Theoretical Study of New Combined Absorption-Ejector Refrigeration System

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Abstract. An improved system of the new combined single stage absorption cycle operated with NH3/H2O as working fluid was performed. In order to enhance performance the cycle a new configuration of absorption system was utilized. The performances of two configurations of the combined absorption cycle were compared; a) with common solution heat exchanger and b) divided the streamline of solution heat exchanger to recover the internal heat. Based on the analysis, it has been shown that the second configuration a significant reduction of the required generator and absorber loads by about 20% and 17% respectively, with increased coefficient of performance (COP) about 12 % compared to the first configuration. This improvement in the overall COP is found due to improve energy utilization efficiency significantly.

1. Introduction
Absorption cycles in cooling systems represent an alternative to predominant compressor refrigerating cycles. Its application to solar cooling makes this technology attractive due to the synchronicity between solar thermal energy and air-conditioning demand. The advantage of the absorption cycles is their much lower consumption of electricity achieved by replacement of a compressor by so called a thermo-chemical compressor. A thermo-chemical compressor ensures absorption of the refrigerant vapor in an absorption liquid, pumping of the created solution to a higher pressure level, and absorption of the refrigerant vapor from the solution. The main benefits of the absorption cycles are the following: very low consumption of electricity (only up to 5 % of the cooling capacity), possible utilizing of thermal energy obtained from renewable sources (solar radiation, biomass combustion) or whatever source of waste heat energy with temperature above 80 °C for the production of cooling. Another positive effect of the absorption cycles is utilizing of ammonia as refrigerant. Disadvantage of the absorption cycles is the overall performance of the absorption cycle in terms of cooling effect per unit of energy input is generally poor. For this reason, the researchers have been devoted to improvement the absorption cooling systems. However, the COP of the NH3/H2O systems is limited due to various constraints posed by the component temperatures and pressures. Combining of the ejector with the single stage absorption cycle showed potential improvement in the COP of the cycle.

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It was showed that COP's as high as 0.86 to 1.04. However, this system must be operated with a high temperature heat source (190 to 210°C) and acceptable surrounding temperature. The advantage of using an ejector refrigerator cycle can utilize thermal energy at temperature levels upward from 333 K [2]. Heat energy at these temperatures is available from the absorber of the absorption refrigerator. Previous researches demonstrated that the COP of absorption and ejector refrigerators are in the ranges 0.5–0.8, while the basic absorption cycle 0.2–0.6 [3, 4]. Many researchers have improved the performance of the single effect absorption cooling system in recent years [5, 6]. It was showed further enhancement by adding flash tank to the cycle with the presence of the ejector. Moreover, experimental studies shows that COP can be increased up to 60% when a solution heat exchanger is used [7]. The effect of the generator–absorber temperature on the system performance has been studied [8-12]. They informed that the system has a low generator temperature limit which that the cycle cannot be operated at generator lower than its limit. The optimum generator temperature determines the optimum solution concentration leaving the generator. Therefore, there is thermal energy wasted within the generator and the absorber, when the solution temperature at the generator higher than its optimum value. From this reasons, for thermal energy saving, the modifications are made in the system to save the energy wasted and to improve the COP of the cycle. In the present study, the effect of streamline of strong solution and heat exchanger effectiveness on the thermal loads of components, entrainment ratio ($\omega$) and coefficient of performance are investigated.

2. System Description

Figure 1 shows schematically the main part of (combined ejector–absorption refrigeration system) was done by [13] and [5]. The schematic diagram of the proposed new design is shown in Figure 2. In this new design the ammonia mixture, leaves the absorber (state 11) as a saturated solution at the low pressure with a relatively high ammonia concentration. It is pumped to the system high pressure (state 12) and divided into two parts. : One part flows to the solution heat exchanger (state 12A) it used to recover the heat from the returning weak liquid solution in the recovery heat exchanger before entering the generator, meanwhile the remaining part flows to the rectifier. In the rectifier the vapor is cooled by the rich solution pumped from the absorber to the generator (state 12B). As the generator operates from high temperature source to separate the binary solution of water and ammonia (strong solution comes from absorber), the basic solution is partially heated to produce a two-phase mixture: a liquid (state 14), which is relatively weak in ammonia, and a vapor (state 1) with a high concentration of ammonia. This two-phase mixture is separated, and the weak liquid transfer heat to the high concentration stream before it is throttled to system low pressure and sprayed into the absorber. The rectifier cools the saturated ammonia vapor (state 2) to condense out any remaining water (state 16). It is assumed that the ammonia vapor concentration leaved the rectifier passed to the ejector is 99.6%. On the ejector, the secondary flow (ammonia vapor from the mixing of the evaporator and flash tank (state 17) and the primary flow of ammonia vapor from the rectifier are mixed and passed to the condenser (state 3). In the combined flash–ejector absorption cooling system done by Sirwan et al.[5], the flash tank still serves as an accumulator storing excessive refrigerant for test conditions requiring smaller mass in the system’s high pressure side. In order to mix the vapor flow with the evaporator exit flow the Ft valve is necessary to create an identical pressure drop across the vapor. The condensed refrigerant from the condenser is separated into two streams: one stream is sucked into the refrigerant heat exchanger (state 5), meanwhile, other flows into the flash tank (state 6). When refrigerant heat exchanger is employed, the refrigerant entering the absorber (state 10A) and the ejector (state 10B) has been superheated by heat exchange with mix liquid exiting from condenser and flash tank (state 7) which causes the liquid to enter the expansion device in a subcooled state (state 7'). This subcooling reduces the fraction of flash gas produced at the downstream side of expansion valves.
3. Thermodynamic analysis of absorption cooling system

