

Thermodynamic Investigation of Two-Stage Absorption Refrigeration System Connected by a Compressor

L. Kairouani, E. Nehdi and R. Ben Iffa

Unité de Recherche Energétique et Environnement, Ecole Nationale d'Ingénieurs de Tunis
 B.P 37 Le Belvédère 1002, Tunis, Tunisia

Abstract: The present work is to analyze a two-stage cycle based on the ammonia-water absorption system, with intermediate compression. The two generators of the system are heated by geothermal energy at low temperature. The study shows that this system makes it possible at lower generator temperature, under the limits permitted by the systems suggested up to now. For $T_g = 335$ K, $T_c = T_a = 308$ K and $T_c = 263$ K, based on the electric consumption, the system efficiency is 8.2. The comparative study of the hybrid system and vapor compression systems shows the superiority of the proposed system. Supplied by the geothermal sources of the Tunisian south, the system makes it possible to obtain for a pilot geothermal station, a production of 75 tons of ice per day. The greenhouse gas emissions should thus be reduced by about 2.38 tons of CO_2 per day. Therefore, based on the typical geothermal energy sources in Tunisia which present a global refrigeration potential of 4.4 MW, the daily quantity of ice that could be produced is about 865 tons. The greenhouse gas emissions should thus be reduced by about 10,000 tons of CO_2 per year.

Keywords: Absorption, ammonia, geothermal, hybrid, refrigeration

INTRODUCTION

Several research tasks were carried out, for a suitable exploitation of geothermal energy. This exploitation is carried out by means of systems, either by absorption, or hybrid absorption/compression^[1-9]. Among these applications one quotes the heat pumps with absorption and air-conditioning with absorption.

To develop these energies at low temperature, this study proposes the use of a hybrid system with absorption/compression using couple ammonia-water, for the production of ice, with relatively low temperatures in the generator of approximately 335 K.

This system is distinguished from the cascaded systems by the fact that it consumes energy at low temperature, even at condensation and absorption temperatures of about 308 K.

A single-stage absorption cycle using the working mixture ammonia-water are presented in the Oldham diagram, Fig. 1, it consists in two concentration lines (x_s, x_w) and two isobars (P_c, P_e).

So that such a machine can function, the difference of title Δx must be higher or equal to 0.1, it follows that the pressure of evaporation will be fixed (intersection between the concentration x_r line and the isotherm of condensation and absorption).

To ensure the operation of a single stage ammonia-water absorption system, in the conditions ($T_a = T_c = 308$ K, $T_c=263$ K and $\Delta x=0.1$), the generator temperature must exceed 378 K (higher than the actual geothermal temperature source), Fig. 1.

To circumvent this limit, certain researchers studied systems with absorption on several floors, laid out, either cascades about it^[7], or in association absorber-generators^[10-12]. In this study we propose the integration of a compressor insert between two-stage of absorption, which function at the same pair of temperature (T_g, T_a). The high pressure stage functions between P_c and P_{a1} and the low pressure stage between P_{g2} and P_e .

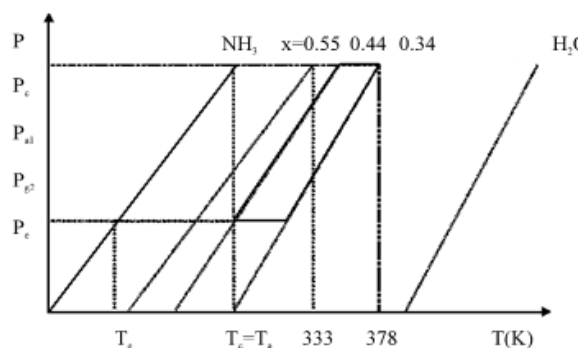


Fig. 1: Diagram of oldham for a one-stage system ($P_c = 276$ kPa, $T_c = T_a = 308$ K)

Corresponding Author: L. Kairouani, Unité de Recherche Energétique et Environnement, Ecole Nationale d'Ingénieurs de Tunis, B.P 37 Le Belvédère 1002, Tunis, Tunisia
 Tel: 216 71874700 Fax: 216 71872729 E-mail: lakdar_kairouani@yahoo.fr

