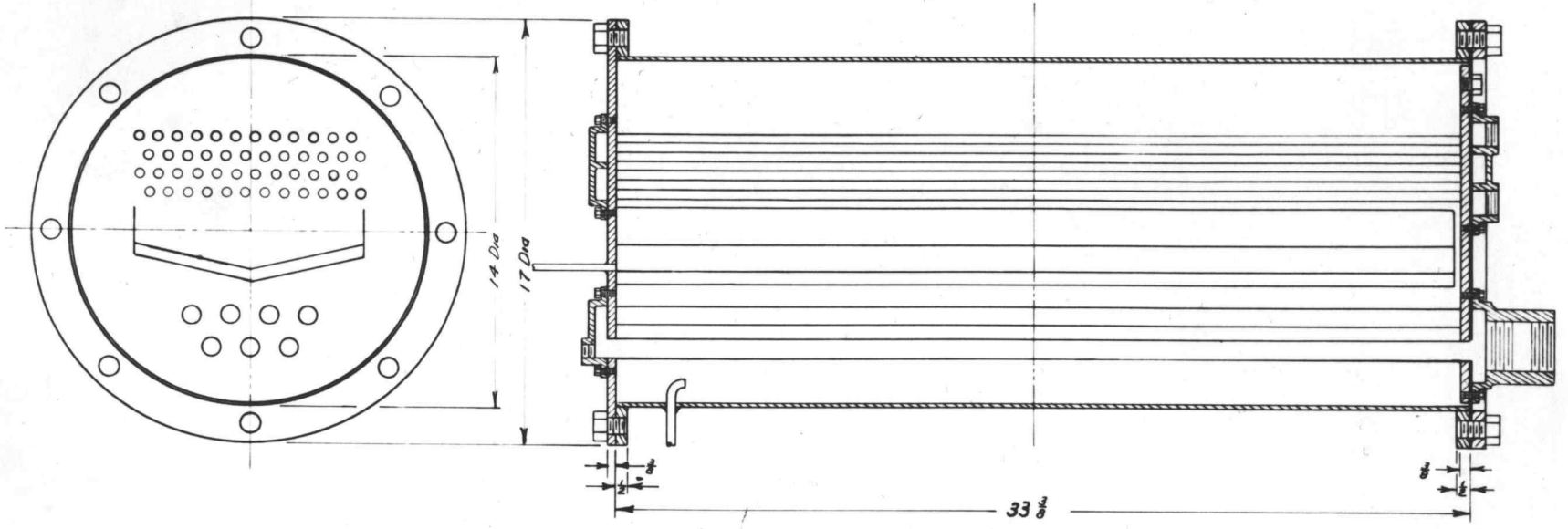


Fig. 6. Generator-condenser Assembly.

192 ft. By using 19 passes the length of tube required per pass was 10.1 ft. The minimum inside diameter of the shell of the unit was then estimated to be 4 in.

After the above values had been tentatively established the theoretical overall heat transfer coefficient was calculated and was found to be within 5% of the assumed value. The assumed value was therefore considered to be sufficiently accurate to serve the purposes of preliminary design, and the following requirements were established:

No. tubes per pass:	1
No. passes:	19
Tube size:	½ in.
Wall thickness:	No. 18 BWG
Length per pass:	10.1 ft.
Material:	Monel.

An assembly drawing and a sectional view showing the proposed construction of the heat exchanger is given in Fig. 7.

Design of the Absorber. Since heat is transferred from liquid to liquid in the absorber, as well as the heat exchanger, 60 BTU/ft^2-hr-F was again taken as an initial estimate of the overall heat transfer coefficient for this unit. By using this value an estimate of the required heat transfer surface area was calculated in the usual manner.

