A New Absorber Heat Recovery Cycle to Improve COP of Aqua-Ammonia Absorption Refrigeration System

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INTRODUCTION

Absorption refrigeration is, in principle, an attractive method for using heat energy directly for cooling purposes. The chief advantage the vapor absorption system enjoys over the vapor compression system is that the former mainly requires heat energy at moderate temperatures, while, in the latter, the energy input is shaft work, which is a high-grade energy. A relatively large amount of work is required, because, during compression, the vapor undergoes a large change in specific volume. On the other hand, in an absorption system, the working medium remains in a liquid phase during compression, there is very little change in the specific volume, and the work input is relatively small.

The absorption process in an absorption refrigeration system releases a large amount of heat. This heat is evolved at temperatures considerably above the ambient temperature, which results in a major irreversible loss in the absorber. This paper proposes a scheme to utilize this heat for improving the Coefficient of Performance (COP). Detailed analyses of the conventional system and the modified system are included to serve as a basis for the component design. The performance curves based on the above analyses are presented to show the effects of various operating parameters, such as the generator temperature, the absorber temperature, and the effectiveness of heat exchangers and the rectification column.

Literature Review

Extensive studies have been reported in the literature on the selection of refrigerant-absorbent combinations. The combination ammonia-water has been particularly attractive. Its performance is better than that of fluorocarbon refrigerants used in absorption systems, and it is free from the limitations imposed by the high freezing temperature of the refrigerant and low crystallization temperature of the solution, as in a water-lithium bromide system, or extreme corrosiveness, as in an ammonia-sodium thiocyanate system. The only disadvantage with the ammonia-water system is the volatile nature of water, used as the absorbent. However, this difficulty can be overcome by incorporating a properly designed rectification column.

The basic ammonia-water refrigeration system, shown in Fig. 1, consists of four major components: (1) generator, rectification column, and dephlegmator assembly; (2) condenser; (3) evaporator; and (4) absorber. Heat is supplied in the generator to raise high pressure vapors of ammonia and water. Water is removed from these vapors in the rectification column, and the necessary reflux is supplied by the dephlegmator, also known as reflux condenser. Ammonia

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vapors of high purity leaving the dephlegmator are condensed in the condenser, after which liquid ammonia is throttled through an expansion valve before entering the evaporator. In the evaporator, the refrigerating load evaporates liquid ammonia at low temperature. Ammonia vapors enter the absorber where they are absorbed in the weak solution stream coming from the generator. The heat energy released during the absorption process is removed from the absorber, and the pressure of strong solution is raised by a liquid solution pump. This high pressure strong solution returns to the generator and the cycle continues.

The basic cycle of Fig. 1, as described above, is not used in practice because of a low COP value owing to irreversible losses in the strong and weak solution streams between the generator and the absorber.

Fig. 2 shows a widely used conventional cycle, which has two additional components: a heat exchanger between the strong and weak solution streams and a precooler between liquid ammonia going to the evaporator and the vapors coming from it. The COP of the conventional cycle is improved considerably as compared to the basic cycle. Sometimes, the strong solution stream coming from the absorber is used to cool the dephlegmator, which may marginally improve the COP.

A few investigators have analyzed the performance of the conventional cycle. Whitlow has analyzed the basic and conventional cycles for two sets of operating conditions. The first set corresponded to an air-cooled operation, with \( T_C = T_A = 51.7^\circ C, T_E = 5.2^\circ C \), and the strong solution concentration, \( X_L \), varying from 0.1 to 0.35. The second set of operating conditions corresponded to a water-cooled operation, with \( T_C = T_A = 30.5^\circ C, T_E = 5.2^\circ C \), and \( X_L \) varying from 0.2 to 0.5.

Stoeker and Reed have computed the effects of different operating temperatures on the COP of the basic and conventional cycles. The calculations were performed by assuming a UA-value for the heat exchangers as 0.316 kW/°C. Also, a temperature rise from 7.2°C at the inlet to 12.8°C at the exit of the evaporator was considered to account for the carryover of small quantities of water vapor from the generator to the evaporator. The maximum COP value obtained was 0.42.