The mass balances, energy balances, and the equations of state for NH₃/H₂O solution, and refrigerant at each component involved in the cycle are required for system study. In order to calculate the heat and mass balance for the proposed cycle, the thermodynamic properties (pressure, temperature, concentration, enthalpy and density) are necessary for the simulation. The binary mixture of NH₃/H₂O and pure NH₃ are used in the proposed system. The detailed thermodynamic property equations of NH₃/H₂O are found by [14] and [15]. To evaluate the COP for the cycle, the governing equation required through the each component.

- Mass balance and Energy balance

\[ \sum_i \dot{m}_a \dot{h}_a = \sum_i \dot{m}_i \dot{h}_i \]  

(1)
\[ Q_K = \sum \dot{m}_k h_k - \sum \dot{m}_k h_i \]  

(2)

where \( Q_K \) is the heat added to component K at temperature \( T_k \).

- **Pump**

The electrical power needed for the pump

\[ w_p = \frac{m_{11} \eta_{11}}{\eta_{sed} \eta_{mec}} \]  

\[ w_p = m_{12} h_{12} - m_{11} h_{11} \]  

(4)

- **Heat exchanger**

The solution and refrigerant heat exchanger performance, expressed in terms of an effectiveness \( \varepsilon \).

\[ T_7' = T_{sHE} + (1 - \varepsilon_{sHE}) T_7, \quad m_7' = m_7, \quad X_7 = X_7', \]  

\[ h_7' = h_7(T_7', P_7') \]  

(5)

\[ T_{14} = T_{14A} \varepsilon_{sHE} + (1 - \varepsilon_{sHE}) T_{14}, \quad X_{14} = X_{14}', \quad P_{14} = P_{s}, \]  

\[ h_{14}' = h_{14}(T_{14}', X_{14}') \]  

(6)

The component's thermal loads of the combined ejector cooling system per unit of refrigerant mass are expressed as follows:

\[ Q_s = m_{11} h_{11} + m_{14} h_{14} - (m_{13} h_{13} + m_{16} h_{16} + m_{13A} h_{13A}) \]  

(7)

\[ Q_s = m_{11} h_9 - h_8 \]  

(8)

\[ Q_s = (1 + \omega) m_2 (h_3 - h_4) \]  

(9)

\[ Q_s = m_{10A} h_{10A} + m_{15} h_{15} + m_{11} h_{11} \]  

(10)

The coefficient of performance COP is defined as

\[ \text{COP} = \frac{Q_s}{Q_0} \]  

(11)

- **Equation of state**

Finally, The thermodynamic properties at states (1)-(10), and (17) in Figure 2 are determined by NH3 and other properties at states (11)-(16) can be calculated based on the binary mixture of NH3/H2O which are given by the correlations of [14] and [15]:

**4. Results and Discussion**

EES software is used to evaluate the performance of the system using NH3/H2O as working fluid. The operation temperatures range of the proposed cycle are selected as \( T_{gen} = 60 - 120 \) °C, \( T_{cond} = T_{abs} = 20 - 50 \) °C, \( T_{evp} = -15 - 15 \) °C, and mass flow rate of refrigerant = 0.0166 kg/s. The effectiveness of the solution and refrigerant heat exchangers is assumed 0.5 and 0.6 respectively.

**4.1 Evaluation of utilizing the streamline of strong solution**

In this work, the effects of the streamline of strong solution on the performance of the proposed cycle are analyzed. The effects of the streamline of strong solution on the thermal loads of generator and absorber are shown in Figs. 3 and 4. It can be seen that the heat load of the generator decreases with an increase temperature of the strong solution entering the generator. Similarly, with a decrease in the weak solution temperature entering the absorber, the heat rejected from the absorber also decreases.
For this reason, the reduction ratios of both generator and absorber thermal loads increase with increases the generator temperature by about 20% and 17% respectively as shown in Figure 3. Also it is found that the decreasing ratios in both generator and absorber thermal loads increase with the increases of SHE effectiveness and reach up to 38% as shown in Figure 4.

Figure 3. Reduction in thermal loads versus the generator temperature.

Figure 4. Reduction in thermal loads versus the effectiveness of SHE

As it can be seen from Figure 5, when the generator temperature increases, the thermal loads of the generator and absorber decrease. The optimum generator temperatures corresponding to the minimum thermal loads of the generator and absorber after this optimum value the thermal loads remain unchanged. If the generator temperature gets higher, the concentration of the solution leaving the generator increase, and hence, the circulation ratio decreases. The thermal loads of the condenser and the evaporator remains constant, as the generator temperatures increases. Moreover, the entrainment ratio $\omega$ remains almost constant under generator temperature range.