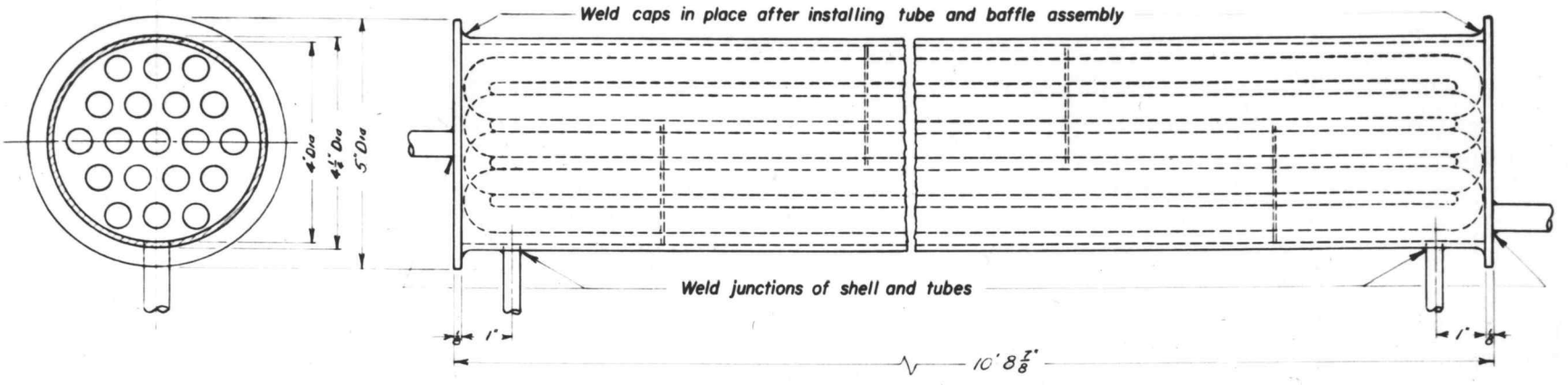


Fig. 7. Heat Exchanger.

The required tube size, length, and number of tubes to be used was then established.

After these dimensions were tentatively set, the theoretical overall heat transfer coefficient for this piece of apparatus was calculated. The value thus obtained was 72 BTU/ft^2-hr-F. Since this calculated value was somewhat greater than the original estimate, the estimate of 60 BTU/ft^2-hr-F was considered to be sufficiently accurate for use in preliminary design computations.

The proposed construction of the absorber is shown in Fig. 8 and is based on the following design data:

No. tubes per pass:	27
No. passes:	4
Tube size:	5/16 in.
Wall thickness:	No. 20 BWG
Length per pass:	10.58 ft.
Material:	Monel.

Design of the Evaporator. Design of the evaporator was based on the assumption that air would approach the cooling coil at 85 F and 50% relative humidity at an average velocity of 400 feet per minute and would leave the cooling coil at 70 F and 67% relative humidity. From this the required cross-sectional area of the duct was calculated. The evaporator was then designed to fit

