CALIFORNIA STATE UNIVERSITY, NORTHRIDGE

Analysis of a Two Stage Absorption Refrigeration Cycle

A Project submitted in partial fulfillment of the requirements for the degree of Master of Science in Engineering

by

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ABSTRACT

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A first law thermodynamic analysis of a two-stage absorption refrigeration cycle using sodium thiocyanate and ammonia is presented. It includes descriptions of the simplified relationships developed for the thermodynamic data and outlines the logic of a computer program developed for the analysis. It is concluded that the cycle coefficient of performance is competitive with the lithium bromide - water system which is presently commercially available.
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I INTRODUCTION

Although interest in utilizing solar energy for cooling of buildings has been in evidence for many years (1, 2)*; the oil embargo of 1973 precipitated an order-of-magnitude increase in activity along these lines. (3) With respect to heating and cooling of buildings, the potential for using solar energy for heating has received the greatest emphasis because in some cases an investment in equipment for this purpose alone could be economically justified, whereas a similar investment for cooling alone (in the continental U.S.) would not. (4) Indeed, the addition of a solar cooling system to an already capitalized solar heating system is barely marginal at the time of this writing. (19) This conclusion could be reversed by a significant improvement in the performance or reduction in first cost of the cooling concepts, or a similar increase in the cost of electricity. Thus several projects were initiated (5) to examine the relative merits of various methods to use solar heat for air conditioning. Absorption coolers using modified commercial equipment were installed in several demonstration projects (6), however optimum performance could not be obtained because the equipment was not specifically designed for the relatively low temperature solar input. Recently, however, several manufacturers have made available equipment which will operate with a nominal coefficient of performance (COP) of approximately 0.7 with an inlet hot water temperature as low as 165°F. (18) These units use lithium bromide-water as the absorption pair. Indeed, the lithium bromide

* Numbers refer to References in the Bibliography
-water pair now dominates the field for absorption water chillers used in air conditioning regardless of the heat source.

One can consider two approaches to the improvement of the economics of using solar driven absorption air conditioning: examine alternative absorption pairs and examine alternative refrigeration cycles.

Many absorption pairs are known to be operable and each of these has its own operating distinctions. Some of these have been used either in commercial machines or in experimental devices. Prominent among them is the ammonia-water pair which was first used by Faraday in 1824 and which was long dominant in the field of absorption cooling. Recently there have appeared in the literature several descriptions of experiments using sodium thiocyanate-ammonia (NaSCN-NH$_3$) in an intermittent refrigeration device. These tests indicated a superior performance over the venerable ammonia-water pair. An unpublished report by the 1976 C. S. U. N. Senior Design Class describes a continuous system using the NaSCN-NH$_3$ pair, but no test data was available at the time of this writing.

With respect to the second approach, many alternative cycles have also been studied. A machine incorporating a two stage cycle is available for the lithium bromide-water pair that boosts the COP from $\sim$0.7 (single stage) to $\sim$1.1. The term 'double effect' is usually applied to such a cycle because the heat input to the high temperature stage is used not only to release refrigerant directly but also again in the low temperature stage where the heat of condensation is used to drive out an additional quantity of refrigerant. The term 'two stage' has also been applied to a cycle using a resorber, but this is not considered here because of the extra equipment required.
The study presented herein deals with a double effect absorption cycle using NaSCN-NH\textsubscript{3} as the absorption pair. Because the ultimate question to be answered is the performance of a complete system including solar collectors and energy storage, it was deemed appropriate to prepare a computer program which could eventually be modified or incorporated into a larger overall analysis. Therefore this report includes a description of the computer code developed.
II SYSTEM DESCRIPTION

Only a brief description of the absorption system will be given here because of the many available references which provide a depth that need not be duplicated. For the reader not familiar with these devices, references 12, 13, 14, 15 and especially 17, provide excellent background material.

Figure 1 depicts a schematic arrangement of a possible configuration for such a device. The hot fluid from the solar input system is brought to the high temperature generator where refrigerant is driven out of the NaSCN-NH$_3$ solution. This refrigerant is condensed in the lower temperature generator where the heat of condensation is used to drive out an additional quantity of refrigerant. The two generators must operate at different pressure levels to maintain the temperature of condensation of the upper (high temperature) stage high enough to drive refrigerant out of solution in the lower (low temperature) stage generator. Because of this pressure difference, a pressure reducing device is required in the line which leads refrigerant out of the upper stage to the lower stage condenser. All the refrigerant developed by both stages is cooled in the condenser by the ultimate heat sink before being sent to the evaporator through a pressure reducing device. The refrigerant vaporizes in the evaporator at low pressure and temperature, extracting heat from the cooling load. The solution returning from the generators (there is a continuous recirculation between generators and absorber) is deficient in refrigerant, and when brought to the absorber and kept cool by the ultimate heat sink, has a strong affinity for the vapor given off in the evaporator. The cycle is completed by the pumps which deliver the enriched (by absorbing refrigerant vapor) solution.
TWO STAGE ABSORPTION CYCLE SCHEMATIC  FIG 1
to the high pressure and temperature generators.

Two heat exchangers are normally used so that the circulation of solution between the generators and absorber does not transfer excessive heat between these components. The solution circuit shown, where a separate recirculation loop and pump is used for each generator, is only one of several possibilities. The commercial unit described in reference 20 uses a single pump to deliver the solution to the upper generator and it passes through the lower generator before returning to the absorber. However, it can be shown that any such series arrangement requires a higher temperature and pressure in the upper generator than the one shown on figure 1, and this is not desirable.
The thermodynamic data for this analysis was all taken from references 12 and 17. Although reference 21 presents relationships for the properties of ammonia, they are relatively complex and unwieldy. Less intricate relationships were developed during this study to simplify the programming effort, and with one minor exception noted below, they are accurate well within 1%. The data for sodium thiocyanate-ammonia in reference 17, which is the only known source for such information, presented a problem because the range of data did not encompass the entire range required for this study. Thus it was necessary to consider the fact that the form of the relationship incorporated into the computer program must be one that can be extrapolated with some confidence. The most critical of these data for this study is that for vapor pressure of solution at a fixed ammonia concentration and variable temperature. Fortunately, as shown in the companion paper (22), a nearly straight line results from a log $P$ vs. $1/T$ plot. (This, incidentally, is the form of the relationship used for pure ammonia.) Therefore it was adopted as having the greatest potential for credible extrapolation. Coefficients were obtained by linearizing between the extreme data point reported and a point corresponding to the expected condition in the absorber. Thus very little error can be expected from the calculation of absorber concentration, and the error from extrapolation is minimized. There is no doubt, however, that any further or more detailed cycle studies using the NaSCN-NH$_3$ pair would benefit greatly from an extended characterization of the properties of this absorption pair.
In the following discussion, where each of the relationships is individually addressed, a table is given for each which shows both the basis data and the corresponding calculated values. And in order to clarify the correspondence between the computer coding and the analytical expression, the function name used in the code is given in parenthesis.

A. Specific Heat of Ammonia Vapor (CPVAP(P, T))

\[ C_p = 0.56 + 1.23 (P - 75.0) / T - 0.91 \] for \( P \geq 75.0 \)

\( C_p \) BTU per pound per \(^\circ\)F

\( P \), PSIA

\( T \), °F

Data below (in parenthesis) is derived from enthalpy tables in reference 12.

<table>
<thead>
<tr>
<th>Temperature</th>
<th>100 PSIA</th>
<th>150 PSIA</th>
<th>200 PSIA</th>
<th>250 PSIA</th>
<th>300 PSIA</th>
</tr>
</thead>
<tbody>
<tr>
<td>50°F</td>
<td>.72(.709)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>100°F</td>
<td>.60(.61)</td>
<td>.69(.69)</td>
<td>.77(.78)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>150°F</td>
<td>.58(.58)</td>
<td>.62(.62)</td>
<td>.66(.66)</td>
<td>.70(.72)</td>
<td>.74(.78)</td>
</tr>
<tr>
<td>200°F</td>
<td>.57(.57)</td>
<td>.59(.59)</td>
<td>.62(.62)</td>
<td>.64(.65)</td>
<td>.67(.68)</td>
</tr>
<tr>
<td>250°F</td>
<td>.57(.57)</td>
<td>.58(.59)</td>
<td>.60(.60)</td>
<td>.61(.62)</td>
<td>.63(.64)</td>
</tr>
<tr>
<td>300°F</td>
<td>.57(.57)</td>
<td>.58(.59)</td>
<td>.59(.60)</td>
<td>.60(.61)</td>
<td>.61(.62)</td>
</tr>
</tbody>
</table>

This relationship was the most troublesome of the ammonia properties because of its dependence on both temperature and pressure. A comparison of calculated and basis values reveals a progressive reduction in accuracy with increasing pressure, with a maximum of about 5% near saturation at 300 PSIA. However, the specific heat of ammonia vapor is used to calculate a heat quantity which is only about 10% of the total heat input to the generator, which fact permits a correspondingly greater error in vapor \( C_p \) without seriously affecting the end result. This is further ameliorated by the fact that the expression actually evaluated is an integral of
Cp dT from condenser temperature to generator temperature. Thus the error incurred is an average over this temperature range. This relationship introduces errors of less than 1% to the final result.

B. Latent Heat of Ammonia (LATENT(T))

\[ LH = 69.615(271.4 - T)^{3.8} \times 10^{-7} T^3 + 274 \]

LH Latent Heat in BTU per pound

\( T \) °F

Data in parenthesis below is taken from reference 12.

<table>
<thead>
<tr>
<th>Temp (°F)</th>
<th>LH (BTU/lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>536.15(536.2)</td>
</tr>
<tr>
<td>50</td>
<td>527.27(527.3)</td>
</tr>
<tr>
<td>75</td>
<td>503.7(503.7)</td>
</tr>
<tr>
<td>100</td>
<td>477.79(477.8)</td>
</tr>
<tr>
<td>125</td>
<td>448.8(449)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Temp (°F)</th>
<th>LH (BTU/lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>150</td>
<td>415.94(416)</td>
</tr>
<tr>
<td>175</td>
<td>377.7(377)</td>
</tr>
<tr>
<td>200</td>
<td>332.03(332)</td>
</tr>
<tr>
<td>225</td>
<td>274.4(273.5)</td>
</tr>
<tr>
<td>250</td>
<td>192.6(192)</td>
</tr>
</tbody>
</table>

The latent heat of ammonia dominates the heat quantities calculated for the generators, condenser, evaporator and absorber. A comparison of basis and calculated values shows that this relationship yields values well within 1% accuracy.

C. Ammonia Vapor Pressure (VP(T))

Ammonia Saturation Temperature (TV(P))

\[ VP = \frac{1362729}{\exp(4915.08/(TV+460.0))} \]

\[ TV = -4915.08/\ln(PV/1362729.) - 460.0 \]

VP PSIA

TV °F

Data in parenthesis below is taken from reference 12.

<table>
<thead>
<tr>
<th>Temp (°F)</th>
<th>VP (PSIA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>73.32(73.32)</td>
</tr>
<tr>
<td>50</td>
<td>88.91(89.19)</td>
</tr>
<tr>
<td>75</td>
<td>139.48(140.5)</td>
</tr>
<tr>
<td>100</td>
<td>210.2(211.9)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Temp (°F)</th>
<th>VP (PSIA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>305.86(307.8)</td>
</tr>
<tr>
<td>150</td>
<td>431.6(433.2)</td>
</tr>
<tr>
<td>175</td>
<td>592.7(593.5)</td>
</tr>
<tr>
<td>200</td>
<td>794.66(794.7)</td>
</tr>
</tbody>
</table>

Errors here are less than 1%.
D. **Enthalpy of Liquid Ammonia**

\[
H = 0.67197^T + 1.14054^T \quad T \geq 200.
\]

\[
H = 52.6 - 0.26T + 0.67197^T \quad T < 200.
\]

*H BTU per pound*

*Data in parenthesis below taken from reference 12.*

<table>
<thead>
<tr>
<th>Temperature °F</th>
<th>Value (°F)</th>
<th>Value (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>86.74</td>
<td>150</td>
</tr>
<tr>
<td>50</td>
<td>97.22</td>
<td>200</td>
</tr>
<tr>
<td>100</td>
<td>157.36</td>
<td>250</td>
</tr>
</tbody>
</table>

This relationship is in error by less than 1%.

E. **Saturation Temperature of Solution**

*Data in parenthesis below taken from reference 17.*

For 54.3% by weight NH\(_3\)

\[
T = -5141.613/ \ln(P/952790.) - 460.
\]

<table>
<thead>
<tr>
<th>Pressure PSIA</th>
<th>Value (°F)</th>
<th>Value (°F)</th>
<th>Value (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>77.5</td>
<td>86. (86)</td>
<td>235</td>
<td>158.9 (158)</td>
</tr>
<tr>
<td>103 PSIA</td>
<td>103 (104)</td>
<td>296</td>
<td>176.6 (176)</td>
</tr>
<tr>
<td>137.5 PSIA</td>
<td>121.4 (122)</td>
<td>367</td>
<td>194. (194)</td>
</tr>
<tr>
<td>179 PSIA</td>
<td>139.3 (140)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Maximum error is just under 1%.

For 46.3% by weight NH\(_3\)

\[
T = -5584.398/ \ln(P/1277352.) - 460.
\]

<table>
<thead>
<tr>
<th>Pressure PSIA</th>
<th>Value (°F)</th>
<th>Value (°F)</th>
<th>Value (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>64 PSIA</td>
<td>104. (104)</td>
<td>152</td>
<td>158. (158)</td>
</tr>
<tr>
<td>84 PSIA</td>
<td>119.9 (122)</td>
<td>193</td>
<td>174.8 (176)</td>
</tr>
<tr>
<td>113.5 PSIA</td>
<td>138.6 (140)</td>
<td>250</td>
<td>194. (194)</td>
</tr>
</tbody>
</table>

Maximum error is less than 1.8%.

For 38.7% NH\(_3\) by weight

\[
T = -4845.7/ \ln(P/247706.) - 460.
\]
77 PSIA  140, (140)  121 PSIA  175.6(176)  97.5 PSIA  158.1(158)  150 PSIA  194.1(194)

Maximum error is much less than 1%.

F. Specific Heat of the Solution (CPSOL(X))

\[ C_p = 0.26 + 0.72X \]

\[ C_p \text{ BTU per pound per } ^\circ F \]

\[ X \text{ NH}_3 \text{ weight fraction} \]

Data below in parenthesis taken from reference 17.

\[ \cdot 4071 \text{ NH}_3 \quad .553(.556) \quad .5882 \text{ NH}_3 \quad .68(.67) \]

\[ .553 \text{ NH}_3 \quad .658(.63) \quad .78 \text{ NH}_3 \quad .82(.82) \]

The maximum error shown here is about 4%. The effect on the overall result is less than 1% however because the heat quantity involved is less than 10% of the total input to the generator.

G. Heat of Solution of NaSCN (HTSOL(X))

\[ H = 267.0X^{0.8475} \]

\[ H \text{ BTU per pound NaSCN} \]

\[ X \text{ NH}_3 \text{ weight fraction} \]

Data below taken from reference 17.

\[ 0.968 \text{ NH}_3 \quad 259.7(260.15) \quad 0.759 \text{ NH}_3 \quad 211.4(217.9) \]

\[ 0.810 \text{ NH}_3 \quad 223.3(226.8) \quad 0.528 \text{ NH}_3 \quad 155.4(158.6) \]

The maximum error is 3%, but again the quantity of heat is a fraction of the total and the effect on the end result is less than 1%. 
IV ANALYSIS PROCEDURE

Figure 2 is an outline of the sequence of calculations on an idealized state diagram. Vapor pressure versus temperature characteristics for a solution of constant composition appears as nearly a straight line on a log P vs. 1/T plot in our range of interest. The leftmost line is for pure ammonia. The letter mnemonic symbols are the same as those used in the computer program. The encircled numbers give the sequence in which the various state points are obtained. Calculations are based on input values of:

- Evaporator Temperature \( TE \)
- Absorber Temperature \( TA \)
- Condenser Temperature \( TC \)
- Temperature Excursion in Lower Generator \( DTG1 \)
- (This is the difference between generator solution outlet temperature and incipient generator inlet solution boiling temperature)
- Temperature Excursion in Upper Generator \( DTG2 \)
- Temperature differential between upper stage condensing temperature and temperature leaving lower generator \( DELT \)
- Heat Exchanger Effectiveness \( FTVNS \)

1) The desired evaporator temperature (input as TE) is used to obtain the evaporator pressure \( PE \) by means of the relationship for vapor pressure of ammonia.

2) The temperature at which the absorber is maintained by the ultimate heat sink (input as TA) is used together with the evaporator pressure \( PE \) to obtain the ammonia concentration leaving the absorber \( XABSO \) by a linear interpolation routine. \( \text{FRAC}\text{T}(P, T, X) \)
TWO STAGE ABSORPTION CYCLE STATE POINTS

FIG. 2
Absorber and evaporator pressures are considered equal. This is possible because there is a unique equilibrium concentration of ammonia in solution at a given temperature and pressure. Subroutine FRAC.T incorporates the saturation temperature relationships for several solution compositions discussed in section III. It first calculates the saturation temperatures corresponding to the absorber pressure for several compositions and interpolates linearly to find a composition corresponding to the absorber temperature (TA).

3) The temperature at which the condenser is maintained by the ultimate heat sink (input as TC) is used to obtain the condenser pressure by means of the relationship for vapor pressure of ammonia.

4) The condenser pressure (PC) is used with the ammonia concentration leaving the absorber and entering the first stage generator (XABSO) to find the temperature at which evaporation begins (TGIIN) by a linear interpolation routine (TMIX(P, X, T)). This is possible because there is a unique equilibrium temperature for a solution of given ammonia concentration at a given pressure. Subroutine TMIX also incorporates the saturation temperature relationships for the several solution compositions discussed in section III. It first calculates the saturation temperatures corresponding to the condenser pressure for several compositions and interpolates linearly to find a temperature corresponding to the generator inlet composition (XABSO). Condenser pressure and first stage generator pressure are considered equal.

5) The temperature of the solution leaving the generator (TCIOUT) is obtained by adding a temperature excursion input (DTGI) to the incoming temperature (TGIIN). This temperature excursion is related to the circulation rate, i.e., the ratio of absorber - generator solution weight flow and the weight flow of refrigerant driven out. A value of about 25°F yields a recirculation rate of 10.
6) The ammonia concentration leaving the lower generator \((X_{GIOUT})\) is obtained from pressure \((P_C)\) and temperature \((T_{GIOUT})\) using subroutine \(FRACT\) as in (2) above.

7) The equilibrium solution temperature entering the absorber from the lower generator \((T_{ABSIN})\) is obtained from \((P_E)\) and \((X_{GIOUT})\) using subroutine \(TMIX\) as in (4) above.

This completes the determination of the cycle state points for the low temperature stage.

The heat rejected in the condenser by the refrigerant driven out of the lower generator \((Q_{AM})\) is composed of a latent heat and ammonia superheat calculated in subroutine \(QCONDNS(P, T_1, T_2, \Omega)\). Ammonia superheat is calculated by integrating \(C_p \, dT\) from the temperature of condensation to an average boiling temperature in the generator. This is valid because there is an almost linear relationship between composition and temperature as the ammonia is being driven out. Both the specific heat and latent heat relationships are discussed in section III.

The heat delivered to the lower stage generator is obtained by adding to the condenser heat rejection \((Q_{AM})\) a heat of solution effect \((Q_{GENS})\), a quantity of heat required because of less-than-unity heat exchanger effectiveness \((Q_{HXGI})\) and the sensible heating of the solution from inlet to exit \((Q_{SENSGI})\). A heat exchanger effectiveness \((FTVNS)\) of unity implies that no heat is transferred from generator to absorber by the recirculating solution. However, it also implies an infinitely large heat exchanger. In practice, an economic trade-off is made between the cost of the heat exchanger and the improvement in cycle performance obtained by using larger and larger heat transfer areas. Real machines normally have heat exchangers with effectiveness values of 0.9 or less.
The heat extracted from the cooling load in the evaporator by the refrigerator generated in the lower stage (QEVAPl) is slightly less than the latent heat of vaporization of ammonia at evaporator pressure because of the fact that refrigerant does not enter the evaporator at the temperature of evaporation. The latent heat must be diminished by the difference in liquid enthalpy at condenser temperature and that at evaporator pressure in order to calculate this cooling effect. This is the same as saying that a constant enthalpy throttling is assumed in the pressure breakdown device between the condenser and evaporator.

The heat rejected in the absorber is composed of the latent heat plus a heat exchanger effectiveness term (QHXABSl), a sensible heating term (QSENSAl) and the heat of solution effect.