Figure 5. Variation of thermal loads with the generator temperature at $T_C=30 ^\circ C$, $T_a=25 ^\circ C$, $T_e=-5 ^\circ C$. 
4.2 Comparison between the first and second configuration

Figures 6 a & b illustrate the variation of the coefficient of performance with generator temperature for five scenarios of different operating conditions (condenser, absorber and evaporator temperature) for combined cycle before and after the modification. These five scenarios can be used to evaluated the COP of the systems under generator temperature range (70-110 °C), condenser temperature range (25-45 °C) to operate under water –cooled or air cooled condenser. The range of the evaporator temperature is (-10 to 10 °C) that shows the best cooling effect under the employed generator and ambient –condensation temperature ranges. The overall COPs are found higher at the new modified combined cycle (second configuration) about 12 % than that of combined cycle (first configuration). This improvement in the overall COP is found due to minimize the energy consumption at the generator, and absorber. As well as, the RHE is used to assist the flash tank to improve the quality of the refrigerant that enters the evaporator.

![Image of COP vs generator temperature]

**Figure 6 (a&b)** variation of COP vs generator temperature at different operating conditions for combined cycle: a) first configuration, and b) second configuration.

5. Conclusion

In this study, a single stage combined ejector–absorption cooling system using NH3/H2O as working fluid was analyzed and then a new improved configuration based on this system was proposed. The effects of the streamline of strong solution on the performance of the proposed cycle were investigated. It has been shown that the second configuration a significant reduction of the required generator and absorber loads by about 20% and 17% compared to the ordinary combined cycle. It was also found that the SHE decreasing the thermal loads of generator and absorber by about 38%. The thermal loads of the condenser and the evaporator remains constant, as the generator temperatures increases. Moreover, the entrainment ratio $\omega$ remains almost constant under generator temperature range. The overall COPs are found higher at the new modified combined cycle about 12 % than that of ordinary combined cycle. This improvement in the overall COP is found due to utilize the internal heat recovery in the new modified cycle.
6. References

[1] Aphornratana S and Eames I W 1998 Experimental investigation of a combined ejector- 
absorption refrigerator International Journal of Energy Research 22 195-207

[2] Sun D-W, Eames I W and Aphornratana S 1996 Evaluation of a novel combined ejector-
absorption refrigeration cycle — I: computer simulation International Journal of 
Refrigeration 19 172-80

[3] Srikhirin P, Aphornratana S and Chungpaibulpatana S 2001 A review of absorption 
refrigeration technologies Renewable and Sustainable Energy Reviews 5 343-72

[4] Ward D S 1979 Solar absorption cooling feasibility Solar Energy 22 259-68

[5] Sirwan R, Alghoul M A, Sopian K, Ali Y and Abdulateef J 2013 Evaluation of adding flash 
tank to solar combined ejector–absorption refrigeration system Solar Energy 91 283-96

[6] Sirwan R, Alghoul M A, Sopian K and Ali Y 2013 Thermodynamic analysis of an ejector-
flash tank-absorption cooling system Applied Thermal Engineering 58 85-97

[7] Aphornratana S 1995 Theoretical and experimental investigation of a combined ejector-
absorption refrigerator. University of Sheffield, Department of Mechanical and Process 
Engineering

[8] Kang Y, Akisawa A and Kashiwagi T 1999 An advanced GAX cycle for waste heat recovery: 
WGAX cycle Applied Thermal Engineering 19 933-47

[9] Kang Y T, Chen W and Christensen R N 1996 Development of design model for a rectifier in 
GAX absorption heat pump systems Ashrae Transactions 102 963-72

[10] Grossman G, DeVault R C and Creswick F A 1995 Simulation and performance analysis of an 
ammonia-water absorption heat pump based on the generator-absorber heat exchange (GAX) 
cycle. Oak Ridge National Lab., TN (United States))

[11] Sun D-W 1997 Computer simulation and optimization of ammonia-water absorption 
refrigeration systems Energy Sources 19 677-90

[12] Sun D-W 1998 Comparison of the performances of NH3-H2O, NH3-LiNO3 and NH3-NaSCN 
absorption refrigeration systems Energy Conversion and Management 39 357-68

[13] Abdulateef J M, Sopian K, Alghoul M A and Sulaiman M Y 2009 Review on solar-driven 
ejector refrigeration technologies Renewable and Sustainable Energy Reviews 13 1338-49

[14] Bourseau P and Bugarel R 1986 Absorption-diffusion machines: comparison of the 
performances of NH3H2O and NH3NaSCN Réfrigération par cycle à absorption-diffusion: 
comparaison des performances des systèmes NH3H2O et NH3NaSCN 9 206-14

[15] Pátek J and Klomfar J 1995 Simple functions for fast calculations of selected thermodynamic 
properties of the ammonia-water system International Journal of Refrigeration 18 228-34

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