ENERGY AND MASS BALANCES

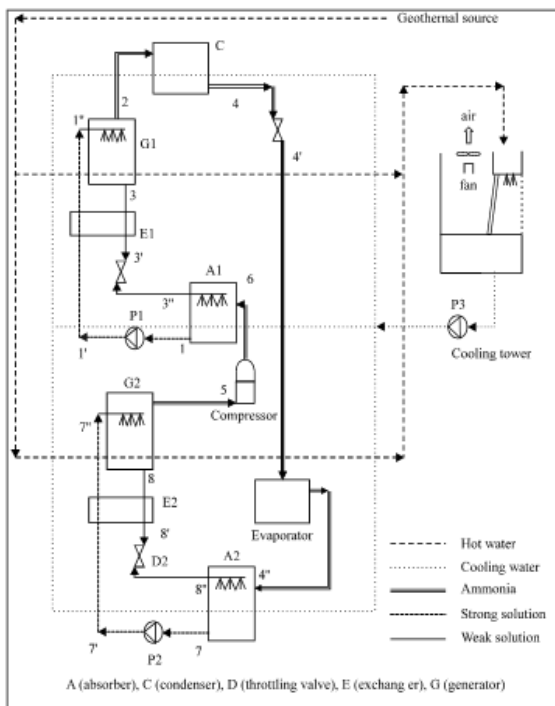


Fig. 2: Schematic view of the hybrid system

The role of the compressor is to raise the pressure of the ammonia vapor coming from the low pressure generator, to supply the high pressure absorber.

DESCRIPTION OF THE HYBRID CYCLE

In order to consider the quantity of ice potentially producible following the use of geothermal water of the Tunisian south, the studied system is formed by the association of two-stage of ammonia-water absorption system supplied with geothermal water coming from drilling Chott Fijj 2 (pilot station), located at 60 km of Gabes. This drilling which is associated others feed out of hot water of the cooling tower, serving the cold water area and of which a part is treated in situations of desalination^[13] (Fig. 2). This geothermal source which used to supply the two generators provides a hot water mass flow of 70 kg s⁻¹ to a temperature of 343 K.

While taking account of the variation ΔT_{Hg} between the heater (geothermal water) and the weak solution in the generator, the temperature of this last is fixed to 335 K.

The condenser and the absorbers are cooled with water, of the basin of collection, located in the bottom of the tower. This cooled water with 300 K, has a total mass flow of 1000 kg s⁻¹.

By taking account of the temperature of cooling waters, T_c and T_a are fixed at 308 K.

For the two stages with absorption, the energy balances are established on the basis of the following assumptions:

- * Absence of imperfections on the absorbers and the generators.
- * The pinch temperature on the generators, the absorbers, the evaporator and the condenser is fixed at $\Delta T_{pin} = 4$ K
- * At the exit of the high pressure generator, the vapor of ammonia is pure.

High pressure generator: The energy and mass balances are established to evaluate the quantity of energy absorbed by the generator and the mass of solution which provides one kg of pure refrigerant (Fig. 2).

It is to be noticed according to Table 1, that the ammonia mass vapor concentration is approximately 0.997, which reduces the needs for a rectification of this vapor.

The specific strong solution circulation is given from the mass balance and it represents the mass of solution per kg of vapor of refrigerant evaporated in the generator:

$$f_1 = \frac{x_{v1} - x_{w1}}{x_{s1} - x_{w1}} \quad (1)$$

The heat supplied to the generator is given by the following expressions:

$$Q_{g1} = m_{NH_3} (h_2 + (f_1 - 1) h_3 - f_1 h_{1'}) \quad (2)$$

The expression becomes in function of the mass flow rate of the source:

$$Q_{g1} = m_{H1} C_{PH} (T_H - (T_g + \Delta T_{pin})) \quad (3)$$

The ammonia mass flow rate is deduced from the expressions (2) and (3)

$$m_{NH_3} = \frac{m_{H1} C_{PH} (T_H - (T_g + \Delta T_{pin}))}{h_2 + (f_1 - 1) h_3 - f_1 h_{1'}} \quad (4)$$

This expression will be used for the determination of the realizable refrigerating potential by geothermal source according to its total mass flow rate m_H and its temperature T_H .

High pressure absorber: To evaluate the heat released by the absorber and to estimate the cooling

water flow, the assessment of energy and mass on the absorber is represented (Fig. 2).