All values are normalized to the first stage solution pump flow. Pump head is calculated (HEADl) in ft. lb. per lb., but no correction for this energy input is made. Other approximations are made where appropriate, as for example taking average values to eliminate elaborate computational procedures. Further, the input to the analysis is designed to avoid troublesome iterative loops.

Analysis of the second stage begins with:
8) The condensing temperature of the second stage (TCG2) is obtained by adding a temperature differential (DELT) to the first stage generator outlet temperature. This is a driving potential required to transfer heat from the condensing high temperature stage refrigerant to the lower generator.
9) Second stage condensing pressure (PG2) is obtained from (TCG2) by means of the relationship for vapor pressure of ammonia.
10) The equilibrium second stage inlet temperature (TG2IN) is obtained from the condensing pressure (PC2) and the solution
concentration leaving the absorber \( X_{\text{ABSO}} \) as in (4) above using subroutine \( \text{TMIX} \).

11) The second stage generator discharge temperature \( (T_{\text{C2OUT}}) \) is obtained by adding a temperature excursion input \( (\Delta T_G2) \) to the inlet temperature, \( (T_{\text{G2IN}}) \). This temperature excursion is not only related to solution recirculation rate as discussed under (5) above, but also affects the ratio of solution flows of the two stages. Again a value of 25°F will give a recirculation ratio of about 10, but it would also yield an upper stage solution flow about twice that of the lower stage. To maintain equal pump flows, the upper stage temperature excursion \( (\Delta T_G2) \) should be about twice that of the lower stage.

12) The second stage generator discharge solution refrigerant concentration \( (X_{\text{G2OUT}}) \) is obtained by a linear interpolation routine \( \text{FRAC T} \) using temperature \( (T_{\text{G2OUT}}) \) and pressure \( (P_G2) \) as in (6) above.

13) The equilibrium solution temperature entering the absorber from the upper stage \( (T_{\text{ABS2I}}) \) is obtained from \( (X_{\text{G2OUT}}) \) and \( (P_E) \) using subroutine \( \text{TMIX} \) as in (7) above.

A similar set of heat quantities is obtained as described for the first stage, with the mnemonic symbols incorporating the digit 2. Because of the nature of the input data, the flow in the second stage must be adjusted to yield the proper heat balance in the first stage generator. The ratio between second and first stage pump flows \( (\text{WRATIO}) \) will depend most strongly on the chosen generator temperature excursions, \( (\Delta T_G1) \) and \( (\Delta T_G2) \). Refrigerant yields for the two circuits are labeled \( (W_{AM1}) \) and \( (W_{AM2}) \).

Again, pump work for 100% efficient pumping is calculated in ft. lb. per lb. but is not included in the heat balance. This is conservative with respect to COP calculations because pump work
will reduce the required generator heat input. It is included because it is indicative of the electrical input required to drive the pumps.

Coefficient of performance is calculated in three ways: for each stage separately (COP1) and (COP2) and for the overall cycle (COPVRAL).

Operating details of the computer program and the form of input and output is described in the following section.
The program is written in FORTRAN and was implemented using the Network Operating System (NOS) through a remote terminal. A listing appears at the end of this section.

Seven input data are required: a baseline case has been incorporated through the use of data statements, but line directed input permits modification of any or all of these data. When running the program, it first responds with a question mark, which is a request for data from the terminal. The base data incorporated is as follows, in order:

- TE (evaporator temperature) 45°F
- TA (absorber temperature) 85°F
- TC (condenser temperature) 90°F
- DTG1 (temperature breadth in lower generator) 30°F
- DTG2 (temperature breadth in upper generator) 30°F
- DELT (difference between second or upper stage condensing temperature and first or lower stage generator outlet temperature) 5°F
- FTVNS (heat exchanger effectiveness) 0.9

If any of these values is to be retained, a comma is typed in response to the question mark; if any is to be changed, the desired value followed by a comma is entered. Thus, seven commas in response to the question mark prompting will cause the base case to be run. Similarly, three commas followed by the number 40.0 and four commas will change the value of DTG2 from 30°F to 40°F.

A study of the listing shows many write statements which are identified by a fixed point number. These were originally incorporated as a diagnostic aid, but were retained because this program
may likely be incorporated in a larger analysis where such diagnostics would be useful. Further, the printout of these intermediate values would be informative for more detailed analyses of the cycle, as for example heat transfer surface area sizing. For the purposes of this study, the printout identified by the number 10 is most significant as the first value following this identifier is the overall system COP. Of interest also is the fourth value following the identifying number 6, which is the solution temperature leaving the second stage generator.
FIGURE 3
COMPUTER PROGRAM FLOW CHART
List of Variables

TE evaporator temperature
TA absorber temperature
TC condenser temperature
DTGI first stage generator temperature excursion
DTG2 second stage generator temperature excursion
DELT temp. differential between 2nd stage condensing temp. and max. temp. in 1st stage generator
FTVNS heat exchanger effectiveness
PE evaporator pressure
PC condenser pressure
XABSO ammonia fraction in solution leaving absorber
TGIIIN 1st stage generator equilibrium inlet temperature
TGIOUT 1st stage generator solution discharge temp.
XGIOUT ammonia fraction in solution leaving lower generator
WAM1 weight flow of refrigerant from 1st stage
TABSIN equilibrium solution temp. inlet to absorber from 1st stage
QAM condenser heat rejection due to 1st stage
QHXGI heat exchanger correction to lower generator heat input
QGENS heat of solution effect in 1st stage generator
QDTGI sensible heat load of solution in lower generator
QDTABSI sensible heat load of solution in absorber due to 1st stage
QGEN1 total heat input to lower temperature generator
QEVAPI cooling in evaporator due to 1st stage
QHXABSI heat exchanger effectiveness correction to absorber heat rejection due to 1st stage
QABS1 heat rejection from absorber due to lower stage
COP1 COP of 1st stage
TCG2 condensing temperature in second stage
PG2 condensing pressure (generator pressure) in 2nd stage
TG2IN  2nd stage generator equilibrium inlet temperature
TG2OUT  maximum solution temperature leaving second stage
XG2OUT  ammonia fraction in solution leaving 2nd stage generator
WAM2  weight flow or refrigerant from first stage
QGEN2S  heat of solution effect in 2nd stage
QAM2  heat of condensation of upper stage
QHXG2  heat exchanger effectiveness correction to upper generator heat requirements
TABS2I  equilibrium solution temperature inlet to absorber from high temperature stage
WRATIO  ratio of solution flow rates between the two stages upper/lower
QDTG2  sensible heat load of solution in upper generator
QDTABS2  sensible heat load of solution in absorber due to 2nd stage
QHXABS2  heat exchanger effectiveness correction to absorber due to upper stage
QGEN2  heat input to upper generator
QEVP2  cooling in evaporator due to 2nd stage
OABS2  heat rejection by absorber due to 2nd stage
COP2  COP of 2nd stage
COPVRAL  overall cycle COP
ABSVAL  total heat rejection from absorber
HEAD1  first stage pump head
HEAD2  second stage pump head
00100 PROGRAM PERFORM(INPUT,OUTPUT,TAPE5=INPUT,TAPE5=OUTPUT)
00110 DATA TE/45.0/,TA/35.0/,TC/90.0/,DTG1/30.0/,DTG2/30.0/
00120 DATA DELT/5.0/,FUUVNS/0.0/
00130 CPVAP(P,T)=0.56+11.23/(T**1.91)*(A(MAXL(P,75.))-75.)
00140 LATENT(T)=69.615*(271.4-T)**0.375-.00000038*T**3.272
00150 VP(T)=1362709.0/EXP(4915.03/(T+460.0))
00160 TV(P)=-4915.03/ALOG(P/1362709.)-450.0
00170 T54(P)=-5141.613/ALOG(P/952790.)-450.0
00180 T45(P)=-5584.393/ALOG(P/1277352.)-450.0
00190 T39(P)=-4845.7/ALOG(P/247706.)-450.0
00200 T34(P)=-5252.312/ALOG(P/739132.)-450.0
00210 NTMLPY(T)=52.25*MIN1(T,200.)+.67197*T**1.14054
00220 HTSOL(PC)=287.0*PCT**0.8475
00230 C(SOL(X)=0.25+0.72*X
00240 READ*,TE,TA,TC,DTG1,DTG2,DELT,FTVNS
00250 WRITE(6,10) TE,TA,TC,DTG1,DTG2,DELT,FTVNS
00260 10 FORMAT(5F7.1,F52,F6.2)
00270 PE=VP(TE)
00280 PC=VP(TC)
00290 T54A=T54(PC)
00300 T45A=T45(PC)
00310 T39A=T39(PC)
00320 T34A=T34(PC)
00330 J=1
00340 WRITE(6,15) J,T54A,T45A,T39A,T34A
00350 T54G=T54(PC)
00360 T45G=T45(PC)
00370 T39G=T39(PC)
00380 T34G=T34(PC)
00390 J=2
00400 WRITE(6,15) J,T54G,T45G,T39G,T34G
00410 15 FORMAT(15,4F15.5)
00420 CALL FRAC(PE,TE,XA80)
00430 CALL TMIX(PC,XA80,TGIN)
00440 TGLOUT=TGIN+DTG1
CALL FRACT(PC,TGLOUT,XG1OUT)
WAM1 = (XABS0-XG1OUT)/(1.0-XG1OUT)

CALL TMIX(PE,XG1OUT,TABSIN)
J=3

WRITE(6,15) J,XABS0,TGLIN,XG1OUT,TABSIN
CALL QCONDNS(PC,TGLIN,TGLOUT,WAM)
QAM=WAM*WAM1
QHXL=CPBSOL(XABS0)*(TGLIN-TA)-FTVNS*(TGLOUT-TA)*(1.0 + WAM1)*CPBSOL(XG1OUT)
QGNS=HTSOL(XABS0)-HTSOL(XG1OUT)
QDTG1=(1.0-WAM1)*CPBSOL(XG1OUT)*(TGLOUT-TGLIN)

QG1N1=QHXL+QGNS+WAM+QDTG1

J=5
WRITE(6,15) J,QAM,QDTG1,QHXL,QGNS
QEVAP1=LATENT(TE)+NTHLPY(TE)-NTHLPY(TC)
QEVAP1=QEVAP1*WAM1
QHABS1=CPBSOL(XG1OUT)*(1.0-WAM1)*(TGLOUT-TABSIN-FTVNS*(TGLOUT-TA))
QABS1=QHABS1+QGNS+QDTABS1+LATENT(TE)*WAM1
QCP1=QEVAP1/QG1N1

J=5
WRITE(6,15) J,QDTABS1,QGEN1, QEVAP1,QHABS1

TCG2=TGLOUT+DELT
PG2=VP(TCG2)

CALL TMIX(PG2,XABS0,TG2IN)
TG2OUT=TG2IN+DTG2

J=6
WRITE(6,15) J,TCG2,PG2,TG2IN,TG2OUT

CALL FRACT(PG2,TG2OUT,XG2OUT)
WAM2 = (XABS0-XG2OUT)/(1.0-XG2OUT)

QGEN2S=HTSOL(XABS0)-HTSOL(XG2OUT)
QGEN2S=QGEN2S*(1.0-XABS0)

CALL QCONDNS(PG2,TG2IN,TG2OUT, QAM2)
00800 QAM2=QAM2*WAM2
00810 QHX32=CPSOL(XG2SO)*TG2IN-TA)-FTVNS*(1.0-WAM2)
00820+ *(TG2OUT-TA)*CPSOL(XG2OUT)
00830 CALL TIMX(PE,XG2OUT,TA,BS2)
00840 WRTIO=QGEN2/QTG2
00850 QDTG2=(1.-WTG2)*CPSOL(XG2OUT)*(TG2OUT-TG2IN)*WRTIO
00860 QDTAB32=1.-WTG2)*CPSOL(XG2OUT)*(TA,BS2-PE)*WRTIO
00870 J=7
00880 WR(5,15) J,XG2OUT,QTG2,QTG2,QHXG2
00890 QHXAB32=1.-WTG2)*CPSOL(XG2OUT)*(TG2OUT-TA,BS2IN-FTVNS*(TG2OUT-TA))
00900 QGEN2S=QGEN2*WRTIO
00910 QAM2=QAM2*WRTIO
00920 QHX32=QHX32*WRTIO
00930 QGEN2=QAM2+QGEN2S+QAM2+QDTG2
00940 QAM2=QAM2*WRTIO
00950 J=75
00960 WRITE(6,15) J,WAM1,WAM2,QTG2,QDTAB32
00970 J=3
00980 WRITE(6,15) J,WRATIO,QHXAB32,QGEN2S,QAM2
00990 QEVAP2=WAM2*QEVAP1/WAM1
01000 QAB2S2=QGEN2S+QHXAB32+QDTAB32+LATENT(TE)*WAM2
01010 COP2=QEVAP2/QGEN2
01020 QCONDXT=WAM2*(NTHLPY(TG2))-NTHLPY(TC))
01030 COPVRAL=(QEVAP1+QEVAP2)/QGEN2
01040 QCONDVAL=QCONDXT+QAM+QAM2
01050 ABSVAL=QAB2S1+QAB2S2
01060 J=9
01070 WRITE(6,15) J,QEVAP2,QAB2S,COP2,QCONDXT
01080 J=10
01090 WRITE(6,15) J,COPVRAL,QCONDVAL,ABSVAL,QHX32
01100 HEAD1=2.554*(PC-PE)
01110 HEAD2=2.554*(PG2-PE)
01120 WRITE(6,15) J,TG1OUT,TG2OUT,HEAD1,HEAD2
01130 END
SUBROUTINE FRACT(P,T,X)
T54A=-5141.613/ALOG(P/952790.0)-460.0
T46A=-5584.393/ALOG(P/1277352.0)-460.0
T39A=-4845.7/ALOG(P/247706.0)-460.0
T34A=-5352.612/ALOG(P/739192.0)-460.0
IF(T.GT.T46A) GO TO 50
X=(T-T54A)/(T54A-T46A)*0.073+0.541
GO TO 150.
50 IF(T.GT.T39A) GO TO 100
X=(T-T46A)/(T46A-T39A)*0.076+0.463
GO TO 150.
100 X=(T-T39A)/(T39A-T34A)*0.045+0.387
150 CONTINUE
RETURN
END

SUBROUTINE TMIX(P,X,T)
T54A=-5141.613/ALOG(P/952790.0)-460.0
T46A=-5584.393/ALOG(P/1277352.0)-460.0
T39A=-4845.7/ALOG(P/247706.0)-460.0
T34A=-5352.612/ALOG(P/739192.0)-460.0
IF(X.LT.0.451) GO TO 50
T=(.541-X)/0.078*(T46A-T54A)+T54A
GO TO 150.
50 IF(X.LT.0.387) GO TO 100
T=(0.463-X)/0.076*(T39A-T46A)+T46A
GO TO 150.
100 T=(0.387-X)/0.045*(T34A-T39A)+T39A
150 CONTINUE
RETURN
END
01440 SUBROUTINE QCONDLS(P,T1,T2,Q)
01450 TSAT=-4915.03/(LOG(P/1362729.0)-460.0)
01460 TAVGEN=(T1+T2)/2.0
01470 QLAT=59.615*(271.4-TSAT)**0.375-0.00000038*TSAT**2.272
01480 QSPRHT=0.56*(TAVGEN-TSAT)+12.34*(1.0/TSAT**0.91-1./TAVGEN**.91)
01490+ *(MAX1(P, 75.0)-75.0)
01500 Q=QLAT+QSPRHT
01510 J=50
01520 WRITE(6,15) J,TSAT,TAVGEN,QLAT,QSPRHT
01530 15 FORMAT(15,4F15.4)
01540 RETURN
01550 END
run
77/05/25, 19.25, 51.
PROGRAM ABSORB

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J*\[t^2\] = 0.497
SRU = 2.131 UNITS.
RUN COMPLETE.
VI RESULTS AND CONCLUSIONS

One of the objectives of this project was to examine the Coefficient of Performance of this cycle and if possible, compare it with published data for the lithium bromide-water system. Performance data for a commercial machine can only be obtained from Trane Company as this is the only company in the U.S. that produces a two-stage water chiller. Their data bulletin (20) for the lithium bromide-water two stage chiller indicates a coefficient of performance of about 1.1 at rated conditions.

Figure 4 presents the results of a series of computer runs with this program for two absorber temperatures with a constant evaporator temperature of 40°F. The object was to examine the variation in COP with second stage maximum solution temperature, which would be closely associated with the temperature of the solar input fluid. Evidently this cycle has about the same performance at rated conditions as the commercial unit, using lithium bromide-water. (COP = 1.1) The evaporator temperature of 40°F was chosen as representative of a requirement to deliver chilled water at about 45°F. Small variations in this temperature would not be expected to change the computed results by a great deal. The absorber temperatures of 75°F and 85°F are representative of cooling water temperature available from a cooling tower. The increase in temperature from absorber to condenser is representative of a series water tower coolant flow where water is first sent through the absorber and then the condenser. This is sometimes done to increase the water temperature rise and reduce the size of the required cooling tower.

It must be noted that the cycle COP is only a part of the overall
FIGURE 4

TWO STAGE NOSCN-NH3
ABSORPTION SYSTEM COP

40°F EVAPORATOR TEMP.
CONDENSER 100 ABOVE ABSORBER

OVERALL CYCLE COEFFICIENT OF PERFORMANCE

75°F ABSORBER TEMP,

85°F ABSORBER TEMP

2ND STAGE GENERATOR SOLUTION OUTLET TEMP, °F
system which must be considered for solar cooling. In particular it is known that collector efficiency is reduced as the collection temperature is increased, so that any increases in absorption cycle COP with higher temperatures may be deceptive. The program lends itself well to modification for studies along those lines.

Comment should be made here on the pressure levels at which the ammonia system operates as the possibility of using air cooled condensers and absorbers is probably impractical because of the extreme pressures required in the second stage under these conditions. Ammonia systems operate at several atmospheres pressure while Lithium bromide systems operate at pressures below atmospheric. This necessitates the incorporation into the lithium bromide cycle of elaborate non-condensibles purge systems which are not necessary for ammonia. But higher temperatures in the ultimate coolant exaggerates this condition and would require pumping to well over 500 psi in the second stage if air cooling were used. The present study, which assumes the use of cooling tower water, shows second stage pressure levels of about 500 psi at representative rated conditions. The energy input to the pumps as well as the mechanical problems associated with achieving such high pressures are a significant consideration.

Ammonia is a Class 2 refrigerant, second only to sulphur dioxide as a hazard. Most building codes place many restrictions on its use: it cannot be used in air ducting, and must be used with an intermediate fluid; it cannot be placed in a living space unless in a specially constructed machinery room. For large installations, chilled water systems are common, so the use of an intermediate fluid is not a problem. But for residential cooling, it may be a significant consideration. And in most cases, it would be practical to place the absorption unit on a pad adjacent to the building as is done with
many residential units of the conventional type. This would obviate the necessity of constructing a machinery room.

Lithium bromide systems have two disadvantages not shared by the ammonia cycle. The first is an obvious limitation to operate above the freezing temperature of water. Ammonia can be used well below 0°F. The second involves the crystallization of lithium bromide solution if the water concentration is reduced below a well defined crystallization line. Most lithium bromide machines have features that either guard against crystallization or readily permit decrystallization should it occur. In either case, extra equipment is required. The sodium thiocyanate-ammonia cycle studied here operates well away from its crystallization line and should not require any decrystallization features.

A final comment should be made regarding the comparison made between the theoretical results of Figure 3 and the published data for an actual machine in Reference 20. It would be an appropriate extension of this work to replace the NaSCN-NH₃ relationships in the computer program with corresponding LiBr-H₂O data so that a comparison could be made on a theoretical-to-theoretical basis.
VII REFERENCES

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   ence of the International Solar Energy Society and Emphasizing
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(Describes studies of potential cycles and absorption pairs for solar cooling)

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University of Maryland, January 1976
(One of a series of reports which present the results of an ongoing study of solar cooling. It includes an extensive description of the two stage absorption cycle)

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G. Blythas and F. Daniels
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(This bulletin describes commercial single stage absorption chillers using lithium bromide-water and suitable for heat
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(This is the predecessor conference to reference 4)

(This bulletin describes their series of two-stage absorption water chillers)

21. International Critical Tables of Numerical Data, Physics, Chemistry and Technology
(A seven volume compilation of data, including thermophysical properties of many substances—e.g., ammonia)

22. "Concentrated Solutions in Liquid Ammonia"
Blytas, Kertesz and Daniels
Journal of the American Chemical Society Vol. 84 No. 7 April, 1962
(This is the companion paper to reference 17. It includes data for other salts and discusses the linearity of vapor pressure data on the logP 1/ Temp. plane)