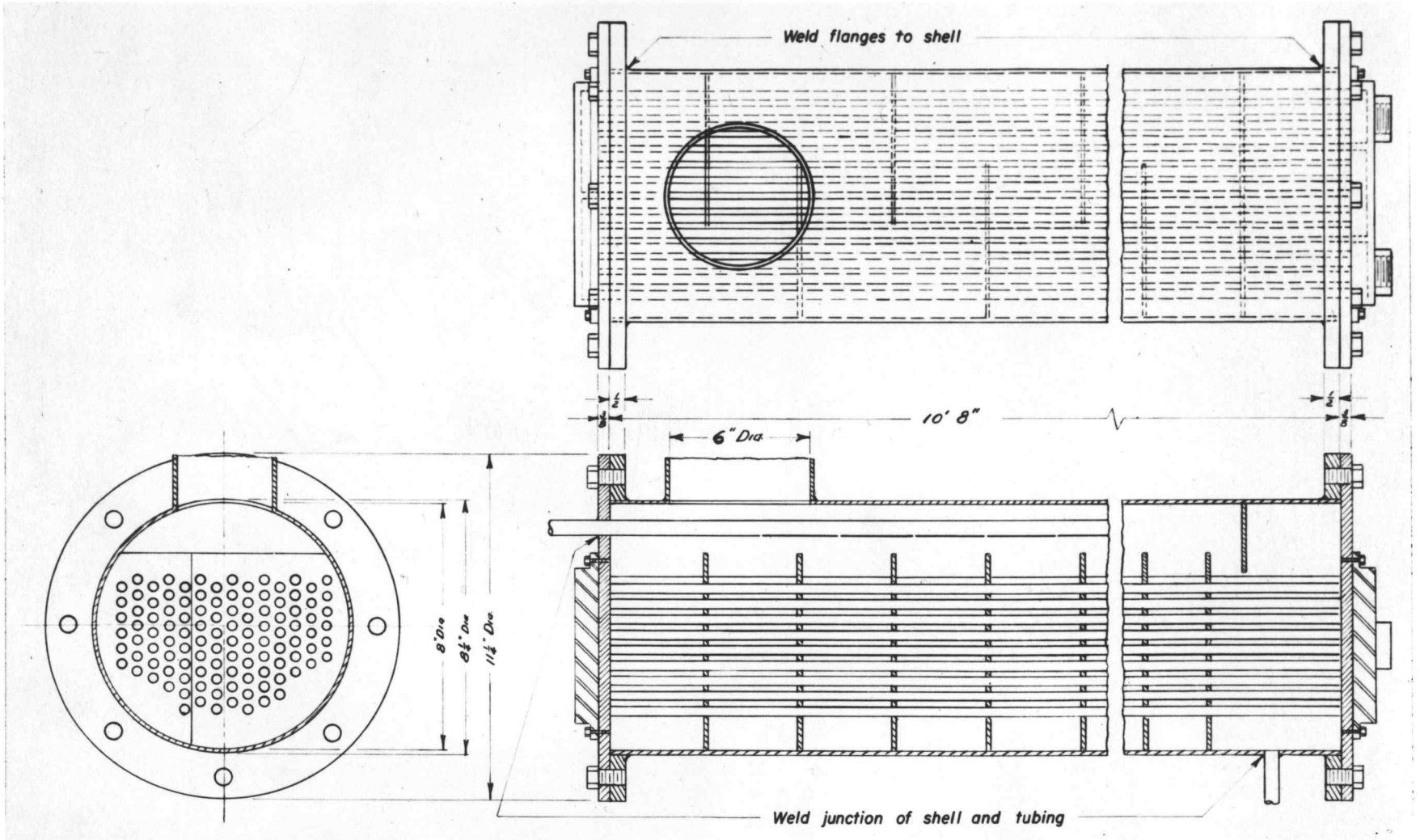


Fig. 8. Absorber.

within this duct area.

The evaporator tubes selected were those commercially available and the manufacturer's recommendations for determining the required surface area were followed (14, pp.6-7). Finned tubes were selected to increase the surface area per unit length of tube, since the overall heat transfer coefficient is very low for this piece of apparatus.

The specific volume of saturated steam at an absolute pressure of 0.3 in. Hg is 2036 cubic feet per pound. Therefore, even though the weight of the refrigerant flowing through the evaporator is small, the velocity of flow must be high in order to use evaporator tubes and headers of reasonable size. The maximum velocity of the vapor flowing through this unit was therefore assumed to be 100 ft/sec. A rough check of the pressure drop in this unit indicated that it would be of the order of 0.002 in. Hg and would therefore be negligible.

The design procedure used yielded the following:

No. tubes per pass:	64
No. passes:	1
Tube type:	Peerless Type 11-AT
Tube size:	3/4 in.
Length per pass:	9½ in.

Fig. 9 indicates the proposed construction of this

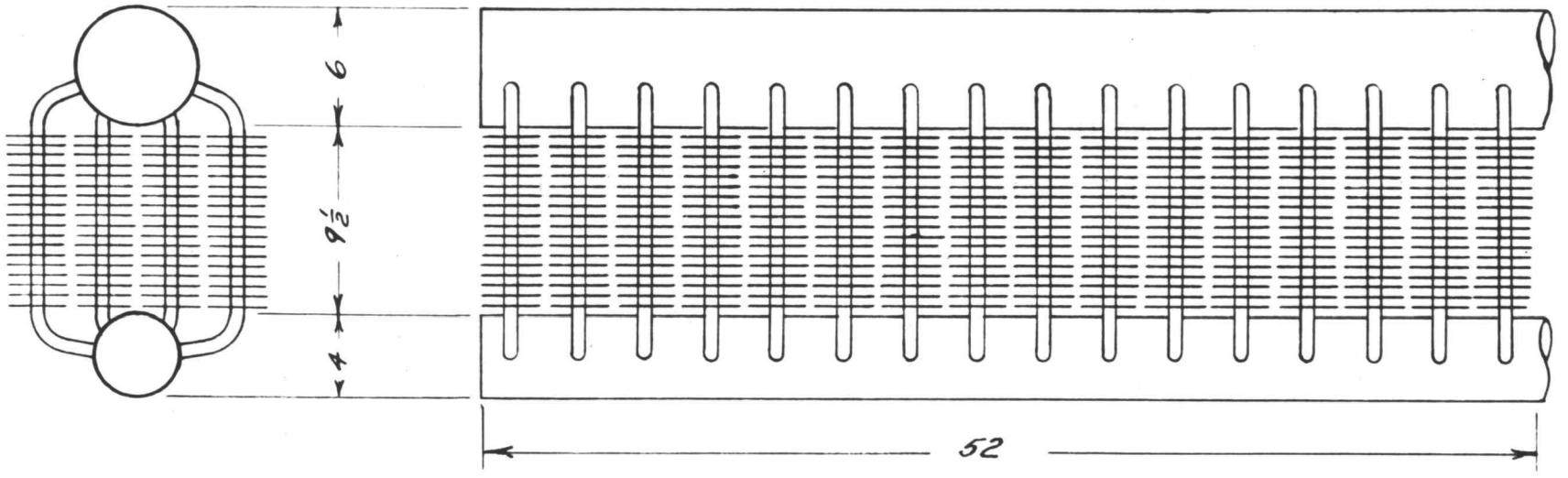


Fig. 9. Evaporator.

piece of apparatus.