The heat released by the absorber is given by:

$$Q_{a1} = m_{NH_3} (h_6 + (f_1 - 1) h_3'' - f_1 h_1) \quad (5)$$

The expression becomes in function of the mass flow rate of the source:

$$Q_{a1} = \frac{m_{H1} C_{PH} (T_H - (T_g + \Delta T_{pin}))}{h_2 + (f_1 - 1) h_3 - f_1 h_1''} \times (h_6 + (f_1 - 1) h_3'' - f_1 h_1) \quad (6)$$

Evaporator: The refrigerating power obtained on the evaporator of the installation with absorption is given by:

$$Q_e = m_{NH_3} (h_4'' - h_4) \quad (7)$$

The expression becomes in function of the mass flow rate of the source

$$Q_e = \frac{m_{H1} C_{PH} (T_H - (T_g + \Delta T_{pin}))}{h_2 + (f_1 - 1) h_3 - f_1 h_1''} (h_4'' - h_4) \quad (8)$$

Considering the socioeconomic conditions of these areas, it is desirable to use this refrigerating potential Q_e for ice production, for storage of agricultural and fishery products.

The daily quantity of ice is given from (8) by:

$$m_i = 3600 * 24 \frac{Q_e}{C_p (T_{in} - 273) + h_i} \quad (9)$$

h_i is the latent heat of fusion and T_{in} is the inlet water temperature.

Condenser: The condenser and the two absorbers of the combined installation are cooled by water coming from the recovery basin of the cooling tower.

The heat released from the condenser is:

$$Q_c = m_{NH_3} (h_2 - h_4) \quad (10)$$

Low pressure generator: The circuit with absorption low pressure (Fig. 2), operates between the pressure of evaporation corresponding to 263 K (276 kPa), the pressure of the low pressure generator $P_{g2}=414$ kPa and the temperature of absorption $T_{a2}=308$ K. These data make it possible to evaluate the thermodynamic properties of the cycle by using the Merkel's diagram^[14].

It is to be noticed according to Table 1, that the vapor mass concentration of ammonia is approximately 0.99, this vapor compressed, then will be forwarded to the low pressure absorber without rectification.

The specific strong solution circulation is given by:

$$f_2 = \frac{x_{v2} - x_{w2}}{x_{s2} - x_{w2}} \quad (11)$$

The heat added to the generator is given by the expressions (12) and (13):

$$Q_{g2} = m_{NH_3} (h_5 + (f_2 - 1) h_8 - f_2 h_7'') \quad (12)$$

The expression becomes in function of the mass flow rate of the source:

$$Q_{g2} = m_{H2} C_{PH} (T_H - (T_g + \Delta T_{pin})) \quad (13)$$

The ammonia mass flow rate is deduced from the expressions (12) and (13):

$$m_{NH_3} = \frac{m_{H2} C_{PH} (T_H - (T_g + \Delta T_{pin}))}{h_5 + (f_2 - 1) h_8 - f_2 h_7''} \quad (14)$$

The equality of the expressions (4) and (14) will be used for the determination of the ratio of the mass flow rate m_{H1} and m_{H2} .

Notice that the total geothermal mass flow rate is known:

$$m_H = m_{H1} + m_{H2} \quad (15)$$

The necessary ammonia mass flow rate is finally given in function of the total mass flow rate of the source by:

$$m_{NH_3} = \frac{m_H C_{PH} (T_H - (T_g + \Delta T_{pin}))}{h_2 + (f_1 - 1) h_3 - f_1 h_1'' + (h_5 + (f_2 - 1) h_8 - f_2 h_7'')} \quad (16)$$

Compressor: The energy balances are established on the basis of the following assumptions:

- * At the entry of the compressor, the vapor which accounts for ammonia 99% and 1% of water, in mass ratio, is comparable with the pure superheated vapor of ammonia ($P_5 = 414$ kPa and $T_5 = 315$ K).
- * The isentropic and electric efficiency of the compressor is given by^[15]
- * $\eta_{is} = 0.874 - 0.0135 \tau$ and $\eta = 0.9$

Table 1: Thermodynamic data and numerical values^[14, 16]

T (K)	P(kPa)	h ₃ =44	x _{s1} =0.66
T _g =335	P _{a1} =750	h _{3'} =50	x _{w1} =0.56
T _c =308	P _{g2} =414	h ₄ =154	x _{v1'} =0.997
T _a =308	P _c =1270	h _{4'} =1250	x _{s2} =0.41
T _e =263	P _{ev} =276	h ₅ =1357	x _{w2} =0.32
DT _{pin} =4	h(kJ kg ⁻¹)	h ₆ =1417	x _{v2'} =0.99
T ₂ =T _{1'} =318	h ₁ =330	h ₇ =-102	r=1000 kg m ⁻³
T _{3'} =315	h ₁ '=-58	h _{7'} =-40	f ₁ =4.37
T ₅ =T _{7'} =315	h _{1'} =28	h ₈ =50	f ₂ =7.44
T _{8'} =319	h ₂ =1349	h _{8'} =h ₈ =0	

