CALIFORNIA STATE UNIVERSITY, NORTHRIDGE

THE DESIGN OF AN EVAPORATOR FOR THE
SOLAR ENERGY POWERED ABSORPTION
REFRIGERATION SYSTEM

A project report submitted in partial satisfaction of the
requirements for the degree of Master of Science in
Engineering
by
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ABSTRACT

THE DESIGN OF AN EVAPORATOR

FOR

THE SOLAR ENERGY POWERED ABSORPTION REFRIGERATION SYSTEM

by

Lieu Van Pham

Master of Science in Engineering

The purpose of this report was to design one of the major components of a newly developed Solar Energy Powered Absorption Refrigeration System which was intended for use in air conditioning systems for residential and commercial projects. This component is the evaporator section of the system.

The basic refrigerative system uses sodium thiocyanate (NaSCN) as the absorbent medium, and ammonia (NH₃) as the refrigerant. The heat required to drive the ammonia out of solution is supplied by the system solar collectors delivering 215° F. water.

In the design and testing of the evaporator, detailed investigation and experiments were made not only on necessary steps required for the design of the evaporator, but also on the related components of the apparatus which have certain influence on the heat transfer process and performance of the evaporator.
In addition to the fundamental steps involved in the design, such as: cooling load analysis, material selections, evaporator housing and heat transfer element design, air duct and piping system design, etc., efforts were also made on investigating various methods for control of the system refrigerant liquid and vapor flows, pressure regulating, and refrigerant distribution techniques required for proper function and performance of the evaporator in connection with the entire absorption system as a whole.
Chapter 1

INTRODUCTION

ABSORPTION REFRIGERATION SYSTEM

The fundamentals of absorption refrigeration were formulated about 1777, but it was 1850 before a successful absorption machine was developed. In 1850 a United States patent was issued to Ferdinand Carre for the first commercial absorption refrigeration machine in this country.\(^1\) By 1890 the absorption machine had been developed for large refrigeration plants and from 1890 to about 1900 many large units were installed, especially in the petroleum industry and in plants manufacturing bulk chemicals. Other large refrigeration plants used steam jet units and huge, slow-speed reciprocating compressors. By 1911, engineers were no longer afraid to operate reciprocating compressors above 50 rpm, and many compressors were operating in the range of 100 to 300 rpm. At this point the development and refinement of absorption refrigeration machines was almost at a standstill as most efforts were concentrated on improving the compression refrigeration systems.\(^2\)

However, as the demand increased for refrigeration systems of larger capacity, electric power costs became increasingly important as an economic factor. It was only logical that attention was given to substitutes for electric power to operate large refrigeration equipment in areas where the cost of electricity was relatively high.
Steam jet units and steam turbine driven centrifugal compressors were heat-motivated refrigeration systems but these were available for only large loads. Therefore, in searching for methods to produce refrigeration with by-product steam or natural gas, interest in the absorption refrigeration cycle was revived. In about 1936 considerable research and development work was begun on the low-pressure absorption refrigeration equipment used for air conditioning.

During the past several years and especially recently, because of numerous economic and political developments in the United States as well as in many other areas of the world, there has been a large amount of interest generated in the field of solar energy. Because of the diminishing supply of fossil fuels on this planet, researchers have been searching for other forms of energy that can be developed and made useful to man.

Because of its near infinite supply, the energy from the sun is especially attractive. With the use of solar collectors, it is hoped that energy will be gathered to heat, cool, and power machinery. It is also, therefore, one of the main objectives of this study to experiment with the application of solar energy in conjunction with the absorption system.

FUNDAMENTAL ABSORPTION CYCLE

Basically, the absorption refrigeration system is not too different in operation from the more familiar mechanical compression system. Both systems accept heat to evaporate a refrigerant at low pressure in the evaporator, and thereby creating a cooling effect. Both also condense the vaporous refrigerant at a higher pressure and temperature.
in the condenser, in order that the refrigerant can be re-used in the cycle.

In mechanical compression systems the vapor, formed when the liquid refrigerant absorbs heat to provide the refrigerant effect, is drawn to a lower pressure area created by the mechanical movement of the piston. In an absorption machine this vapor is also removed to lower pressure area. However, the lower pressure area in the absorption machine is created by controlling the temperature and concentration of a solution (sodium thiocyanate for this project) that absorbs the vapor.

In compression system the refrigerant vapor is mechanically compressed and moved from the low-pressure to high-pressure side of the system. In an absorption system the vapor is first condensed into a solution. This solution is then pumped to a higher pressure area (Figure 1) and heat applied. Heat causes the solution to boil, driving off the refrigerant vapor at the higher pressure.

It is therefore evident that exactly the same function—that of taking low pressure refrigerant vapor from the evaporator and delivering high pressure refrigerant vapor to the condenser—has been performed in both compression and absorption cycles. The only difference here has been in the method of transporting the vapor from the low to the high-pressure side (see Figure 2).