Selection of the Solution Pump. The solution pump is required to deliver 0.866 gpm against an estimated total head of fifty feet. Since the flow is small, the pump horsepower required is only of the order of 0.02 hp. Assuming that the pump and motor assembly has an overall efficiency of only 20%, the gross horsepower of the pump motor required would be 0.1 hp.

Valves and Control Devices. Since it was desired to design this refrigerating unit to be automatic in operation, controls were selected which would fulfill this requirement. The controls proposed are self-acting, thus requiring no source of auxiliary power such as compressed air or electricity, and are therefore simple and relatively inexpensive.

A self-acting temperature controller consists of a pressure-thermometer bulb and capillary connected to the diaphragm or bellows motor of a valve. Liquid of a suitable vapor pressure is contained in the bulb and exerts a pressure which is transmitted through the capillary to the valve motor. The force exerted on the diaphragm or bellows due to this pressure is opposed by the force exerted by a compression spring. The point of balance of these forces determines the valve position.

The locations of the control devices are shown in

Fig. 10. Their functions and effects on the operation of the unit are as follows:

Control A--Self-acting temperature controller. The valve in the pump by-pass line is positioned according to the temperature of the air leaving the evaporator. Since this control operates at temperatures in the neighborhood of 70 F, a dual fill should be used to eliminate the lag in response due to the cross-ambient effect. The function of this control is to open the pump by-pass valve when the cooled air temperature falls, thus reducing the flow of solution to the generator.

Control B--Self-acting temperature controller. The valve in the steam supply line is positioned in accordance with the temperature of the strong solution leaving the generator, and should be adjusted to maintain this temperature constant at 192 F.

Control C--Self-acting temperature controller. The valve in the condenser cooling water line is positioned in accordance with the temperature of the condensate, thus indirectly controlling the pressure within the generator-condenser unit. Controls B and C together maintain the concentration of the solution leaving the generator at 60%.

Valve D. This valve is a pressure-reducing valve adjusted to maintain a constant evaporator pressure of