In a later paper, Whitlow calculated the COP values as functions of the generator temperature and the strong solution concentration by using P-T-X charts. Relative circulation rates (defined as the weak solution circulation rate per kilogram of refrigerant circulated) were also plotted against the generator temperature. Calculations were made for water-cooled and air-cooled systems working on basic as well as conventional cycles. Since no consideration was given to component efficiencies, the calculated COP values represent the maximum, or ideal, cycle performance.

A number of researchers have worked toward obtaining higher COP values. They suggested operating the system at optimum conditions and modifying the absorption cycle. Multistage absorption, multistage evaporation, or both have been suggested by Richter. He has shown that multistaging improves COP and also allows operation of the system at evaporator temperatures lower than those possible with single-stage operation. Phillips classified these high efficiency cycles as:

1. Double-effect distillation cycle, in which the distillation is carried out in two stages. It results in a high pressure of 7.6 MPa in the upper stage of the generator, which may be objectionable from a practical standpoint.

2. Double-evaporation cycle, in which evaporation is carried out at two different temperatures by means of a resorption loop. This cycle
needs considerable additional equipment and is recommended where
two-level refrigeration is required.

3. Absorber heat recovery cycle or absorber-generator heat exchange
cycle, in which the heat released during the absorption process is
used in the generator. The theoretical COP values of greater than
1.0 are possible with this cycle. It needs only a modification in
the absorber design, and the cost may be comparable to the conven-
tional system shown in Fig. 2.

Scope of the Present Work

Analyses of the absorption refrigeration cycles have been made by earlier
investigators for specific operating conditions. In this paper, a complete
analysis of the conventional cycle is presented by considering the individual
component efficiencies. The analysis is extended to a modified absorber heat
recovery cycle. Using this analysis, performance curves are obtained over a
range of operating temperatures for ideal cycles. Similar curves are generated
for practical systems by taking appropriate component efficiencies.

ANALYSIS OF A CONVENTIONAL CYCLE

The analysis of a conventional cycle (Fig. 2) is made under the following
assumptions.

1. The vapor leaving the dephlegmator (12) is saturated at the condenser
temperature.
2. The liquid leaving the condenser (1) is saturated at the condenser
temperature.
3. The strong solution leaving the absorber is saturated at the absorber
temperature.
4. The weak solution leaving the generator is saturated at the generator
temperature.
5. The temperature at the outlet of the evaporator is $\Delta T_E$ above the
temperature at the evaporator inlet.
6. The strong solution (8) is heated only up to the saturation temper-
ature, and no vapor generation takes place in the heat exchanger.

In an ideal cycle, the heat exchangers may have a maximum possible effec-
tiveness of 1.0. However, in a practical cycle, the effectiveness, $E$, of the
heat exchangers is below 1.0, and this fact is accounted for in the analysis.

Equilibrium state thermodynamic properties of ammonia-water solutions are
required for the analysis. The equations developed by Jain and Gable have been
used to determine these properties.\textsuperscript{13} Jain and Gable have used the data
published by the Institute of Gas Technology.\textsuperscript{14} Appx 1 summarizes these
property equations.

Given the generator, condenser, absorber, and evaporator temperatures,
along with the cooling load, the state point properties and the quantities of
heat supplied and rejected in the cycle are calculated as described below.

The high side pressure, $P_H'$, in the generator and condenser is calculated
from the refrigerant concentration in the condenser and the condensation
temperature:

$$P_H = P_H' (T_C' X_{RL})$$  \hspace{1cm} (1)

Refer to Appx 1 for equilibrium property equation.
The low side pressure, $P_L$, in the absorber and evaporator, is calculated from the given evaporator inlet temperature and the refrigerant concentration:

$$P_L = P_L \left( T_E, X_{RL} \right)$$  \hspace{1cm} (2)

The strong and the weak solution concentrations are then given by

$$X_{SL} = X_{SL} \left( T_A, P_L \right)$$  \hspace{1cm} (3)

and

$$X_{WL} = X_{WL} \left( T_G, P_H \right)$$  \hspace{1cm} (4)

**The Precooler--Evaporator Loop**

The enthalpies for vapor state and liquid state, $H_{12}$ and $H_1$, are calculated by the following equations:

for vapor state \hspace{1cm} $H_{12} = H_{12} \left( P_H, T_C \right)$  \hspace{1cm} (5)

for liquid state \hspace{1cm} $H_1 = H_1 \left( P_H, T_C \right)$  \hspace{1cm} (6)

In the evaporator, small quantities of water vapor are carried from the generator due to imperfect rectification. The overall concentration at state 4, $X_4$, is, however, very close to unity (above 0.99). The concentration of water in the liquid phase increases along the evaporator as more volatile ammonia is vaporized faster. This raises the saturation temperature of the evaporating liquid, and the temperature, $T_4$, at the exit of the evaporator is higher than the temperature, $T_3$, at the inlet.

It can be written

$$T_4 = T_3 + \Delta T_E$$  \hspace{1cm} (7)

The equilibrium liquid and vapor phase concentrations at the exit of the evaporator are given by

$$X_{4L} = X_{4L} \left( P_L', T_4 \right)$$  \hspace{1cm} (8)

and

$$X_{4V} = X_{4L} \left( P_L', T_4 \right)$$  \hspace{1cm} (9)

The ratio of the vapor phase flow rate, $G_{4V}$, to the liquid phase flow rate $G_{4L}$, is given by

$$\frac{G_{4V}}{G_{4L}} = \frac{X_R - X_{4L}}{X_{4V} - X_{4L}}$$  \hspace{1cm} (10)

Enthalpies of the liquid and the vapor phases at state 4 are calculated from respective saturation equations,

$$H_{4L} = H_{4L} \left( P_L', T_4 \right)$$  \hspace{1cm} (11)

$$H_{4L} = H_{4V} \left( P_L', T_4 \right)$$  \hspace{1cm} (12)

From Eqs 10, 11, and 12, the enthalpy at state 4 is given by

$$H_4 = H_{4L} + \frac{(X_R - X_{4L})}{(X_{4V} - X_{4L})} \cdot (H_{4V} - H_{4L})$$  \hspace{1cm} (13)

Using the definition of effectiveness $E_{PC}$ for the precooler, we get

$$T_2 = T_1 - (T_1 - T_4) E_{PC}$$  \hspace{1cm} (14)
The enthalpy of subcooled liquid at state 2 is calculated by the same equations as for saturated liquid at temperature $T_2$,

$$H_2 = H_2 \left( P_h, T_2 \right)$$

(15)

Further, for the throttling process from state 2 to state 3, we have

$$H_3 = H_2$$

(16)

A heat balance across the precooler yields

$$H_5 = H_4 + (H_1 - H_2)$$

(17)

The refrigerant flow rate per ton (1 ton $\approx 3500$ W) is calculated by a heat balance across the evaporator:

$$G_R = \frac{3500}{(H_4 - H_3)} \text{ kg/sec-ton}$$

(18)

Heat rejected in the condenser is given by

$$Q_C = G_R (H_{12} - H_1)$$

(19)

The Absorber--Generator Loop

This loop includes the absorber the heat exchanger, and the generator. As states 6, 9, and 5 are already defined, we can evaluate the following properties:

$$H_6 = H_6 \left( P_L, T_A \right)$$

(20)

$$H_9 = H_9 \left( P_H, T_G \right)$$

(21)

$$X_6 = X_{SL}$$

(22)

and

$$X_9 = X_{WL}$$

(23)

The mass flow rates of strong and weak solutions are calculated by making a heat balance over the absorber

$$G_S = G_R \left( X_R - X_w \right) / (X_S - X_w)$$

(24)

and

$$G_W = G_S - G_R$$

(25)

The enthalpy at state 8 is calculated for saturated liquid state

$$H_8 = H_8 \left( P_H, X_S \right)$$

(26)

Neglecting the pump work,

$$H_7 = H_6$$

(27)

A heat balance across the heat exchanger yields

$$H_{10} = H_9 - (H_8 - H_7) \frac{G_S}{G_W}$$

(28)

At this stage, a check on the heat exchanger effectiveness is carried out. An upper limit of 1.0 is set for ideal cycle, while it is specified by $E_{HE}$ for a practical cycle. If the actual effectiveness given by

$$E_{HE} = \frac{T_9 - T_{10}}{T_9 - T_7}$$

(29)
exceeds the upper limit, the foregoing calculations are corrected for the limiting heat exchanger effectiveness. Note that in Eq 29,

$$T_7 = T_A$$  \hspace{1cm} (30)$$

and $T_{10}$ is calculated from

$$H_{10} = H_{10} \left( P_H, T_{10} \right)$$  \hspace{1cm} (31)$$

since $H_{10}$ is known from Eq 28.

For the throttling process from state 10 to 11,

$$H_{11} = H_{10}$$  \hspace{1cm} (32)$$

Heat rejected in the absorber is calculated by making a heat balance over the absorber

$$Q_A = G_w H_{11} + G_R H_5 - G_5 H_6$$  \hspace{1cm} (33)$$

The Generator--Deephlegmator Loop

This loop is represented on a P-X diagram in Fig. 3. The strong solution enters the generator at state 8. Its equilibrium vapor phase composition $X_{8V}$ is given by

$$X_{8V} = X_{8V} \left( X_{8L}, P_H \right)$$  \hspace{1cm} (34)$$

The extension of the line joining 8 and 8V in Fig. 3 intersects the line $X = X_R^*$. This gives the minimum operating reflux enthalpy $H_{MIN}$ as

$$H_{MIN} = [(X_R - X_{8L}) / (X_{8V} - X_{8L})] (H_{8V} - H_{8L}) + H_{8L}$$  \hspace{1cm} (35)$$

The minimum reflux $R_{MIN}$ in the deephlegmator is given by

$$R_{MIN} = \frac{H_{MIN} - H_{12}}{H_{12} - H_1}$$  \hspace{1cm} (36)$$

To limit the number of plates, or the size of the rectification column, to practical values, a reflux ratio, $F$ (defined as the ratio of the operating reflux to the minimum reflux), of 2 to 3 is commonly employed. Thus the operating reflux enthalpy, $H_{OPER}$, is greater than $H_{MIN}$ and is given by

$$H_{OPER} = H_{12} + F \cdot R_{MIN} \cdot (H_{MIN} - H_{12})$$  \hspace{1cm} (37)$$

The heat rejected in the deephlegmator is given by

$$Q_D = G_R (H_{OPER} - H_{12}) \text{ W/ton}$$  \hspace{1cm} (38)$$

From the graphical construction shown in Fig. 3, $H_{POLE}$ is given by

$$H_{POLE} = H_{OPER} \cdot \frac{(X_S - X_W)}{(X_R - X_S)} \cdot (H_{OPER} - H_8)$$  \hspace{1cm} (39)$$

and the heat input to the generator is given by

$$Q_G = G_w (H_9 - H_{POLE}) \text{ W/ton}$$  \hspace{1cm} (40)$$

COP and Overall Heat Balance

The COP of the system is given by

$$\text{COP} = \frac{Q_E}{Q_G}$$  \hspace{1cm} (41)$$

A final check is made by performing an overall energy balance

$$Q_G + Q_E = Q_C + Q_A + Q_D + \text{Error}$$  \hspace{1cm} (42)$$
The error term as determined from Eq 42 should be close to zero.

Results for a Conventional Cycle

The performance of a conventional cycle under ideal and practical conditions is computed from the above analysis by a computer. The following values are used in the computations:

- Maximum possible effectiveness of heat exchangers = 1.0 for ideal cycle
- Reflux ratio in the dephlegmator = 0.8 for practical cycle
- Temperature rise in the evaporator = 2 for practical cycle
- Range of generator temperatures: 90 - 180°C
- Range of absorber and condenser temperatures: 30 - 50°C
- Evaporator temperature: 0°C

The variation of COP with the generator temperature $T_G$ is depicted in Fig. 4. Two values of absorber and condenser temperatures, 30°C and 45°C, are selected to represent the water-cooled and the air-cooled systems, respectively. COP for an ideal as well as for a practical cycle, in general, increases with the generator temperature up to a maximum value and then decreases slowly. This is because at low generator temperatures, the strong solution (state 8) entering the generator is subcooled, and increasing generator temperature increases $T_B$ thereby reducing $Q_G$. However, the upper limit for state 8 is taken as saturation condition corresponding to the strong solution concentration, and no vapor is generated in the heat exchanger. This limits the heat transferred to the strong solution, and the COP reduces with a further increase in the generator temperature.

Two sets of graphs are shown in Fig. 4 for water-cooled and air-cooled systems. For water-cooled systems, a maximum COP value of 0.67 is observed at $T_G = 90°C$ for ideal cycle, while it is 0.65 for practical cycle at $T_G = 97°C$. The difference in the ideal and practical curves is due to the difference in the individual component performance of the dephlegmator and heat exchanger. For air-cooled systems, a maximum COP value of 0.67 is observed at $T_G = 108°C$ for ideal cycle, while for the practical cycle, a value of 0.505 is obtained for $T_G$ between 135°C and 145°C.

There is a minimum value of $T_G$ for a given set of operating conditions or system parameters, below which a system cannot function. At lower values of $T_G$, the difference between the concentrations of weak and strong solution becomes small, leading to higher circulation rates of the solution streams per unit mass flow rate of the refrigerant. Relative circulation, $R_C$, defined as

$$\frac{R_C}{C_R} = \frac{G_W}{G_R} = \frac{X_R - X_S}{X_S - X_W}$$

increases sharply owing to a small difference between $X_S$ and $X_W$. The variation of $R_C$ with $T_G$ is shown in Fig. 5. The relative circulation is lower for water-cooled systems due to lower $X_W$ values. Higher values of relative circulation combined with low values of heat exchanger effectiveness make the operation of the system inefficient. For increasing values of $T_G$ above 120°C, the relative circulation becomes smaller, offsetting the increased heat exchanger losses. This results in virtually a flat curve of COP for higher values of $T_G$ (Fig. 4).
A MODIFIED ABSORBER HEAT RECOVERY CYCLE

A modified absorber heat recovery cycle can be explained with the help of an ln p-1/T diagram shown in Fig. 6. The conventional cycle is represented by lines a-b, c-d-e-f in Fig. 6. Point a represents ammonia vapors coming from the generator and also condensed liquid leaving the condenser at a temperature, \( T_C \), and a pressure, \( P_H \), corresponding to saturation conditions. Liquid ammonia at state a is then throttled to state b, corresponding to lower pressure, \( P_L \), and evaporator temperature, \( T_E \). Point b also represents vapors leaving the evaporator. Saturated weak solution corresponding to point f absorbs ammonia vapors (point b) and the resulting saturated strong solution at pressure \( P_L' \), represented by state c, is pumped to a higher pressure and heated in the heat exchanger to point d. In the generator, the strong solution at d is separated into ammonia vapors (point a) and saturated weak solution (point e).

In the heat exchanger, the strong solution is heated from c to d (Fig. 6), while the weak solution is cooled from e to f. Heat evolved during the absorption process along f'-c is, however, wasted in this cycle. As the generator temperature is raised from \( T_G \) to \( T_G' \), point e would shift to e' and f to f'. Now, the heat evolved during the absorption process along f'c is at a considerably higher temperature, up to \( T_f' \). Heat released along f'-c can be used to heat the strong solution along c-d.

Further heating would result in vapor generation in the heat exchanger along d-d' while cooling the weak solution along e'-f'. The heat exchanger now acts partially as generator, thereby reducing the generator heat requirement and, consequently, improving the COP.

Fig. 7 shows a schematic arrangement of the modified cycle. The absorber is modified to transfer the absorption heat to strong solution from 7 to 7A in the absorber heat exchanger. An externally cooled absorber portion is included to maintain the necessary temperature differences between the two streams in the absorber heat exchanger.

Analysis of the Modified Absorber Heat Recovery Cycle

Assumptions 1 through 5 made in the analysis of the conventional cycle are also applicable here. Assumption 6 is modified as follows:

The strong solution leaving the absorber (point 7) is heated up to saturation condition in the absorber heat exchanger. Further heating in the heat exchanger may result in vapor generation.

The analysis of the modified absorber heat recovery cycle differs only in the absorber-generator loop.

Absorber-Generator Loop

The absorber is analyzed in two parts—the absorber heat exchanger, and the externally cooled absorber. In the absorber heat exchanger (Fig. 7), weak solution (1l), which is cooled up to saturation condition in the heat exchanger, is sprayed to absorb ammonia vapors at state 5. Ammonia-water solution at intermediate concentration leaves the absorber heat exchanger at 11A, along with the equilibrium vapors at 5'. A heat balance over the absorber heat exchanger yields

\[ G_S (H_{7A} - H_7) = G_W H_{1l} + G_R H_5 - G_{1lA} H_{1lA} - G_5 H_5' \]

A mass balance on the absorption side gives

\[ G_W + G_5 = G_{1lA} + G_5', \]  

(45)
From Eqs. 44 and 45, we get

\[
G_{11A} = \frac{G_5 (H_{7A} - H_7) + C_W (H_5, - H_{11}) - G_R (H_5 - H_5,)}{H_5, - H_{11A}}
\]  \( \text{(44)} \)

To account for the effectiveness of the absorber heat exchanger, it is assumed that

\[
T_{7A} = T_{11} - S
\]  \( \text{(47)} \)

where

\[
S \text{ is called 'approach', and practical values of 10 or 12°C may be assumed. } H_{7A} \text{ is then calculated by}
\]

\[
H_{7A} = H_7 + C_{PS} (T_{7A} - T_7)
\]  \( \text{(48)} \)

where

\[
C_{PS} \text{ is specific heat of the strong solution. A mass balance for ammonia fraction yields}
\]

\[
G_W X_{11} + G_R X_5 = G_{11A} X_{11A} + G_5, X_5
\]  \( \text{(49)} \)

As 5' and 11A are in equilibrium, we have

\[
T_{5'} = T_{11A}
\]  \( \text{(50)} \)

In Eqs. 44, 45, and 49, exit conditions at 11A and 5' are unknown. However, being at saturated condition, the enthalpies and concentrations are uniquely related to the pressure and temperature through property equations. The following iterative procedure may be used to solve the above equations:

1. Assume \( X_{11A} = X_5 + 0.05 \)
2. For saturated liquid condition at \( P_L \) and \( X_{11A} \), evaluate \( T_{11A} \) and \( H_{11A} \) using property equations.
3. From Eq. 50, \( T_{5'} = T_{11A} \). For saturated vapor condition at \( P_L \) and \( T_{5'}, \) evaluate \( X_5', \) and \( H_5' \).
4. Using Eq 46, calculate \( G_{11A} \).
5. Using Eq 45, calculate \( G_5', \).
6. Using Eq 49, obtain a new value of \( X_{11A} \).
7. Compare the new value of \( X_{11A} \) from step 6 with the assumed value before step 2. If the difference is larger than a set error limit of, say, 0.0001, recalculate from step 2 onwards until the error limit is satisfied.

In the externally cooled absorber, heat rejected \( Q_A \) is given by

\[
Q_A = G_{11A} H_{11A} + G_5, H_5, - G_S H_6
\]  \( \text{(51)} \)

The heat exchanger effectiveness is given by

\[
E_{HE} = \frac{(T_9 - T_{10})}{T_9 - T_{7A}}
\]  \( \text{(52)} \)

If the effectiveness is larger than the actual value, \( T_{10} \) is recalculated from Eq 52 using the actual effectiveness value, and the absorber analysis has to be repeated. Enthalpy at state 8 is calculated by a heat balance over the heat exchanger.
\[ H_8 = H_{7A} + \frac{G_W}{G_S} \cdot (H_9 - H_{10}) \]  \( (53) \)

State 8 lies in the two-phase region. Knowing \( H_8 \) and \( P_H \), then \( X_{8L}, X_{8V}, \)
\( H_{8L}, \) and \( H_{8V} \) can be calculated from property equations using an iterative
procedure. Further procedure is similar to that described for the conventional
cycle.

RESULTS

The performance of an ideal and a practical absorber heat recovery cycle is
computed from the above analysis using a computer. The following values are
used in the computations.

- Maximum possible effectiveness of heat exchangers
  - 1.0 for ideal cycle
  - 0.8 for practical cycle

- Reflux ratio in the dephlegmator
  - 1.0 for ideal cycle
  - 2.0 for practical cycle

- Temperature rise in the evaporator
  - 5°C

- Range of generator temperatures
  - 90 to 180°C

- Range of absorber and condenser temperatures
  - 30 to 50°C

- Evaporator temperature
  - 0°C

The variation of COP with generator temperature is shown in Fig. 8. Two
values for 30°C and 45°C for absorber and condenser temperatures are selected
to represent water-cooled and air-cooled systems. The general nature of the
curves is similar to those in Fig. 4 for a conventional cycle. However, there
is a considerable improvement in the COP values. The maximum values of COP are
0.705 and 0.555 for water-cooled and air-cooled practical absorber heat
recovery cycles, respectively. The maximum COP value occurs at generator
temperatures between 110 and 120°C for a water-cooled practical cycle, while
for an air-cooled practical cycle, the maximum COP value is obtained for \( T_G \)
between 140°C and 170°C. Fig. 5 is applicable to the absorber heat recovery
cycle as well, because the relative circulation depends only on the concentra-
tions that are decided by the operating temperatures.

Effect of Condenser and Absorber Temperatures

The influence of the heat rejection temperature on performance is studied
by varying the absorber and the condenser temperatures. It is clear from
erlier discussion that for air-cooled systems with their higher heat rejection
temperature, the COP values are lower than those corresponding to water-cooled
systems. There is also an upper temperature limit for heat rejection above
which the system cannot function for a given generator temperature.

Fig. 9 shows the variation of COP with the absorber and condenser
temperatures for a \( T_G \) of 120°C. Solid lines represent a practical absorber
heat recovery cycle, and dashed lines represent a practical conventional cycle.
Two sets of curves are plotted. One set represents COP versus absorber
temperature for a condenser temperature of 35°C. The other set represents COP
versus condenser temperature for an absorber temperature of 35°C. It can
be seen that COP is more sensitive to absorber temperature, and the effect of condenser temperature is quite small. Similar results for generator temperature of 150° are shown in Fig. 10. The effect of varying condenser temperature is rather strange in this case. COP increases with increasing condenser temperature for both the cycles before it levels off and starts dropping again. This happens because of the peculiar property relationships for ammonia-water solutions. One has to be careful in selecting the operating temperatures, as lower heat rejection temperatures or higher generator temperatures do not necessarily give higher COP values.

Solar Cooling Possibilities

Since an absorption refrigeration system can operate with moderately low grade thermal energy, solar energy can be used effectively for cooling purposes. The cost effectiveness of such a system is generally poor in comparison to that of vapor compression systems. Any improvement in the COP can bring about a considerable reduction in the cost of the system by the decreased collector area requirements, since the cost of collectors is a major portion of the overall system cost.

With the modified absorber heat recovery cycle, there are two major advantages. First, an improvement in COP of the order of 10% can be realized. For a typical system of (100 tons) operating for (3000 hours per year), it would amount to a saving of (350 tons) of steam or approximately $3500 per year. In terms of solar collector area, the reduction could be of the order of 100 m².

The second advantage of the modified absorber heat recovery cycle is the possibility of using air-cooled systems at COP values comparable to those for water-cooled conventional systems. With air cooling, the cost of a cooling water loop is eliminated, along with the auxiliary power consumption for water circulation. Heat rejected from the absorber to atmosphere is greatly reduced, as this occurs only through the lower, externally cooled absorber portion.

CONCLUSIONS

In the present paper, a detailed methodology is presented to analyze the conventional ammonia-water absorption refrigeration cycle. A new modified absorber heat recovery cycle, which utilizes the heat released during absorption process, is presented and also analyzed. On the basis of the analyses, results are computed for ideal as well as practical systems by taking representative values of the system-parameters.

From the results, it is observed that an improvement of about 10 percent in COP is obtained with the modified absorber heat recovery cycle, as compared to the conventional cycle. The possibilities of using solar energy and also incorporating air-cooled condenser and absorber are greatly improved. Further, the effects of operating variables, like condenser temperature, are not straightforward. It is recommended that the detailed analysis be used in conjunction with other coupled system-parameters to arrive at optimum design conditions.

REFERENCES


APPENDIX

Property equations for aqua-ammonia mixtures at equilibrium conditions developed by Jain and Gable have been used in this paper. These equations are in the following form:

\[ T = F_1 \left(X_L, P\right) \]  
\[ H_L = F_2 \left(X_L, P\right) \]  
\[ X_V = F_3 \left(l_L, P\right) \]  
\[ H_V = F_4 \left(X_V, T\right) \]

In order to obtain greater accuracy, two sets of polynomial equations were developed for high-pressure range (17-24 bar) and for low-pressure range (3.5-5.5 bar).

For calculating any property on the right-hand side of the above equations, an iterative scheme is followed. For example, given \( H_L \) and \( P \) in Eq 2, \( X_L \) is found by an iterative scheme using the Newton-Raphson method. For actual equations, see Ref 13.
NOMENCLATURE

C_p  - Specific heat, kJ/kg-K
E    - Effectiveness of heat exchanger
F    - Reflux ration
G    - Mass flow rate, Kg/S-ton
H    - Enthalpy, kJ/kg
P    - Pressure, N/m²
Q    - Heat quantity, kJ
R    - Reflux
RC   - Recirculation ratio
T    - Temperature, K
X    - Concentration of ammonia in ammonia-water solution

SUBSCRIPTS

A    - Absorber
C    - Condenser
D    - Dephlegmator
HE   - Heat exchanger
L    - Liquid
MIN  - Minimum
OPER - Operating
R    - Refrigerant
S    - Strong ammonia solution
V    - Vapor
W    - Weak ammonia solution
Figure 1. Basic ammonia water absorption refrigeration system

Figure 2. Conventional ammonia-water absorption refrigeration system
Figure 3. Generator – dephlegmator loop representation on a H-X diagram (state points corresponding to Fig. 2)

Figure 4. Effect of generator temperature on COP for conventional absorption refrigeration system
Figure 5. Effect of operating temperatures on relative circulation

Figure 6. Absorber heat recovery cycle representation on a \( \ln p - 1/T \) diagram
Figure 7. Modified aqua-ammonia absorption refrigeration system using absorber heat recovery cycle.

COP vs. $T_C$ for $T_A = 35^\circ C$

$T_g = 150^\circ C$

$T_e = 0^\circ C$

COP vs. $T_A$ for $T_C = 35^\circ C$

Figure 8. Effect of generator temperature on COP for modified absorber heat recovery cycle.
Figure 9. Effect of condenser and absorber temperature on COP for $T_G = 120^\circ C$, $T_E = 0^\circ C$

Figure 10. Effect of condenser and absorber temperature on COP for $T_G = 150^\circ C$, $T_E = 0^\circ C$