SYSTEM OPERATION

As can be seen from Figure 3, refrigerant vapor \(\text{NH}_3\) is produced in the generator by boiling the absorbent and refrigerant \(\text{NaSCN+NH}_3\) solution. The heat source required for boiling the solution is
supplied by hot water from the solar heat collecting apparatus. The refrigerant vapor is then condensed to liquid in the condenser. Forced convection is used in the prototype model to transfer heat from the NH₃ to the surrounding air producing the phase change. Liquid ammonia is passed through the evaporator where heat from the incoming air is transferred to the evaporating NH₃.

The ammonia vapor is passed through the absorber where the vapor is absorbed into solution with sodium thiocyanate. The refrigerant vapor produced in the evaporator flows to the absorber because of the lower pressure in this area. This low pressure exists because the concentrated absorbent solution exerts a strong attractive force on the molecules of refrigerant vapor. The molecules of refrigerant vapor condense into the liquid as they come in contact with the molecules of the absorbent solution. In the absorber, two processes release heat: the heat of condensation of NH₃ vapor, and the heat of dilution as the vapor goes into solution with the absorbent. In order to remove this heat and maintain a constant temperature in the absorber, the absorbent solution falls over a cooling coil after being sprayed into the absorber. Cooling is provided by a secondary circuit connected to the absorber and the condenser of the system. This secondary refrigerant circuit removes the heat from the absorber. If this heat were not removed, the temperature and pressure in the absorber would rise, and flow from the evaporator would stop.

The solution of thiocyanate and ammonia is then pumped through a heat exchanger where heat is transferred to it from the hot NaSCN returning from the generator to the absorber. When the solution
reaches the generator the cycle is repeated.

The absorption cycle shown in Figures 1 and 3 illustrates the relationship between the main components of the system:

1. Generator
2. Condenser
3. Evaporator
4. Absorber
5. Solar Collector
6. Heat Exchanger
7. Solution Circulating Pump

Because the main purpose of this report is to carry out the detailed design of the evaporator in the absorption system, other major components listed above, although mentioned and shown in various parts of this study, will not be investigated in detail in the following sections of the report.
Chapter 2

PROPERTIES OF THE SELECTED SYSTEM REFRIGERANT

Refrigerants are the vital working fluids in refrigeration systems. They transfer heat from one place to another. Heat is removed from the system through the condenser and is added by being absorbed into the system at the evaporator.

Ammonia is selected to be used as the system refrigerant for this project. It has been known as refrigerant 717. It is an inorganic compound and has a chemical formula: \( \text{NH}_3 \). It is also a powerful solvent which readily removes from the inside of pipes, valves, fittings, etc., any dirt, scale, sand, or moisture which has remained in them during installation. These foreign substances are soon swept along with the suction gas to the absorber and are a distinct menace to various components of the system. The physical properties of ammonia are listed as follows:

- Molecular weight \( \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldOTS
Viscosity of ammonia liquid at saturation pressure:

<table>
<thead>
<tr>
<th>Temperatures °F</th>
<th>Viscosity</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.250</td>
</tr>
<tr>
<td>20</td>
<td>0.240</td>
</tr>
<tr>
<td>40</td>
<td>0.230</td>
</tr>
<tr>
<td>60</td>
<td>0.220</td>
</tr>
<tr>
<td>80</td>
<td>0.210</td>
</tr>
<tr>
<td>100</td>
<td>0.200</td>
</tr>
</tbody>
</table>

Viscosity of ammonia vapor at one atm. pressure:

<table>
<thead>
<tr>
<th>Temperatures °F</th>
<th>Viscosity in Centipoise</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.0085</td>
</tr>
<tr>
<td>20</td>
<td>0.0088</td>
</tr>
<tr>
<td>40</td>
<td>0.0093</td>
</tr>
<tr>
<td>60</td>
<td>0.0097</td>
</tr>
<tr>
<td>80</td>
<td>0.0101</td>
</tr>
<tr>
<td>100</td>
<td>0.0105</td>
</tr>
<tr>
<td>120</td>
<td>0.0105</td>
</tr>
<tr>
<td>140</td>
<td>0.0116</td>
</tr>
<tr>
<td>150</td>
<td>0.0116</td>
</tr>
</tbody>
</table>

Thermal conductivities of ammonia at one atm.:

<table>
<thead>
<tr>
<th>Temperatures °F</th>
<th>K (Btu/hr./sq. ft./°F.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.290 (liquid)</td>
</tr>
<tr>
<td>32</td>
<td>0.013 (vapor)</td>
</tr>
<tr>
<td>122</td>
<td>0.0157</td>
</tr>
<tr>
<td>302</td>
<td>0.0235</td>
</tr>
</tbody>
</table>
Heat capacity of liquid ammonia: 6

<table>
<thead>
<tr>
<th>Temperatures °F.</th>
<th>Btu/lb./°F.</th>
</tr>
</thead>
<tbody>
<tr>
<td>-30</td>
<td>1.05</td>
</tr>
<tr>
<td>0</td>
<td>1.10</td>
</tr>
<tr>
<td>60</td>
<td>1.10</td>
</tr>
<tr>
<td>90</td>
<td>1.10</td>
</tr>
</tbody>
</table>