Table 2: Thermodynamic characteristics of geothermal sources and cooling potential

Number of source	Geothermal source	T _H (K)	Mass flow rate (kg s ⁻¹)
1	Chott Fijj 2 (pilot)	343	70
2	El Mahassen	348	100
3	Dégache CI2	346	22
4	Nefta CI2	349	15
5	Tozeur CI1	349	22
6	Zaouiet Echourfa	343	90
7	Zaouia NCI5	343.9	70
8	Menchia NCI6	344	70
9	Limaguess NCI8	345.5	70

$$W_{co} = m_{NH3} \frac{h_{6'} - h_5}{\eta_{hs} \eta_e} \quad (18)$$

Low pressure absorber: To evaluate the heat released by the absorber, the energy and mass balances are represented on the absorber (Fig. 2).

The heat released by the absorber is given by:

$$Q_{a2} = m_{NH3} (h_{4'} + (f_2 - 1)h_{8'} - f_2 h_7) \quad (19)$$

Water cooling pumps: To evacuate the heat released on the condenser and the absorbers, defined by the expressions (6), (10) and (19), a cold water circulation must be ensured by pumps which estimate their electric consumption:

$$W_{Pel} = m_C \frac{\Delta P}{\rho \eta_g} \quad (20)$$

The cold water mass flow rate m_C is given by:

$$m_C = \frac{Q_c + Q_{a1} + Q_{a2}}{C_{PC}((T_c + \Delta T_{pin}) - T_c)} \quad (21)$$

HYBRID INSTALLATION PERFORMANCE AND COMPARISON WITH R134A VAPOUR COMPRESSION SYSTEM

According to the expression (8), the refrigerating potential produced by this hybrid installation is directly related to the characteristics of geothermal energy, like the source mass flow rate and the water temperature.

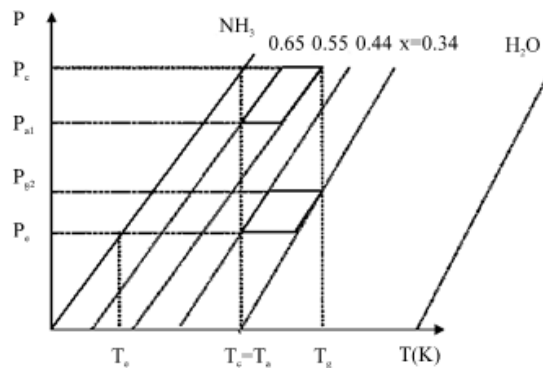


Fig. 3: Diagram of Oldham for a two-stage system (T_c=T_a=308 K, P_{a1}= 750 kPa, P_{g2}= 414 kPa, P_c= 276 kPa)

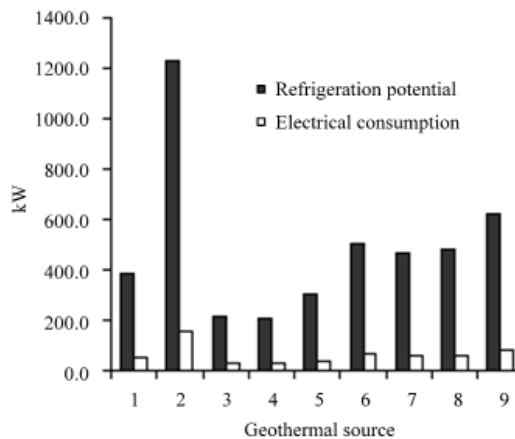


Fig. 4: Histogram of the Powers

This refrigerating potential Q_e is produced by means of a free energy contribution, provided by the geothermal source and of a paying contribution represented by the total electric power, consumed by the compressor and the pumps.

The installation efficiency is given by:

$$\xi = \frac{Q_e}{W_{co} + W_{Pel}} \quad (22)$$

Applying to the pilot drilling, the system generates a refrigerating potential of about 380 kW. If this potential is produced by the R134a vapor compression installation, working at the temperature T_c= 263 K and T_c = 308 K, it would have consumed 113.5 kW, which represents 314%, moreover than the hybrid installation.

It would have also released an additional quantity of CO₂ in the same proportions that is about 1500 tons per year (calculates as in^[9], where 1MWhour electric is equivalent to 0.283 toes and one toe releases 2400 kg of CO₂).

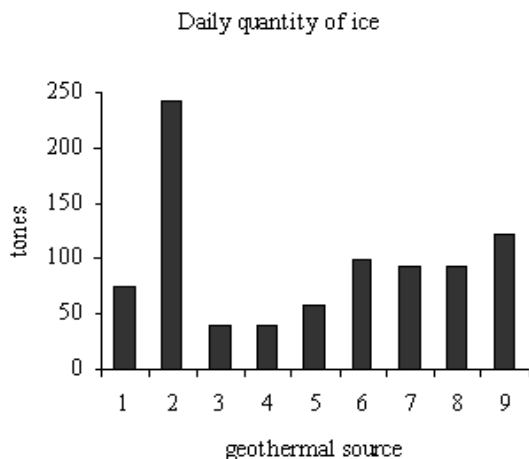


Fig. 5: Histogram of ice production

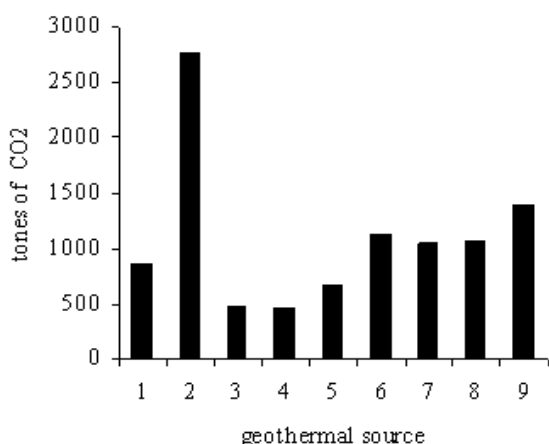


Fig. 6: Histogram of early CO₂ emissions

The study generalized to some geothermal sources of the Tunisian south, Table 2, for the same operating conditions, provides the results represented by the Fig. 4-6.

These histograms highlight the importance of the refrigerating potential (4.4 MW), awaited from these sources, the daily quantity of ice (865 tonnes) and quantity of CO₂ saved (10,000 tons).

Considering sources with temperatures higher than 343 K, this makes it possible to increase the generator temperature and consequently to lower the pressure variation between the high pressure absorber and the low pressure generator, it follows an increase in the efficiency of the hybrid installation (the actual efficiency is $\xi=8.2$).

CONCLUSION

To develop these energies at low temperature, this study proposes the use of a hybrid system with absorption/compression using couple ammonia-water, for the ice production. It shows that for fixed

temperatures T_e , T_c and T_a , this hybrid system can work with levels of generator temperature for which none the cycles suggested up to now can function.

Compared to the conventional compression refrigeration cycles, the hybrid system with absorption/compression can constitute an alternative for the reduction of the electric consumption which does not cease increasing because of the growth of the living standard in Tunisia and makes it possible to reduce the emissions of CO₂ in the same proportions.

REFERENCES

- Goktun, S., 1999. Optimal performance of an irreversible, heat engine-driven, combined vapor compression and absorption refrigerator. *Applied Energy*, 62: 67-79.
- Goktun, S., 2000. Performance analysis of a heat engine driven combined vapor compression-absorption-ejector refrigerator. *Energy Conversion and Management*, 41: 1885-1895.
- Syed, M.T. and M.A. Siddiqui, 1999. Performance and economic study of the combined absorption/compression heat pump. *Energy Conversion and Management*, 40: 575-591.
- Misra, R.D., P.K. Sahoo and A. Gupta, 2002. Application of the exergetic cost theory of the LiBr/H₂O vapor absorption system. *Energy*, 27: 1009-1025.
- Hulten, M. and T. Berntsson, 2002. The compression/absorption heat pump cycle-conceptual design improvements and comparisons with the compression cycle. *Intl. J. Refrigeration*, 25: 487-497.
- Bulgan, A.T., 1997. Use of low temperature energy sources in aqua-ammonia absorption refrigeration systems. *Energy Conversion and Management*, 38: 1431-1438.
- Duminil, M., 2003. Machines thermo frigorigifques, Systèmes à éjection et à absorption. *Techniques de l'Ingénieur BE 9 735*, pp: 1-23.
- Kara, Y.A. and B. Yuksel, 2001. Evaluation of low temperature geothermal energy through the use of heat pump. *Energy Conversion and Management*, 42: 773-781.
- Kairouani, L. and E. Nehdi, 2005. Thermodynamic Analysis of an Absorption/Compression Refrigeration System Using Geothermal Energy. *Am. J. App. Sci.*, 2: 914-919.
- Thioye, M., 1997. Amélioration de la performance des machines frigorigifques à absorption par l'utilisation de cycles à absorption et désorption étagés. *Intl. J. Refrigeration*, 20: 136-145.
- