Solar Absorption Refrigeration System Using New Working Fluid Pairs

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Abstract- Absorption refrigeration systems powered by solar energy increasingly attract research interests in the last years. In this study, thermodynamic analyses for different working fluid pairs are performed. A computer simulation model has been developed to predict the performance of solar absorption refrigeration system using different working fluid. The model is based on detailed mass and energy balance and heat and mass transfer for the cycle component. Detailed thermodynamic properties for ammonia- water, ammonia-lithium nitrate and ammonia-sodium thiocyanate are expressed in polynomial equations and used in cycle simulation. The performances of these three cycles against various generator, evaporator, and condenser temperatures are compared. The results show that the ammonia-lithium nitrate and ammonia-sodium thiocyanate cycles give better performance than the ammonia-water cycle. The ammonia-sodium thiocyanate cycle cannot operate at evaporator temperatures below -10°C for the possibility of crystallization. Increasing condenser temperatures cause a decrease in system performance for each cycle. With the increase in evaporator temperature, the COP values for each cycle increase. These results can serve as a source of reference for developing new cycles and searching for new working fluids pairs. They can also be used in selecting operating conditions for existing systems and achieving automatic control for maintain optimum operation of the system.

Keywords- performance; absorption; solar energy; NH_3-$LiNO_3$; crystallization; refrigeration; generator

I. INTRODUCTION

DURING the last few decades, an increasing interest, based on research and development, has been concentrated on utilization of non-conventional energy sources, namely solar energy, wind energy, tidal waves, biogas, geothermal energy, hydropower, hydrogen energy, etc. Among these sources, solar energy, which is an energy source for cooling applications, is a highly popular source due to the following facts: direct and easy usability, renewable and continuity, maintaining the same quality, being safe, being free, being environment friendly and not being under the monopoly of anyone.

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The absorption refrigeration system, which has some advantages, such as silent operation, high reliability, long service life, simpler capacity control mechanism, easier implementation, and low maintenance, is widely acknowledged as a prospective candidate for efficient and economic use of solar energy for cooling applications. Also, the absorption refrigeration cycle is usually a preferable alternative, since it uses the thermal energy collected from the sun without the need to convert this energy into mechanical energy as required by the vapor compression cycle. In addition, the absorption cycle uses thermal energy at a lower temperature than that dictated by the vapor compression cycle. The binary systems of NH_3-H_2O and LiBr-H_2O were well known as working fluid pairs to be applied both in absorption heat pumps and in absorption refrigerators currently. Theoretical and experimental studies have been conducted to optimize the performance of absorption refrigeration cycles using NH_3-H_2O and LiBr-H_2O as refrigerant- absorbent combination.

The advantage for refrigerant NH_3 is that it can evaporate at lower temperatures (i.e. from -10 to 0°C) compared to H_2O (i.e. from 4 to 10°C). Therefore, for refrigeration, the NH_3-H_2O cycle is used. Research has been performed for NH_3-H_2O systems theoretically [1-4] and experimentally [5, 6]. These studies show that the NH_3-H_2O system exhibits a relatively low COP. Efforts are being made to search for better working fluid pairs that can improve system performance. It is proposed that NH_3-$LiNO_3$ and NH_3-NaSCN cycles can be alternatives to NH_3-H_2O systems [7, 8].Therefore, in the present study, the comparisons of the performances of NH_3-H_2O, NH_3-$LiNO_3$ and NH_3-NaSCN absorption cycles driven by solar energy are performed. It is hoped that these results could serve as a source of reference for designing and selecting new absorption refrigeration systems, developing new working fluid pairs and optimizing suitable operating conditions.

II. CYCLE PERFORMANCE ANALYSIS

The absorption cycle powered by solar energy is illustrated in Fig. 1. Low-pressure refrigerant vapor from the evaporator is absorbed by the liquid strong solution in the absorber. The pump receives low-pressure liquid weak solution from the absorber, elevates the pressure of the weak solution and delivers it to the generator. By weak solution (strong solution) is meant that the ability of the solution to absorb the refrigerant vapor is weak (strong) according to ASHRAE definition [9]. In the generator, heat from a high-temperature source by solar energy drives off the refrigerant vapor in the weak solution. The liquid strong solution returns to the absorber through a throttling valve whose purpose is to provide a pressure drop to maintain the pressure difference between the generator and the absorber.

The high-pressure refrigerant vapor condenses into liquid in the condenser and enters the evaporator through a throttling valve, maintaining the pressure difference between the condenser and the evaporator. In order to improve cycle performance, a solution heat exchanger is normally added to the cycle, as shown in Fig. 1. The cycle performance is measured by the coefficient of performance (COP), which is defined as the refrigeration rate over the rate of heat addition at the generator plus the work input to the pump, that is

$$COP = \frac{Q_{evp}}{Q_{gen} + W_{me}} \tag{1}$$

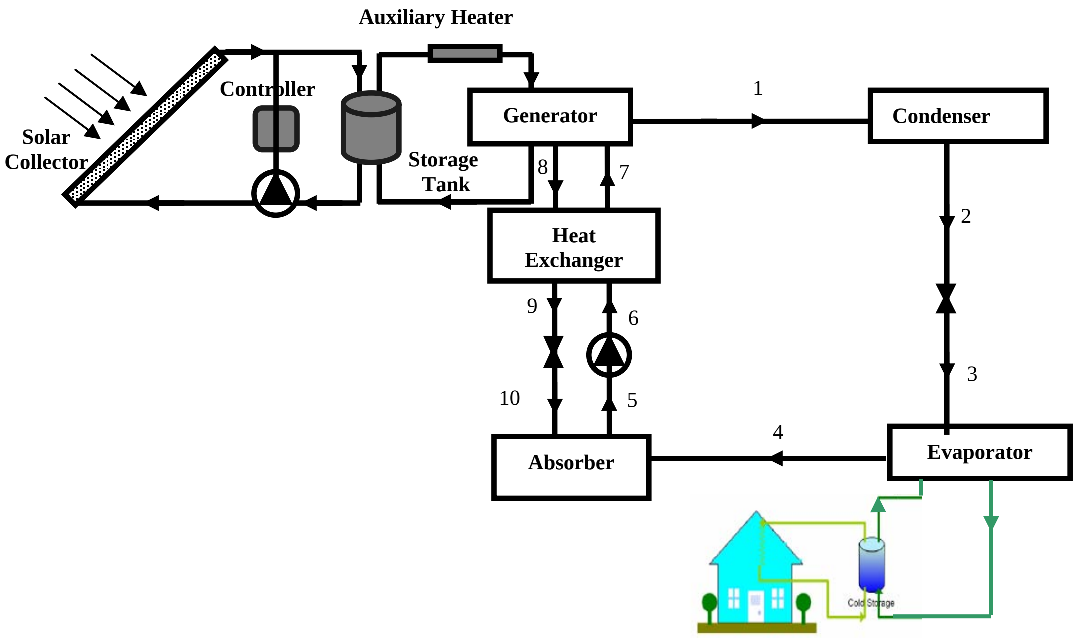

Fig. 1. The schematic illustration of the solar absorption refrigeration system

In order to use equation (1), mass and energy conservation should be determined at each component. For the generator, the mass and energy balances yield:

$$m_7 = m_1 + m_8 \tag{2}$$

$$m_7 X_7 = m_1 X_1 + m_8 X_8 \tag{3}$$

$$Q_{gen} = m_1 h_1 + m_8 h_8 - m_7 h_7 \tag{4}$$

From equations (2) and (3), the flow rates of the strong and weak solutions can be determined:

$$m_7 = \frac{1 - X_8}{X_7 - X_8} m_1 \tag{5}$$

$$m_8 = \frac{1 - X_7}{X_7 - X_8} m_1 \tag{6}$$

The mass flow ratio of the system, circulation ratio, is defined as the mass flow rate of solution from the absorber to the generator to the mass flow rate of working fluid (refrigerant), that

$$CR = \frac{m_7}{m_1} \tag{7}$$

The energy balance for the solution heat exchanger is as follows:

$$T_9 = E_{ex} T_6 + (1 - E_{ex}) T_8 \tag{8}$$

$$h_7 = h_6 + \frac{m_8}{m_6}(h_8 - h_9) \tag{9}$$

The energy increase by pumping is

$$h_6 = h_5 + (P_6 - P_5) v_6 \tag{10}$$

$$W_{me} = (P_6 - P_5) v_6 \tag{11}$$

Finally, energy balances for the absorber, condenser and evaporator yield

$$Q_{abs} = m_4 h_4 + m_{10} h_{10} - m_5 h_5 \tag{12}$$

$$Q_{cond} = m_1 (h_1 - h_2) \tag{13}$$

$$Q_{evp} = m_1 (h_4 - h_3) \tag{14}$$

III. SOLUTION PROPERTIES

The thermodynamic properties for NH_3-H_2O, NH_3-$LiNO_3$ and NH_3-NaSCN solutions are pressure, temperature, concentration, enthalpy and density, these properties are interdependent and are necessary for computer simulation of absorption refrigeration systems. For NH_3-H_2O, NH_3-$LiNO_3$ and NH_3-NaSCN absorption refrigeration cycles, NH_3 is the refrigerant, H_2O, $LiNO_3$ and NaSCN are absorbents. The thermodynamic properties at outlet of generator to inlet of absorber in Fig. 1 are determined by NH_3, and other properties can be calculated based on the binary mixture of NH_3-H_2O, NH_3-$LiNO_3$ or NH_3-NaSCN solutions.

3.1 Refrigerant NH_3

In the usual ranges of pressure and temperature concerning refrigeration applications, the two phase equilibrium pressure and temperature of the refrigerant NH_3 are linked by the relation:

$$P(T) = 10^3 \sum_{i=0}^{6} a_i (T - 273.15)^i \qquad (15)$$

The specific enthalpies of saturated liquid and vapor NH_3 are expressed in terms of temperature as follows:

$$h_l(T) = \sum_{i=0}^{6} b_i (T - 273.15)^i \qquad (16)$$

$$h_v(T) = \sum_{i=0}^{6} c_i (T - 273.15)^i \qquad (17)$$

3.2 NH_3-H_2O solution

The relation between saturation pressure and temperature of an ammonia-water mixture is given as [10]:

$$LogP = A - \frac{B}{T} \qquad (18)$$

where

$$A = 7.44 - 1.767X + 0.982X^2 + 0.362X^3 \qquad (19)$$

$$B = 2013.8 - 2155.7X + 1540.9X^2 - 194.7X^3 \qquad (20)$$

The relation among temperature, concentration and enthalpy is as follows;

$$h(T, \overline{X}) = 100 \sum_{i=1}^{16} a_i \left(\frac{T}{273.16} - 1\right)^{mi} \overline{X}^{ni} \qquad (21)$$

where $\overline{X}$ is the ammonia mole fraction and is given as follows:

$$\overline{X} = \frac{18.015X}{18.015X + 17.03(1 - X)} \qquad (22)$$

The relation among specific volume, temperature and concentration is given as;

$$v(T, X) = \sum_{j=0}^{3} \sum_{i=0}^{3} a_{ij} (T - 273.15)^i X^j \qquad (23)$$

3.3 NH_3-$LiNO_3$ solution

The relation between saturation pressure and temperature of an ammonia-lithium nitrate mixture is given as [7]:

$$LnP = A + \frac{B}{T} \qquad (24)$$

where

$$A = 16.29 + 3.859(1 - X)^3 \qquad (25)$$

$$B = -2802 - 4192(1 - X)^3 \qquad (26)$$

The relation among temperature, concentration and enthalpy is as follows [7]:

$$h(T, X) = A + B(T - 273.15) + C(T - 273.15)^2 + D(T - 273.15)^3 \qquad (27)$$

where

$$A = -215 + 1570(0.54 - X)^2 \quad \text{if } X \le 0.54 \qquad (28)$$

$$A = -215 + 689(X - 0.54)^2 \quad \text{if } X \ge 0.54 \qquad (29)$$

$$B = 1.15125 + 3.3826X \qquad (30)$$

$$C = 10^{-3}(1.099 + 2.3965X) \qquad (31)$$

$$D = 10^{-5}(3.93333X) \qquad (32)$$

The solution density is related to concentration and temperature as [7]:

$$\rho(T, X) = 2046.22 - 1409.65X^{0.5} - 1.3463 (T - 273.15) - 0.0039(T - 273.15)^2 \qquad (33)$$

3.4 NH_3-NaSCN solution

The relation between saturation and temperature of an ammonia-sodium thiocyanate mixture is given as [7]:

$$LnP = A + \frac{B}{T} \qquad (34)$$

where

$$A = 15.7266 - 0.298629X \qquad (35)$$

$$B = -2548.65 - 2621.92(1 - X)^3 \qquad (36)$$

The relation among temperature, concentration and enthalpy is as follows [7]:

$$h(T, X) = A + B(T - 273.15) + C(T - 273.15)^2 + D(T - 273.15)^3 \qquad (37)$$

where

$$A = 79.72 - 1072X + 1287.9X^2 - 295.67X^3 \qquad (38)$$

$$B = 2.4081 - 2.2814X + 7.9291X^2 - 3.5137X^3 \qquad (39)$$

$$C = 10^{-2}(1.255 - 4X^2 + 3.06X^3) \qquad (40)$$

$$D = 10^{-5}(-3.33X + 10X^2 - 3.33X^3) \qquad (41)$$

The solution density is related to concentration and temperature as [7]:

$$\rho(T, X) = A + B(T - 273.15) + C(T - 273.15)^2 \qquad (42)$$

where

$$A = 1707.519 - 2400.4348X + 2256.5083X^2 - 930.0637X^3 \qquad (43)$$

$$B = -3.6341X + 5.4552X^2 - 3.1674X^3 \qquad (44)$$

$$C = 10^{-3}(5.1X - 3.6X^2 - 5.4X^3) \quad (45)$$

All coefficients of equations are listed by Sun [11].

IV. RESULTS AND DISCUSSION

In order to provide details optimum operating conditions for solar absorption refrigeration systems, the computer program was used to search for different operation conditions with which an absorption cycle reaches its maximum performance. Table 1 shows the comparison of the various thermodynamic states in the cycle operating at T_{gen} = 100°C, T_{cond} = 30°C, T_{abs} = 25°C and T_{evp} = -5°C, with the effectiveness of the solution heat exchanger of 80%.

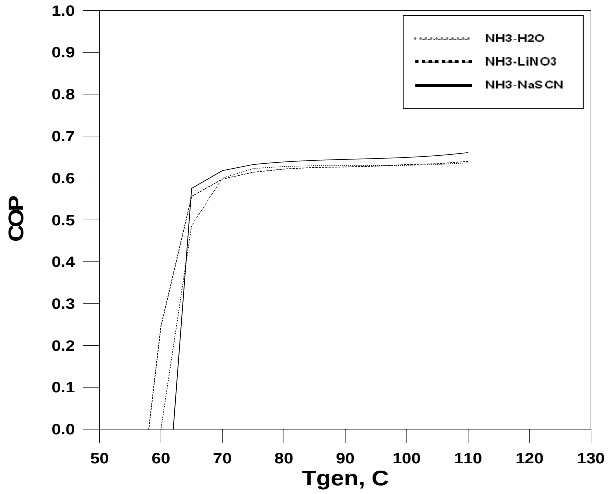

Fig. 2. Variation of COP with generator temperature

The total solution amounts circulated are 3.5, 3.95 and 5.07 kg/min for NH_3-H_2O, NH_3-$LiNO_3$ and NH_3-NaSCN respectively. This means that more refrigerant can be boiled off in the generator for the NH_3-H_2O cycle than for the other two.

Table 1.Thermodynamic properties at various states in absorption cycles driven by solar energy

Fluid state	T, °C	P, Kpa	X%	m, Kg/min
NH_3-H_2O cycle				
Generator ref exit	100	1167	100	1
Condenser ref exit	30	1167	100	1
Evaporator ref exit	-5	354	100	1
Absorber sol. exit	25	354	52.2	3.5
Generator sol inlet	66	1167	52.2	3.5
Generator sol exit	100	1167	33.5	2.51
Absorber sol inlet	40.8	354	33.5	2.51
NH_3-$LiNO_3$ cycle				
Generator ref exit	100	1167	100	1
Condenser ref exit	30	1167	100	1
Evaporator ref exit	-5	354	100	1
Absorber sol. exit	25	354	52.8	3.95
Generator sol inlet	64.5	1167	52.8	3.95
Generator sol exit	100	1167	37.5	3.01
Absorber sol inlet	40.8	354	37.5	3.01
NH_3-NaSCN cycle				
Generator ref exit	100	1167	100	1
Condenser ref exit	30	1167	100	1
Evaporator ref exit	-5	354	100	1
Absorber sol. exit	25	354	48.7	5.07
Generator sol inlet	68.4	1167	48.7	5.07
Generator sol exit	100	1167	36.8	4.1
Absorber sol inlet	40.8	354	36.8	4.1

As a result, a bigger pump is needed for the NH_3-NaSCN cycle. Fig. 2 shows the comparison of COP values vs generator temperatures for NH_3-H_2O, NH_3-$LiNO_3$ and NH_3-NaSCN absorption cycles. The COP values for these three cycles increase with generator temperatures. For the NH_3-$LiNO_3$ cycle a lower generator temperature can be used than for the others. It is shown that, for generator temperatures higher than 80°C, the NH_3-NaSCN cycle gives the best performance, and the NH_3-H_2O cycle has the lowest COP.

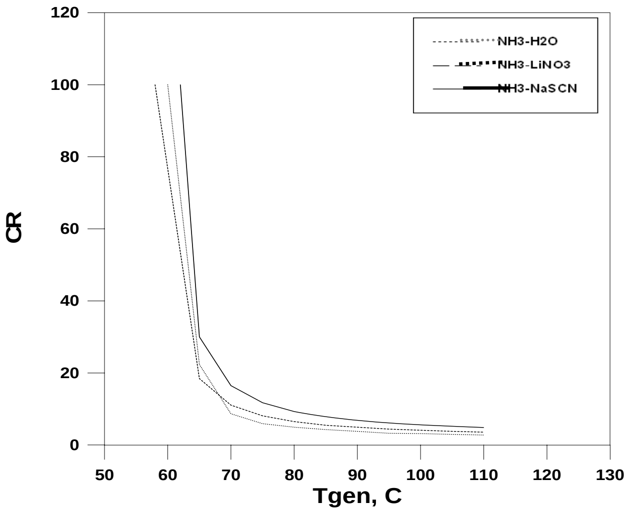

Fig. 3. Variation of CR with generator temperature

However, the differences among them are not very remarkable. Fig.3 shows the corresponding comparison of circulation ratios vs generator temperatures. It is illustrated that the circulation ratio for the NH_3-NaSCN cycle is higher than for the other two cycles. This means that either the solution pump needs to run faster or a bigger pump is required.

Fig. 4 gives the comparison of COP values vs evaporator temperatures for NH_3-H_2O, NH_3-$LiNO_3$ and NH_3-NaSCN absorption cycles. With the increase in evaporator temperature, the COP values for each cycle increase. For evaporator temperatures lower than zero, which is the temperature range for refrigeration, the NH_3-NaSCN cycle gives the best performance, and the NH_3-H_2O cycle has the lowest COP values. For high evaporator temperature, the performance of the NH_3-H_2O cycle is better than that of the NH_3-$LiNO_3$ cycle. The corresponding comparison of circulation ratios vs evaporator temperatures is given in Fig. 5. Again, it is shown that the circulation ratio for the NH_3-NaSCN cycle is higher than the other two cycles.

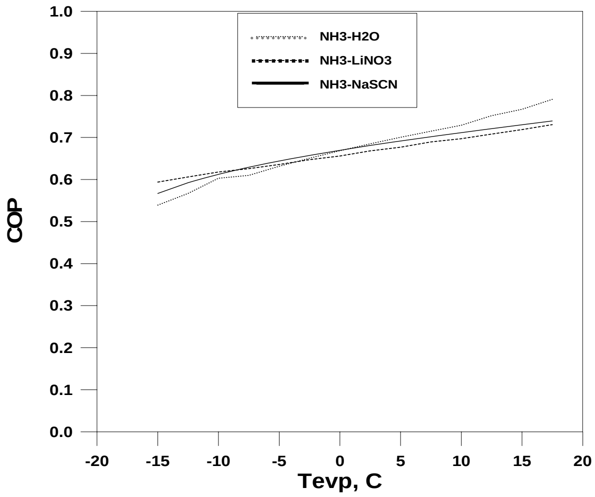

Fig. 4. Variation of COP with evaporator temperature

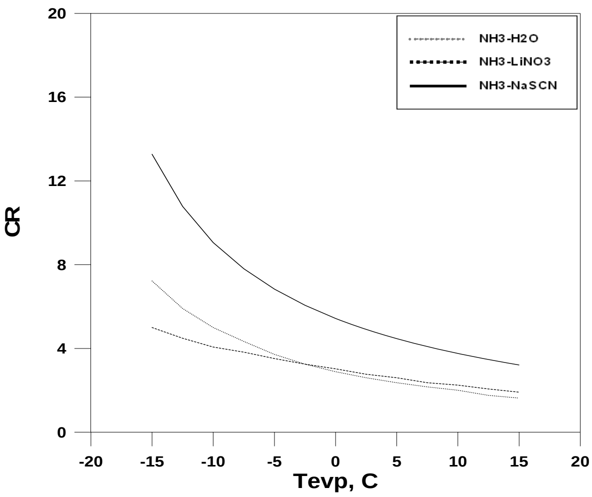

Fig. 5. Variation of CR with evaporator temperature

Fig. 6 illustrates the comparison of COP values vs condenser temperatures for HN_3-H_2O, NH_3-$LiNO_3$ and NH_3-NaSCN absorption cycles. Increasing condenser temperatures cause a decrease in system performance for each cycle. Fig. 7 illustrates the corresponding comparison of circulation ratios vs condenser temperatures. The circulation ratio for the NH_3-NaSCN cycle is still higher than for the other two cycles. For condenser temperatures ranging from 20 C to 40 C, both the NH_3-NaSCN and NH_3-$LiNO_3$ cycles show better performance than the NH_3-H_2O cycle. The effect of absorber temperature is similar to that of condenser temperature. The advantages for using the NH_3-NaSCN and NH_3-$LiNO_3$ cycles are very similar, however, for the NH_3-NaSCN cycle, it cannot operate below -10°C evaporator temperature because of the possibility of crystallization [7].

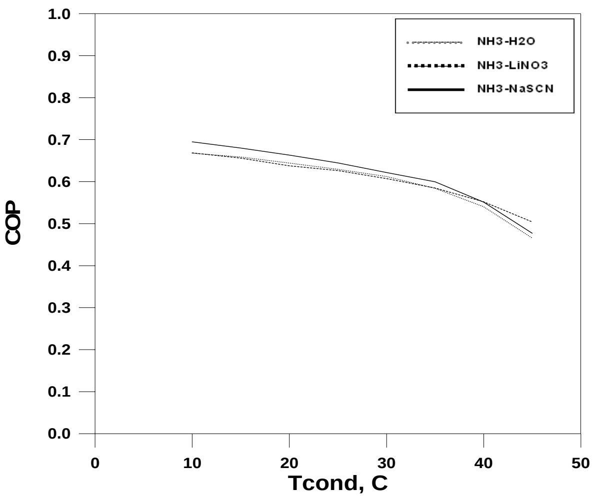

Fig. 6. Variation of COP with condenser temperature

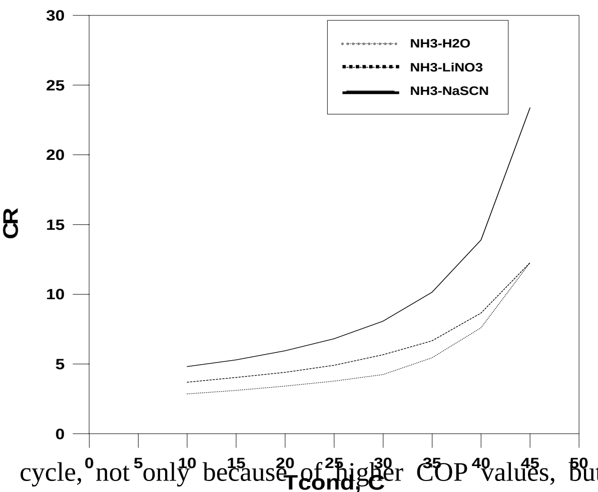

Fig. 7. Variation of CR with condenser temperature

V. CONCLUSIONS

Detailed thermodynamic design data and optimum results to compare the performance of ammonia-water, ammonia-lithium nitrate and ammonia-sodium thiocyanate solar absorption cycles are presented. The results are calculated using computer program based on thermodynamic properties data for the working fluids. The ammonia-water absorption cycle is mainly used for refrigeration temperatures below 0°C. Alternative refrigerant-absorption pairs are being developed for improving system performance. The results show that the ammonia-lithium nitrate and ammonia-sodium thiocyanate cycles give better performance than the ammonia-water cycle, not only because of higher COP values, but also because of no requirement for analyzers and rectifiers. Therefore, they are suitable alternatives to the ammonia-water cycle. Generally speaking, the performance for the ammonia-lithium nitrate and ammonia-sodium thiocyanate cycles are similar, with the latter being slightly better than the former. However, the ammonia-sodium thiocyanate cycle cannot operate at evaporator temperatures below -10°C for the possibility of crystallization. It is hoped that these results can serve as a source of reference for comparison in developing new cycles and new working fluid pairs. These results can be used to select operating conditions for these cycles and realize automatic control for maintaining optimum operating of these systems under different conditions.

Nomenclatures

COP Coefficient of performance
CR Circulation ratio
E Effectiveness
h Enthalpy (kJ/kg)
m Mass flow rate (kg/s)
P Pressure (kPa)
Q Thermal energy (kW)
X Ammonia mass fraction in solution
T Temperature (K)
W work input to pump (kW)

Subscripts

abs Absorber
cond Condenser
evp Evaporator
ex Solution heat exchanger
gen Generator
l Liquid
me Mechanical
v Vapor

Greek

v Specific volume (m_3/kg)
ρ Density (kg/m_3)

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