Heat capacity of ammonia vapor: 7

<table>
<thead>
<tr>
<th>Temperatures °F.</th>
<th>Btu/lb./°F.</th>
</tr>
</thead>
<tbody>
<tr>
<td>32</td>
<td>0.50</td>
</tr>
<tr>
<td>59</td>
<td>----</td>
</tr>
<tr>
<td>212</td>
<td>0.53</td>
</tr>
</tbody>
</table>

Latent heat of vaporization versus boiling point:

(Btu/mol.) Trouton Constant

10,036 23.25

Electrical properties of liquid and vapor ammonia:

<table>
<thead>
<tr>
<th>Temperatures °F.</th>
<th>Dielectric Constant</th>
</tr>
</thead>
<tbody>
<tr>
<td>69</td>
<td>Liquid 15.5</td>
</tr>
<tr>
<td>32</td>
<td>Vapor 1.0072 atm.</td>
</tr>
</tbody>
</table>

Ammonia refrigerant performance per ton (based on 5° F. evaporation and 86° F. condensation): 8

Evaporation pressure . . . . . . . . . . . . . 19.60 psig
Condensing pressure . . . . . . . . . . . . . . 154.50 psig
Compression ratio . . . . . . . . . . . . . . 4.94
Net refrigerating effect ............... 474.40 Btu/lb.
Refrigerant circulated ............... 0.422 lb./min.
Liquid circulated ..................... 19.6 cu. in./min.
Specific volume of suction gas ........ 8.15 cu. ft./lb.
Coefficient of performance ............ 4.76

Ammonia refrigerant performance per ton at various condensing temperatures:

<table>
<thead>
<tr>
<th>Condensing temperatures (F.)</th>
<th>Evaporator temperatures (F.)</th>
<th>Suction gas temperatures (F.)</th>
<th>Evaporator pressures:</th>
<th>Condensing pressures:</th>
<th>Refrigerant circulated (lb./min.):</th>
<th>Specific volume of vapor (cu. ft./lb.):</th>
<th>Net refrigerant effect (Btu/lb.):</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>-76</td>
<td>-76</td>
<td>23.4</td>
<td>19.6</td>
<td>0.37</td>
<td>75.30</td>
<td>533.5</td>
</tr>
<tr>
<td>68</td>
<td>-40</td>
<td>-40</td>
<td>8.7</td>
<td>100.6</td>
<td>0.42</td>
<td>24.86</td>
<td>479.3</td>
</tr>
<tr>
<td>100</td>
<td>40</td>
<td>40</td>
<td>9.0</td>
<td>197.2</td>
<td>0.44</td>
<td>11.50</td>
<td>453.3</td>
</tr>
</tbody>
</table>

EFFECT OF AMMONIA ON CONSTRUCTION MATERIALS

Ammonia (NH₃) can be used satisfactorily under normal conditions with most of the common metals, such as: steel, cast iron, tin-lead, and aluminum. Under more severe conditions, the various metals will affect such properties as hydrolysis and thermal decomposition in differing degrees. The tendency of metals to promote thermal decomposition of the compounds is the following general order:

(least decomposition) inconel < 18-8 stainless steel < nickel < copper < 1340 steel < aluminum < bronze < brass < zinc < silver (most decomposition).
This order is only approximate and exceptions may be found for individual compounds or for special conditions of use. The effect of metals on hydrolysis would probably be similar.

In particular, copper should not be used with ammonia.

DETECTION OF AMMONIA LEAK

Ammonia can be detected by burning a sulfur candle in the vicinity of the suspected leak or by bringing a solution of hydrochloric acid near the object. If ammonia vapor is present, a white cloud or smoke of ammonium sulfite or ammonium chloride will be formed. Ammonia can also be detected with any indicating paper which changes color in the presence of a base.

The presence of leaks can also be determined by pressurizing or evacuating and observing the change in pressure of vacuum over a period of time.

SAFETY

Ammonia is a poisonous gas if inhaled in sufficient quantities and in lesser quantities is irritating to the eyes, nose and throat. In certain proportions ammonia and air form an explosive mixture.

It is classified among group 2 of the Underwriters' Laboratories Comparative Hazard to Life of Gases and Vapors. This group represents gases and vapors which in concentrations of about one-half to one percent for duration of exposure of about one-half hour are lethal or produce serious injury.
Chapter 3

THE EVAPORATOR

FUNCTION

The evaporator section in an absorption refrigeration system is the place where liquid refrigerant evaporates in order to produce cooling effect. It is one of the most important parts of the system in which the desired process of heat transfer takes place. Normally, evaporators are designed in accordance with predetermined parameters upon which system applications are based.

SYSTEM DESIGN CONDITIONS

Because the intended application of the Solar Energy Powered Absorption Refrigeration System is for use in air conditioning systems in residential and commercial structures, the evaporator of this system is designed to meet the following conditions:

1. Conditioned space to have temperature of: 75° F. dry bulb, 62.5° F. wet bulb.

2. Location of the project is Northridge, California, with a summer outdoor weather design condition of: 98° F. dry bulb, 72° F. wet bulb, and the outdoor daily range of 38° F.

3. System to have refrigerant-to-air heat transfer operation.

4. Prototype model to have a nominal cooling capacity of 2370 Btu/hr.

5. \( \text{NH}_3 \) is the system refrigerant.

6. NaSCN is the system absorbent.
7. Aluminum is proposed as the coil tube element of the evaporator.

It is worth mentioning that, though the following steps are taken to complete the design of an efficient evaporator to meet the conditions and criteria shown above, the availability of various existing components in the laboratory dictates the use of certain preselected equipment of the system in the prototype model for simulation tests. Other auxiliary equipment such as: circulating air fan, electric heater, thermometers, etc., will also be used to create necessary conditions required to test the evaporator and measure its performance efficiency.

The evaporator unit used in the prototype mockup system test was donated by Reynolds Metal Company. It is a simple aluminum fin tube construction using stainless steel fittings for all tubing transitions. A 7-micron filter is also incorporated into the one-eighth inch input line to the evaporator for protection against suspended particles in the system. A manual throttle valve is installed in the one-eighth inch liquid line to control the \( \text{NH}_3 \) entering the evaporator.

**PROTOTYPE SYSTEM COOLING LOAD CALCULATIONS**

The prototype unit is intended to handle one twelfth load capacity of an actual model which has a load of 8,333 watts. The actual unit cooling load is, therefore, calculated as follows:

Cooling load: 8,333 watts x (3.41 Btu/watt) = 28,415.53 Btu/hr.

or: 28,415.54 / 12,000 Btu/ton = 2.368 tons

Therefore, the prototype model unit has a cooling load of:

\[ \frac{28,415.53}{12} = 2368 \text{ Btu/hr}. \]
or: 0.197 ton of refrigeration

In air conditioning applications for commercial and residential systems, normally a delta T (delta T = the difference in °F. between the dry bulb temperature of the room to be conditioned, and the dry bulb temperature of air leaving the evaporator or cooling coil) of 20° F. is maintained. Based on this criterion, the quantity of air required to deliver a cooling load of 2,368 Btu/hr. to the space can be calculated as follows:

Air quantity: \(\frac{2,368 \text{ Btu/hr.}}{1.08 \times 20° \text{ F.}} = 109.63 \text{ CFM}\)
in which:

CFM = cubic feet per minute

1.08 = Btu/cu. ft. - °F. - hr. \(C_p, \text{ dry air at 55° F.} = 0.01851\)

Btu/cu. ft. - °F.

From the above calculations, it is seen that in normal operation the prototype model should be able to produce a cooling load of 2,368 Btu per hour. A circulating (supply) fan with a capacity of approximately 110 CFM is required to transfer the above cooling to the conditioned space. In actual application, such cooling and air loads are adequate for maintaining a normal level of comfort for a room of approximately 110 square feet in area.

Also, because the prototype model is intended to simulate air conditioning systems (to maintain: 75° F. db, 62.5° F. wb, and 50% relative humidity in room) the dry bulb temperature of air leaving the evaporator coil must be approximately:

Leaving air temperature: 75° F. - 20° F. = 55° F.
Since there is a heat loss (assumed to be 2° F.) due to convection in the ductwork system conveying evaporator air from the system evaporator to the conditioned space, the actual dry bulb temperature of air leaving the cooling coil should be:

Temperature of air leaving coil: 55° F. - 2° F. = 53° F.

See Figure 4.

In an actual air conditioning system, the amount of air entering a cooling coil to be cooled is a mixture of system return air and fresh outside air. The volumes of these two quantities are normally determined by local code standards. However, in typical systems for commercial and residential projects the satisfactory proportions of the two volumes are usually recommended as follows:

Fresh (outside) air: 25%

System recirculated air: 75%

Based on the above ratios of both quantities, the temperatures of the mixed air entering the evaporator for the prototype unit could be calculated in the following manner:

Total air quantity entering evaporator: 110 CFM

Amount of outside air: 110 CFM x 0.25 = 27.5 CFM

Amount of return air: 110 CFM x 0.75 = 82.5 CFM

Since the outdoor design temperature of air in Northridge, California is 98° F. db, 72° F. wb, and that of the system return air is 75° F. (approximately), the temperature of the air mixture is:

\[
\frac{(27.5 \times 98° \text{ F.}) + (82.5 \times 75° \text{ F.})}{110} = 80.75° \text{ F.}
\]

Therefore, it could be concluded that if the project Solar Energy
Powered Absorption Refrigeration System is used in a commercial air conditioning system, then the load characteristics of the system evaporator could be summarized and presented as follows:

- Unit cooling load: 4,258 Btu/hr.
- Unit air capacity: 110 CFM.
- Temperature of air entering evaporator: 75°F.
- Temperature of air leaving evaporator: 53.0°F.

**EVAPORATOR COOLING COIL CONSTRUCTION AND ARRANGEMENT**

Basically, there are two types of cooling coils which are used in air conditioning systems, those consisting of bare tubes or pipe, and those having extended or finned surfaces. The former are little used for the applications covered by this report, but are sometimes used where conditions cause frost accumulation, and for cooling within sprayed coil humidifiers.

The design and arrangement of a coil, constructed with extended type surface on the air side, involves consideration of such items as:

1. Materials
2. The type, thickness, height, and spacing of the fins.
3. The ratio of the extended surface area to that of the tube.
4. The tube nesting center dimensions.
5. The use of a staggered or an in-line tube arrangement.
6. Provisions to increase air turbulence with use of discontinuous, louvered, corrugated, or other designs of configurated rather than flat fins. Staggered tubes increase the total heat transfer, as opposed to the in-line arrangement, and configurated fins may be more effective than flat. The design and surface arrangement have a great effect on the air-film heat transfer.
resistance and associated air-side pressure drop.

Because of its advantages over other types of coils in air conditioning systems applications, the finned tube or extended surface coil is selected for this project.

In fin or extended surface cooling coils, the external surface of the tubes is known as primary, and the fin surface is called secondary. The primary surface consists generally of rows of round tubes or pipes which may be staggered, or placed in line with respect to the air flow. The inside surface of the tubes is usually smooth and plain, but some designs use various forms of internal fins or turbulence promoters (turbolators) either fabricated or extruded, to provide additional inside surface area for enhancing performance. Numerous types of fin arrangement are used, the most common of which are smooth spiral, crimped spiral, flat plate, and configurated plate, all are shown in Figure 5. While the spiral fin surrounds each tube individually in all cases, the plate types may be continuous (including several rows of tubes), or they may be round or square, with individual fins for each tube. The most important factor in the performance of extended-surface coils is the bond between the fin and the tube. An intimate contact between the tube and the fin must be maintained permanently to assure unimpeded heat transfer from fin to tube.

In some coils, fins are wound on the tubes under pressure, in order to upset the metal slightly at the fin root, and are then given a coating of solder while the fin and tube are still revolving, for the purpose of assuring a uniform coating of solder. In other types, the spiral fin may be knurled into a shallow groove on the exterior of the tube. The tube may be expanded after the fins are assembled, or
FIG. 5 - TYPICAL TYPE OF FIN-COIL ARRANGEMENT

CRIMPED SPIRAL FINS
CONTINUOUS PLATE FINS

SMOOTH SPIRAL FINS
PLATE FINS ON INDIVIDUAL TUBES
the tube-hole flanges of a flat or configurated fin may be made to override those in the preceding fin and so compress them upon the tube. There are also types of construction where the fin is formed out of material of the tube itself.

Coils for volatile refrigerants present more complex problems of cooling fluid distribution than water or brine coils. It is desirable that the coil be effectively and uniformly cooled throughout. Direct-expansion coils are used on two types of refrigeration systems, namely: flooded systems and dry-expansion systems (Figure 6).

The flooded system is used mainly where evaporator coil performance for low temperature applications at a small temperature difference between the air and refrigerant is advantageous. However, a relatively large volume of refrigerant is required, together with extra supporting equipment such as surge tank and interconnecting piping.

For dry-expansion systems, two of the most commonly used refrigerant liquid metering devices are the capillary tube and the thermostatic expansion valve (Figure 7).

The capillary tube system is normally applied on evaporator coils in factory-assembled self-contained air conditioners up to approximately 10 tons capacity, and is used extensively on the smaller capacity models such as window or room type units.

In this system, the bore and length of a capillary tube are sized so that a full load under design conditions just enough refrigerant liquid is metered from the condenser to the evaporator coil to be completely evaporated. While the capillary tube system does not operate as efficiently over a wide range in operating conditions as a
FIG. 6 - FLOODED COOLING COIL
NO SCALE
thermostatic expansion valve system, its performance is good at design conditions. More details on description and design information for this system will be covered in the system control section of this report.

The thermostatic expansion valve system is in common use for most dry-expansion type coil applications, particularly for field-assembled hermetic type air conditioners. A typical schematic of a coil and thermostatic expansion valve system is shown in Figure 7. The thermostatic expansion valve system depends upon the thermostatic expansion valve to regulate automatically the rate of refrigerant liquid flow to the coil in direct proportion to the rate of evaporation of refrigerant liquid in the coil and thereby maintain the super-heat at the coil suction outlet within the usual predetermined limits of 6° to 10° F.

The housing of the evaporator could be made of galvanized steel, with welded construction and one inch thickness fiberglass insulation to prevent heat loss due to natural convection. Since air is used as the heat transfer medium in the prototype model, it is delivered to the evaporator through a 6-inch round duct. Because the total air quantity supplied to the evaporator was calculated to be 110 CFM, the approximate velocity of it through the duct system will be 550 feet per minute and, therefore, causes a friction loss within this section of duct at approximately 0.10 inch of water per 100 feet (according to the values obtained from the Trane Company's "ductulator").

In order to ensure smooth air flow throughout the duct system, especially at the section near the face of the evaporator coil, a smooth round-to-rectangular transition is provided to connect the rain duct to the evaporator housing (Figures 8 and 9).
The angle of this duct transition is not to exceed 20° as recommended by the American Society of Heating, Ventilation, Air Conditioning, and Refrigeration Engineers.

The electric duct heater is installed at a 4-foot minimum distance from the discharge opening of the circulating fan to warrant its performance as recommended by the manufacturer. The main function of this electric heater in the system is to provide the heat source needed for the measuring level of refrigeration effect delivered by the system evaporator.

A condensate pan is also provided to collect possible condensate water formed around the evaporator when the air velocity at the evaporator surface is too low.

The design features of the coil, such as fin spacing, tube spacing, type of fins, etc., together with the amount of moisture on the coil and the degree of surface cleanliness, determine at which air velocity condensed moisture will be blown off the coil. In this project, since the electric duct heater is installed in the duct system, all the heat absorbed by the evaporator is considered to be sensible heat. Therefore, no moisture is formed at the surface of the evaporator coil. The danger of condensed moisture being carried into the ductwork system is consequently eliminated.

PERFORMANCE OF SENSIBLE COOLING EVAPORATOR COIL

The performance of sensible cooling coils depends in general upon:

1. The overall coefficient (U) of sensible heat transfer between the air stream and the coolant fluid.
2. The mean temperature difference ($\Delta t_m$) between the air stream and the coolant fluid.

3. The physical dimensions and data for the coil, such as the coil face area ($A_a$) in square feet, and total external surface area ($A_0$). (See symbol definitions, p. 32).

The sensible heat cooling capacity of the coil is expressed by the following basic equation:

$$Q = U_0 F_S (A_a) N_r (\Delta t_m)$$  \[1\]

where:

$$F_S = A_0 / (A_a) N_r$$

Assuming no extraneous heat losses, the same amount of sensible heat is lost from the air stream:

$$Q = W_a (C_p) (ta_1 - ta_2)$$  \[2\]

where:

$$W_a = 4.5 (A_a) V_a$$

The same amount of sensible heat is absorbed by the coolant, and for a nonvolatile type is:

$$Q = W_r (C_r) (tr_2 - tr_1)$$  \[3\]

The mean temperature difference in Equation 1 for a non-volatile coolant in thermal counterflow with the air, based on the conventional logarithmic value, is expressed as:

$$\Delta t_m = \frac{(ta_1 - tr_2) - (ta_2 - tr_1)}{(ta_1 - tr_2)} \ln \frac{(ta_1 - tr_2)}{(ta_2 - tr_1)}$$  \[4\]

Considering any coil, whether of bare pipe or of the finned type, the overall heat transfer coefficient ($U_0$) for a given design of coil
with clean, nonfouled surfaces consists of the combined effect of
three individual heat transfer coefficients:

1. The film coefficient of sensible heat transfer \( (f_a) \) between air and the external surface of the coil.

2. The unit conductance \( (1/R) \) of the coil material, i.e., tube wall, fin, ribs, etc.

3. The film coefficient of heat transfer \( (f_r) \) between the internal surface of the coil and the coolant fluid with the coil.

For finned coils the equation for the overall coefficient of heat transfer can be conveniently written:

\[
U = \frac{1}{\frac{1}{n(f_a)} + \frac{B}{f_r}} \tag{5}
\]

in which the term \( n \), called the fin effectiveness, is introduced to allow for the resistance to heat flow encountered in the fins and is defined as:

\[
n = \frac{E(A_s) + A_p}{A_o} \tag{6}
\]

For typical designs of cooling surfaces, the surface ratio \( B \) varies from about 1.03 to 1.15 for bare-pipe coils and ranges from approximately 10 to 30 for fin coils. 11

**Prototype Unit Evaporator Coil Calculations**

**Summary of design criteria:**

1. Unit sensible cooling load: 2,368 Btuh
2. Unit air load: 110 CFM
3. Evaporator entering air dry bulb temperature: 75° F.
4. Evaporator leaving air dry bulb temperature: 53° F.
5. System suction temperature (assumed): 45° F.

7. Film coefficient of sensible heat transfer \( (f_a) \) (assumed) between air and coil surface: 45 Btuh/°F./sq. ft. external.

8. Film coefficient of heat transfer \( (f_r) \) between the internal surface of the coil and the refrigerant: 800 Btuh/°F./sq. ft.

9. Isothermal dry surface air side friction at standard 70° F. conditions, \( \Delta P_{st} \), in inches of water: 0.03 per row (assumed).

EVAPORATOR COIL CALCULATIONS

1. Coil surface area: 110 CFM/200 ft. per min. = .55 sq. ft.

2. Neglecting effect of tube wall, from Equation 5 above.

\[
U = \frac{1}{\frac{1}{34.35 Btuh/\degree F./sq. ft.} + \frac{1.07}{(.8)(45)/800}}
\]

3. From Equation 4:

\[
\Delta t_m = \frac{(75 - 45) - (53 - 45)}{\ln \left( \frac{75 - 45}{53 - 45} \right)} = 16.65
\]

4. From Equation 1, the required surface area is:

\[
A = \frac{2368}{34.35 \times 16.65} = 4.14 \text{ sq. ft. external}
\]

5. Required number of rows:

\[
N_r = \frac{4.14}{1.41 \times .55} = 5.34 \text{ rows deep}
\]

Use 6 rows.

6. The air pressure drop through the evaporator:
\[ P = 0.03 \times 6 = 0.18 \text{ inches of water} \]

**SYMBOLS AND ABBREVIATIONS USED FOR CALCULATIONS**

Aa = Coil face area, sq. ft.

\[ A_0 = \text{External coil surface area, sq. ft.} \]

\[ A_p = \text{Exposed external primary surface area, sq. ft.} \]

\[ A_s = \text{External secondary surface area, sq. ft.} \]

\[ \Delta t_m = \text{Mean temperature difference between air stream and fluid} \]

B = Ratio of external to internal surface area, dimensionless

\[ C_r = \text{Specific heat of nonvolatile coolant, Btu/(pound refrigerant)(Fahrenheit degree)} \]

E = Fin efficiency, dimensionless

\[ f = \text{Convection heat transfer coefficient, Btu/(hour)(Fahrenheit degree)(square foot)} \]

\[ F_s = \text{Coil core surface area parameter} \]

n = Fin effectiveness as defined in Equation 6, dimensionless

\[ N_r = \text{Number of coil rows deep in air flow direction, dimensionless} \]

t = Temperature, Fahrenheit

\[ U = \text{Overall coefficient of sensible heat transfer between the air stream and the coolant fluid} \]

\[ V_a = \text{Coil air face velocity at standard 70° F. conditions, feet per minute} \]

\[ W_a = \text{Mass flow rate, pound per hour} \]

\[ W_r = \text{Fluid mass flow rate, pound per hour} \]

**EVAPORATOR PIPING**

Proper evaporator piping and control are necessary to keep the
conditioned space at the desired temperature and also to the system from surges of liquid ammonia out of the evaporator.

The evaporator piping layout shown in Figure 10 is designed to accomplish the following:

1. Insure proper feed of ammonia refrigerant to the evaporator.

2. Provide practical line sizes without excessive pressure drop.

3. Protect other system components by:
   a. Preventing excessive water and thiocyanate solution from being trapped in the system.
   b. Preventing liquid NH₃ from entering the absorber during operation and shut-down.

The system pipe sizing is based on the Darcy-Weisbach formula:

\[ h = f \times \frac{L}{D} \times \frac{V^2}{2g} \]

where:

- \( h \) = loss of head in feet of fluid
- \( f \) = friction factor
- \( L \) = length of pipe in feet
- \( D \) = diameter of pipe in feet
- \( V \) = velocity in feet per second
- \( g \) = acceleration of gravity = 32.17 ft./sec.²

**Suction Line Design**

Suction lines are the most critical from a design standpoint. The suction line must be designed to return carried-over thiocyanate solution from the evaporator to the absorber and generator under minimum load conditions.

Thiocyanate mist which leaves the generator and passes to the
condenser through the liquid supply lines to the evaporator is almost completely separated from the refrigerant vapor. In the evaporator a distillation process occurs and continues until an equilibrium point is reached. The result is a mixture of thiocyanate and NH₃ rich in liquid NH₃. Therefore the mixture which is separated from the NH₃ vapor can be returned to the absorber by entrainment with the returning gas.

Thiocyanate entrainment with the return gas in a horizontal line is readily accomplished with normal design velocities (500 feet per minute in horizontal line, 1,000 feet per minute in vertical line). Therefore horizontal lines must be installed dead level or pitched towards the absorber.

Suction Risers

The evaporator piping system contains a suction riser. The thiocyanate vapor and solution circulating in the system is returned up the riser only by entrainment with the returning gas. Thiocyanate solution returning up a riser creeps up the inner surface of the pipe. Whether or not thiocyanate solution moves up the inner surface is dependent on the mass velocity of the gas at the liquid surface. The larger the pipe diameter, the greater the velocity required at the center of the pipe to maintain a given velocity at the liquid surface. A riser selected on this basis may be smaller in diameter than its branch or than the suction main and, therefore, a relatively higher pressure drop may occur in the riser (Figure 10).

In actual practice in air conditioning system design, when the suction riser is sized to permit solution mist return at the minima-
operating capacity of the system, the pressure drop in this portion of the line may be too great when operating at full load. If a correctly sized suction riser imposes too great a pressure drop at full load, a double suction riser is used (Figure 11).

**Liquid Line Design**

The amount of liquid line pressure drop which can be tolerated is dependent on the number of degrees subcooling of the liquid NH$_3$. The system liquid line is sized for 2° F. drop under normal circumstances. Since the testing apparatus is set up within air conditioned space (approximately 75° F.), the liquid line need not be insulated.

Since the actual cooling capacity of the prototype model is quite small, the capillary tube or adjustable valve method is suitable for controlling liquid NH$_3$ flow into the evaporator.

The capillary operates on the principle that liquid passes through it much more readily than gas. It consists of a small diameter line which, when used for the purpose of controlling the flow of refrigerant in a system, connects the outlet of the condenser to the inlet of the evaporator.
Chapter 4

SYSTEM CONTROL

As illustrated schematically in Figure 12, the control of various components operating in conjunction with the evaporator of the proposed Solar Energy Powered Absorption Refrigeration System is arranged to achieve the following objectives:

1. The evaporator is not operational when the system solution pump is not in operation.

2. The air circulating fan will not operate unless the system pump is operational.

3. The system air duct electric heater will not be actuated unless the circulating fan is in operation. This is so arranged so that the heater will not be accidently activated to cause fire hazard while the entire system is off.

In designing the control system to achieve the above purposes, all major system components mentioned above are interlocked with the system solution pump through a single pole triple throw relay, and the electric duct heater is directly connected in series with the circulating fan. An air flow switch is also utilized and located in the ductwork system to provide a secondary safe-proof station for the system.

System Sequence of Control

Upon pushing the system master switch button, the solution pump is activated. The solution of thiocyanate and ammonia is pumped from the system absorber to the generator. This action closes the system
main relay holding coil which connects electric circuits at system major components such as: hot water supply pump, condenser fan, evaporator circulating fan, electric duct heater, etc.

As the solar energy heated water reaches the generator heat exchanger coil, the temperature of the solution of NH$_3$ and NaSCN in the generator is rapidly increased. In the generator, NH$_3$ vapor is released when temperature reaches the optimum point, and it starts flowing to the condenser section where heat is rejected into the condenser air flow. Ammonia liquid is, therefore, formed. Because of the high differential in pressure between the system condenser and the evaporator liquid NH$_3$ is transported to the evaporator through the evaporator capillary tube or the adjustable control valve.

As the NH$_3$ liquid evaporates in the evaporator section, the heated air supplied by the circulating fan is blown through the evaporator coil, and the system heat exchanging process occurs. The temperature of air moving through the evaporator is lowered due to the system refrigerant effect.

The actual cooling capacity of the system could be readily calculated from the known quantity of heat supplied by the electric duct heater, or by the measured difference in temperatures of the entering $T_1$ and leaving air $T_2$ flowing through the evaporator. This is, of course, easy to accomplish by installing two thermometers: one at the upstream and the other at the downstream of the evaporator housing, (Figure 1).

An anemometer is used to measure the mass flow of air through the ductwork system (in cubic feet per minute—CFM). When the above data is obtained the total system sensible heat cooling capacity can be
calculated based on the formula:

$$\text{Btu/hr.} = \text{CFM} \times 1.08 \times (T_2 - T_1)$$

where:

- $\text{CFM}$ = cubic feet of air per minute
- $T_1$ = temperature of the entering air
- $T_2$ = temperature of the leaving air
Chapter 5

CONCLUSION

The design of the evaporator section of the Solar Energy Powered Absorption Refrigeration System is only one of the major steps towards completion of the entire project.

The incorporation of solar energy into present technology for controlling the environment is indeed a major step forward in the efforts of engineers and scientists to find alternate sources of energy for solving the current problem of shortage of fossil fuel.

Like its many predecessors in the field of refrigeration, this system will need further testing and improving before it is commercially feasible. On the other hand, unlike its numerous predecessors, the Solar Energy Powered Absorption Refrigeration System, when perfected, will certainly be one of the most lasting cooling systems ever developed, due to the fact that the availability of the energy from the sun is a replaceable source.

Although the stated purpose of applications for the designed system is for use in air conditioning systems in residential and commercial projects, this system will certainly have many other feasible applications in both industrial and commercial processes.

In the light of many problems such as safety, environmental effect, and costs confronted in the search for alternate sources of energy with nuclear systems, wind energy, geothermal, biothermal, and others, the method of direct utilizing solar energy to heat, cool, and
power machinery may prove to be the most practical of all.

Together with the applications of systems recently developed for use in various commercial and industrial projects such as heat reclama-
tion and economizer cycle, the Solar Energy Powered Absorption Refrig-
eration System could be one of the most effective systems for answer-
ing the problems of energy conservation and fossil fuel shortage.

The design of the evaporator for this project system may, there-
fore, be considered as a small part in the contributing efforts
towards achieving the above goal.
REFERENCE LIST


2. Same as (1) above.

3. Calcium Chloride for Refrigeration Brine (Manual RM-1, Calcium Chloride Institute, Washington, 6, D.C.

4. Same as (3) above.


7. Data from E. I. DuPont de Nemours & Company.


12. Same as (11) above, p. 6.11.


ADDENDUM
Typical Solar Collector Panel Installations

Figure 13