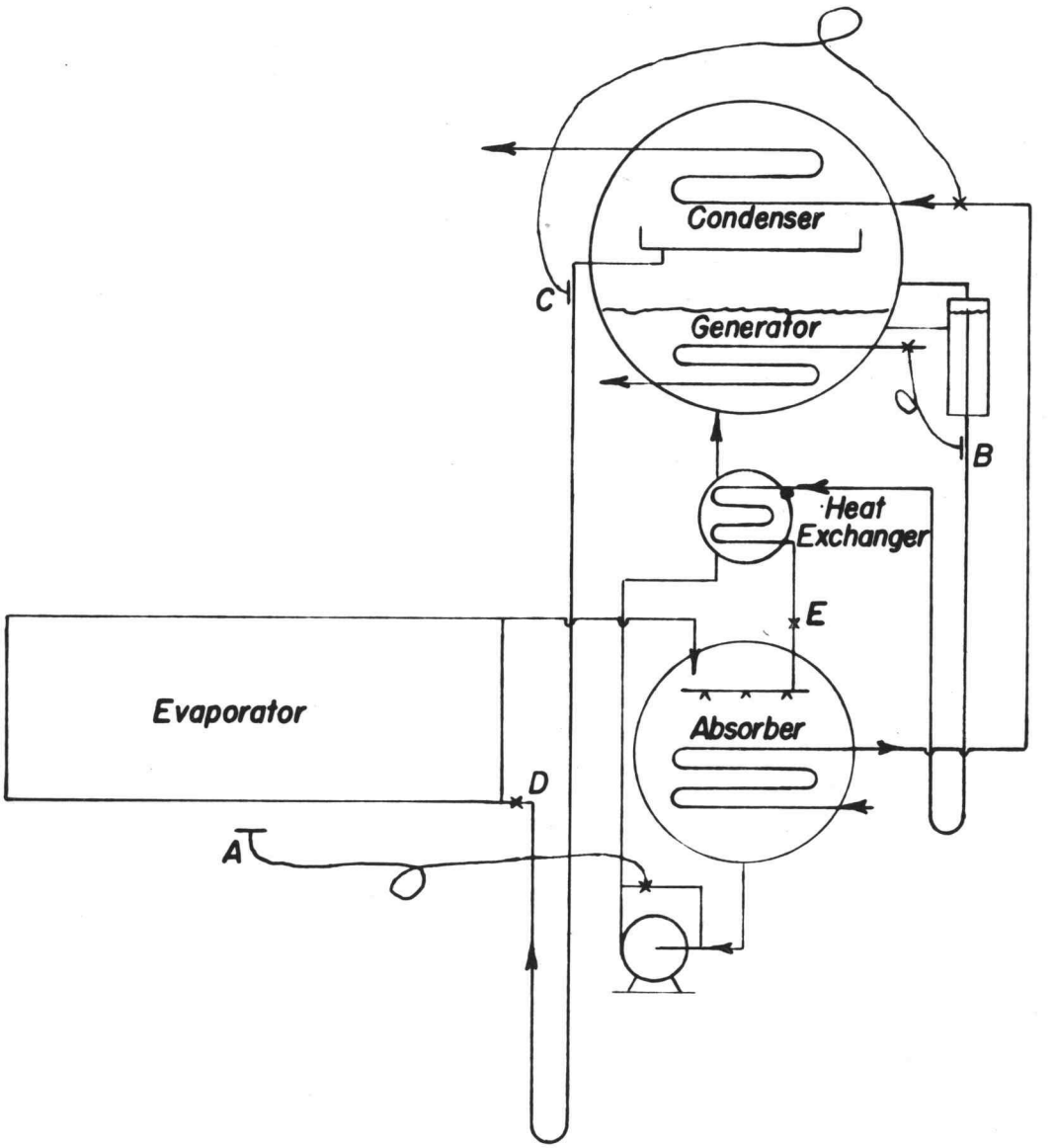


Fig. 10. Equipment Arrangement.

0.3 in. Hg.

Valve E. This valve is a pressure-reducing valve adjusted to maintain a constant pressure of 0.3 in. Hg in the absorber.

The level of the solution in the generator is maintained by means of the device shown in Fig. 10. A loop was provided in the line carrying the 60% solution from the generator to the absorber. This loop acts as a seal against the difference in pressure between these units and must extend a minimum vertical distance of two feet below the heat exchanger inlet. A similar loop was incorporated in the line returning the condensate to the evaporator to provide a seal against the difference in pressure between the generator and the evaporator. This loop must have a vertical rise of not less than three feet.

ADVANCE BOND

CONCLUSIONS

The theoretical performance ratio for the unit designed is 82.4%. The actual performance ratio for the 150-ton units manufactured by Carrier Corporation is reported to be over 70% throughout loads ranging from 30% to 100% of rated capacity (3, p.219). These ratios compare favorably with ratios ranging from 40% to 45% for the ammonia absorption system and about 50% for the Williams Oil-O-Matic absorption unit (1, p.54). It is evident, therefore, that of the absorption refrigeration cycles thus far developed, the one using water as the refrigerant and an aqueous solution of lithium bromide as the absorbent is most efficient.

Whether or not the refrigerating unit which has been designed would be practical for domestic air conditioning service is dependent on such factors as the availability of cooling water and the relative cost of steam and electricity. Based on operating cost data obtained under actual service conditions, Carrier Corporation has found that the break-even point in operating cost between their absorption machine and an electrically driven compression machine occurs where the cost of steam in dollars per thousand pounds is equal to fifty times the cost of electricity in dollars per kilowatt hour (2, p.608).

The unit designed should be quiet in operation and practically free from vibration since there are no moving parts other than the small circulation pump and motor.

The materials selected for the unit were chosen for their ability to resist the corrosive action of the fluids with which they would be in contact. A series of laboratory tests should be made to determine whether or not cheaper and more readily available materials could be used by adding corrosion inhibitors to the solutions.

Not all of the physical properties of aqueous lithium bromide solutions are available from the literature. Laboratory tests should be performed to determine these properties under conditions of temperature, pressure, and concentration encountered in the unit.

The maintenance of the vacuum and removal of non-condensible gases from the unit impose a problem that may be solved only by tests of an actual unit.

The purpose of this thesis--to design an absorption refrigeration unit for domestic air conditioning service--has been accomplished. It is evident, however, that a considerable amount of developing and testing remains to be done.

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