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Thermodynamic Analysis of a Novel Thermal Driven Refrigeration System

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***Abstract*—**Thermal-driven refrigeration systems have attracted increasing research and development interest in recent years. These systems do not cause ozone depletion and can reduce demand on electricity. The main objective of this work is to perform theoretical analyses of a thermal-driven refrigeration system using a new sorbent-sorptive pair as the working pair. The active component of sorbent is sodium thiocyanate (NaSCN). Ammonia (NH_3) is chosen as sorptive. Based on the thermodynamic properties of the working solution, a mathematical model is introduced to analyze the system characteristics and performance. The results are used to compare with other thermal-driven refrigeration systems. It is shown that the advantages provided by this system over other absorption units include lower generator and evaporator temperatures, a higher coefficient of performance (COP). The COP is about 10 percent higher than the ones for the NH_3-H_2O system working at the same conditions.

***Keywords*—**Absorption; Ammonia-Sodium thiocyanate; Exergy; coefficient of performance (COP).

I. Introduction

NOWADAYS, with the rapid technology development, advanced absorption cooling system has become an accepted and sound alternative for cooling residential, light commercial, and industrial process application. The air-cooled, NH_3-H_2O and H_2O-LiBr cycle continues to offer many distinct advantages to this field. Both working fluid mixtures had some disadvantages to limit their applications. The disadvantage of H_2O-LiBr systems came from its negative pressure and corrosion, while the main disadvantage of the NH_3-H_2O system was its high water content in the vapour phase which requires an auxiliary rectifier to separate it from NH3. Furthermore, the NH_3-H_2O cycle exhibits a relatively low COP. Efforts are being made to search for better refrigerant-absorbent pairs that can improve the system performance [1].

In our recent research, a salt-mixture has been chosen as the sorbent that has been developed by Thermalfrost Inc. (Ottawa, Canada). and has been used in the ice-maker fishery industry [2]. The major active component of the sorbent is sodium thiocyanate (NH_3-NaSCN). NH_3 is chosen as sorptive. Among the first studies on the NH_3-NaSCN system are those of Blytas and Daniels [3] and Sargent and Beckman [4]. Tyagi [5] and Aggarawal et al. [6] provide related studies and reviews on ammonia-salts absorption refrigeration systems. Thermodynamic and physical properties for the NH_3-$LiNO_3$, and NH_3-NaSCN solutions have been presented by C. A. Infante Ferreira [7], while interesting comparisons between NH_3-H_2O, NH_3-$LiNO_3$ and NH_3-NaSCN absorption systems may be found in the works of Bourseau and Bugarel [8] and Sun [9].

In this paper, the first and second law based thermodynamic analysis is carried out for to evaluate the thermodynamic performance of an NH_3-NaSCN absorption system. Second law analysis is based on the concept of exergy, which can be defined as a measure of work potential or quality of different forms of energy relative to environmental conditions [10, 11]. A number of researchers have conducted energy and exergy analysis for refrigeration systems. However, there is no academic work on exergy analysis for the NH_3-NaSCN system found in open literature. This study focuses on performing an energy and exergy analysis on a NH_3-NaSCN absorption system at different operating conditions.

II. Mathematical Model

Based on the basic thermal properties of the working pair, and the heat and mass transfer relationships for the cycle components, a mathematical model was built to predict the effect of various operating conditions on the system performance. In this model, simultaneous heat and mass transfer has been considered instead of only heat transfer as in previous works. The influences of absorber, generator, condenser and evaporator were simulated independently to predict the system performance.

A. Thermodynamic Properties

The performance and efficiency of an absorption refrigeration system are determined to a large degree by the properties of the working fluids. In this new thermal-driven refrigeration system, ammonia/sodium thiocyanate was chosen as the main working fluid because of its great advantages mentioned above. For this cycle, ammonia is the refrigerant, and sodium thiocyanate is the absorbent, so the thermodynamic properties for ammonia and ammonia/sodium thiocyanate solution were collected. For ammonia/sodium thiocyanate mixture, the thermodynamic properties have been studied by C. A. Infante Ferreira [7]. The relation among saturation equilibrium pressure, concentration and temperature is given

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as:

$$\ln P = A + \frac{B}{T} \tag{1}$$

Where

$$A = 15.7266 - 0.298628\,X \tag{2}$$

$$B = -2548.65 - 2621.92(1 - X)^3 \tag{3}$$

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

$$h = A + B(T - 273.15) + C(T - 273.15)^2 + D(T - 273.15)^3 \tag{4}$$

Where

$$A = 79.72 - 1072\,X + 1287.9\,X^2 - 295.67\,X^3$$

$$B = 2.4081 - 2.2814\,X + 7.9291\,X^2 - 3.5137\,X^3$$

$$C = 10^{-2}\left(1.255\,X - 4\,X^2 + 3.06\,X^3\right)$$

$$D = 10^{-5}\left(-3.33\,X + 10\,X^2 - 3.33\,X^3\right)$$

Within the ranges of pressure and temperature concerning refrigeration applications, the two phase equilibrium pressure and temperature of the refrigerant ammonia are linked by the relation:

$$P = 10^3 \sum_{i=0}^{6} a_i (T - 273.15)^i \tag{5}$$

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

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

$$h_v = \sum_{i=0}^{6} c_i (T - 273.15)^i \tag{7}$$

The above three equations are fitted by Sun [9] with source data taken from ASHRAE handbook [12]. The coefficients are listed in Table I.

The relation between temperature and specific enthalpies of superheated vapor ammonia is given as:

$$h_{superheated}(T_2) = h_{saturated}(T_1) + \int_{T_1}^{T_2} c_p\, dT \tag{8}$$

Where T_1 is the saturated temperature corresponding to the pressure of this superheated vapor ammonia, and hsaturated(T_1) is the corresponding enthalpy, T_2 is temperature of the super-heated vapor ammonia, hsuperheated(T_2) is the corresponding enthalpy. Cp is the specific heat and is given as:

$$C_p = \sum_{i=0}^{6} d_i (T - 273.15)^i \tag{9}$$

The above equations are collected by Zhu et al .[13] with source data taken from the properties of R-717 (ANHYDROUS AMMONIA) handbook presented by Industrial Refrigeration Consortium at University of Wisconsin [14], and the coefficient di is listed in Table I.

TABLE I
COEFFICIENTS OF EQUATIONS 5, 6, 7 AND 9

i	a_i	b_i	c_i	d_i
0	4.2871×10^{-1}	1.9879×10^{2}	1.4633×10^{3}	1.7467×10^{4}
1	1.6001×10^{-2}	4.4644×10^{0}	1.2839×10^{0}	-3.3129×10^{2}
2	2.3652×10^{-4}	6.2790×10^{-3}	-1.1501×10^{-2}	2.6189×10^{0}
3	1.6132×10^{-6}	1.4591×10^{-4}	-2.1523×10^{-4}	-1.1045×10^{-2}
4	2.4303×10^{-9}	-1.5262×10^{-6}	1.9055×10^{-6}	2.6214×10^{-5}
5	-1.2494×10^{-11}	-1.8069×10^{-8}	2.5608×10^{-8}	-3.3202×10^{-8}
6	1.2741×10^{-13}	1.9054×10^{-10}	-2.5964×10^{-10}	1.7539×10^{-11}

B. Cycle Description

Fig. 1 illustrates the main components of our system. High-pressure liquid refrigerant (2) from the condenser passes into the evaporator through an expansion valve V1 that reduces the pressure of the refrigerant to the low pressure level as in the evaporator. The liquid ammonia vaporizes in the evaporator by absorbing heat from the material being cooled and resulting low partial pressure vapor (3) passes to the absorber, where it is absorbed by the strong solution (8) coming from the generator through the generator absorber exchanger (GAX), and form the weak solution (4). The weak solution is pumped to the solution heat exchanger and then the generator, and the solution is boiled in the generator. The remaining solution (7) flows back to the absorber and the superheated vapor ammonia passes into the condenser and liquefied to high pressure liquid ammonia by release heat to the cooling material and, thus, completes the cycle. And the function of GAX is to improve system performance.

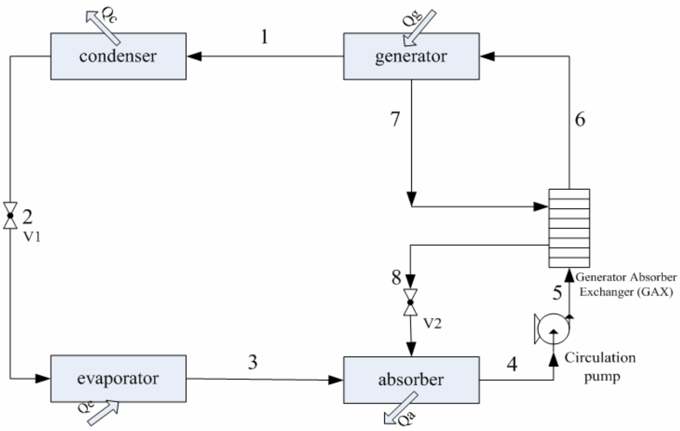

Fig. 1 The schematic of the absorption refrigeration cycle

Based on the properties of the working solution pair, the equations of mass and energy conservation are determined to describe the heat and mass transfer in every component. For the absorber, they are given in final forms as follows:

Mass conservation equation:

$$\dot{m}_3 + \dot{m}_8 = \dot{m}_4 \quad \text{(Total mass balance)} \tag{10}$$

$$\dot{m}_3 + \dot{m}_8 X_8 = \dot{m}_4 X_4 \quad \text{(Ammonia mass balance)} \tag{11}$$

Energy conservation equation:

$$Q_a = \dot{m}_3 h_3 + \dot{m}_8 h_8 - \dot{m}_4 h_4 \tag{12}$$

the circulation ratio of the system can be defined as:

$$f = \frac{\dot{m}_4}{\dot{m}_3} \tag{13}$$

The system performance is measured by the coefficient of performance (COP):

$$COP = \frac{Q_e}{Q_g + W_{pump}} \tag{14}$$

The exergetic coefficient of performance is:

$$ECOP = \frac{Q_e\left(1 - {T_0}/{T_e}\right)}{Q_g\left(1 - {T_0}/{T_g}\right) + W_{pump}} \tag{15}$$

Where T_0 is the reference temperature.

Finally, energy and exergy balances for the generator, condenser and evaporator can be obtained in the same way as the absorber. If the generator, condenser, absorber and evaporator temperatures and the refrigerant mass flow rate or the required refrigeration load are given, combined with the thermodynamic properties of NH_3-NaSCN solution and liquid and vapor ammonia, the above equations can be solved simultaneously to predict the system performance.

III. Results and Discussion

As described above, the system components include a generator, a condenser, an evaporator, an absorber and a solution heat exchanger. The effect of each component on the system performance was simulated and the results were presented in the form of graphs. During each simulation, only one of the components' parameters, such as the generator temperature or solution exchanger efficiency changes while the others remain constant.

Fig. 2 shows the influence of generator temperature on the system performance. It can be observed that with the generator temperature increase, the required circulation ratio decreases, and the system COP value increases. If the generator temperature is below 58°C, the system COP will be close to zero, the system does not work at all. And from 58°C to 75°C, the COP increase rapidly, after 80°C of generator temperature, the COP will increase in a slower pace. This system can be operated with a relative low generator temperature to reach low evaporator temperature with an acceptable system COP, which is a main advantage of this new refrigeration system, as it could then utilize industry or civil waste heat and solar energy since fluid temperatures of this kind of heat source are generally low.

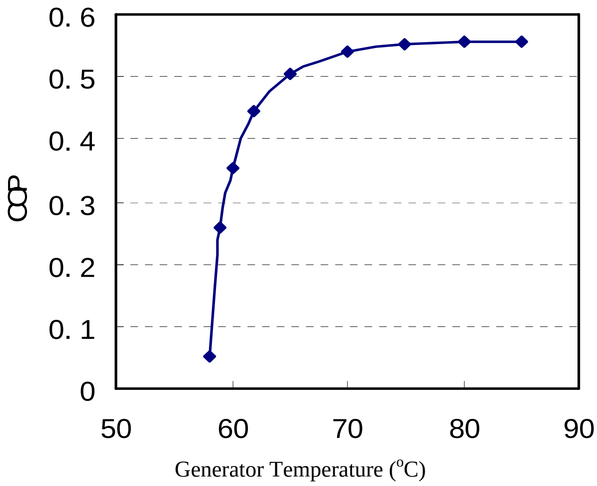

Fig. 2 The influence of the generator temperature on system performance (T_c=15°C, T_e=-20°C, T_a=15°C)

Fig. 3 shows the influence of condenser temperature on the system performance. The system COP decreases with the condenser temperature increase, at the same time, circulation ratio will increase. The influence of absorber temperature is almost the same as the influence of condenser temperature. The influences of the evaporator temperature are given in Fig. 4 and Fig. 5, for evaporator temperature changing from -35°C to 5°C, both cooling capacity and COP will increase.

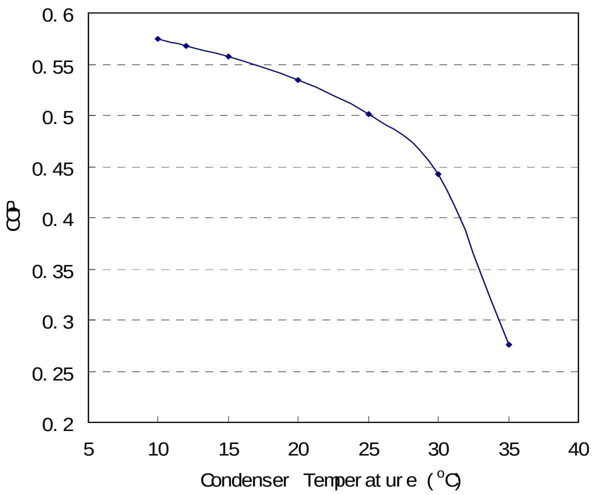

Fig. 3 The influence of the condenser temperature on system performance (T_g=85°C, T_e=-20°C, T_a=15°C)

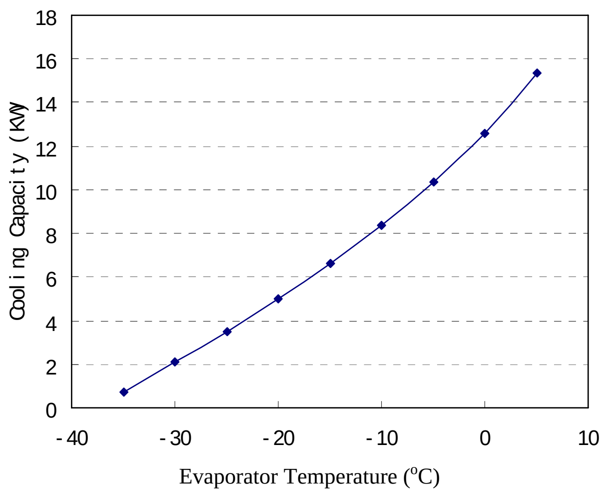

Fig. 4 The influence on cooling capacity of the evaporator temperature on system performance (T_g=85^{o}C, T_a=15^{o}C, T_c=15^{o}C)

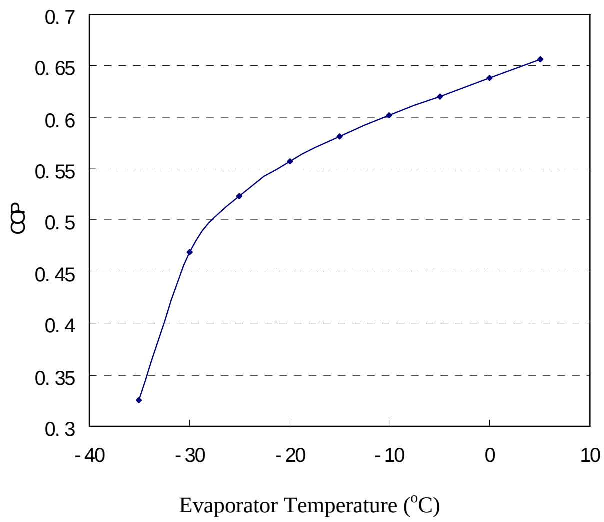

Fig. 5 The influence on COP of the evaporator temperature on system performance (T_g=85^{o}C, T_a=15^{o}C, T_c=15^{o}C)

Fig. 6 gives the influence of the GAX on the system performance. It can be observed that the system performance is improved with the solution heat exchanger effectiveness. If the GAX effectiveness value is zero, it can be understood that no solution heat exchanger is included in the cycle. At this working condition, the COP value is only about 0.39. With the solution heat exchanger effectiveness increase, the system COP will increase gradually. In generally, a heat exchanger effectiveness value of 0.8 is commonly chosen for this simulation. It can be observed that for this effectiveness the COP value is about 0.56, which is 43.6% higher than that without a solution heat exchanger. The effect of a solution heat exchanger on the whole system performance is significant.

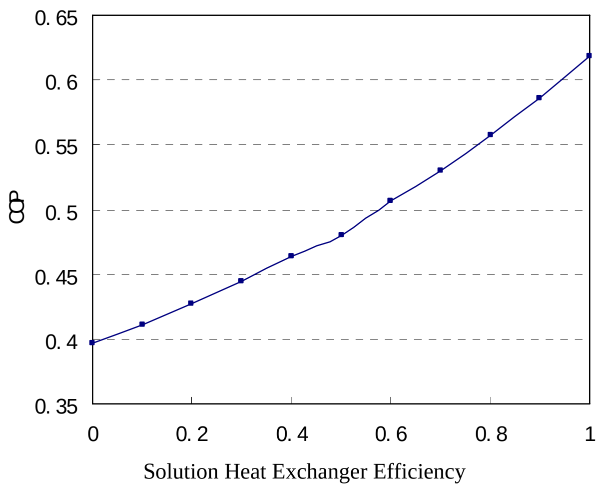

Fig. 6 The influence of the GAX on system performance (T_g=85^{o}C, T_e=-20^{o}C, T_c=T_a=15^{o}C)

Fig. 7 presents that the comparison of the coefficient of performance and exergetic efficiency of the absorption system, versus different generator temperatures. It is shown that the exergetic efficiency initially increases and declines continuously as the generator temperature increases. It also should be noted that the COP initially exhibits significant increase with an increasing generator temperature, and when the generator temperature is higher than 75^{o}C, the slope of the COP curves becomes almost flat. This suggests that, increasing the generator temperature higher than a certain value will not provide much improvement in COP. This can be explained by the fact that although a system with a high generator temperature can produce more hot ammonia vapour, more input exergy is supplied; it also generates more exergy losses in the generator, condenser and absorber as their average temperatures rise up. This contributes negatively to the exergetic efficiency of the system. Since this negative effect of increasing the generator temperature is more dominant on the exergetic efficiency of the system, while the system COP stays almost constant. This negative result on the exergetic efficiency and COP trades off the beneficial effect of the generator temperature increase.

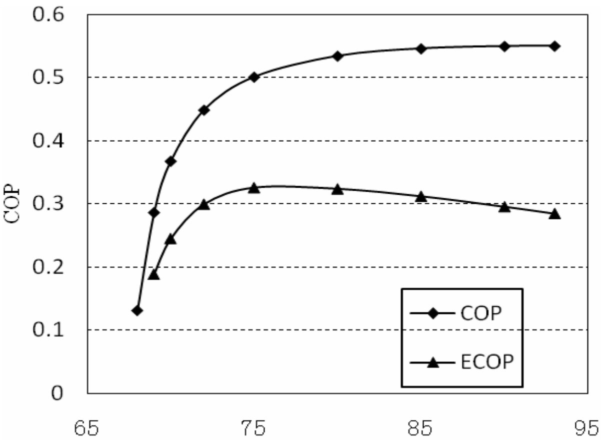

Fig. 7 Comparison of the COP and exergetic efficiency of the system with the generator temperature (T_e=-10ºC, T_c=T_a=25ºC)

Fig. 8 shows comparison of the COP and exergetic efficiency of the absorption system with the evaporator temperature. As it can be seen, the cooling COP increases when the evaporator temperature increases. The reason is that a higher evaporator temperature will cause a higher absorbing pressure, which will greatly increase the absorption efficiency of the strong solution. Unlike COP, the exergetic efficiency of the system decreases with the increase of the evaporator temperature. It can also be explained by the definition of the second law of thermodynamics that the lower evaporator temperature has a bigger potential to create cooling effect.

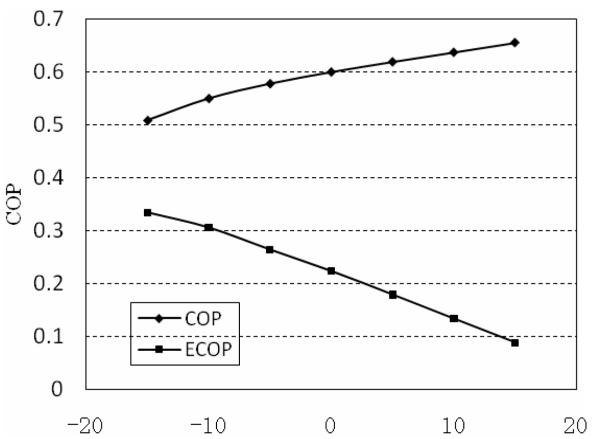

Fig. 8 Compariaon of the COP and exergetic efficiency of the system with evaporator temperature (T_g=90ºC, T_c=T_a=25ºC)

IV. Conclusion

In this paper, the first and the second law of thermodynamics are applied to a single stage NH_3-NaSCN absorption system, the performance analysis of each component are calculated. Results shows that the system COP increase with increasing generator and evaporator temperatures, but decrease with increasing condenser and absorber temperatures. But the first law based thermal analysis is not adequate. In cretain operating conditions, while the COP increases, the efficiency of the second law of thermodynamics may decrease, the reason is that more exergy loss occurs in some components of the system. In these cases, attention can be focused on such components that have high exergy losses in order to increase the exergetic efficiency of the system.

Finally, the results of the energy and exergy analysis presented in this paper can be applied as a useful tool for evaluation and improvement of the absorption systems, it provides a simple and effective method to identify how losses at different devices are interdependent and where a given design should be modified for the best performance.

Acknowledgement

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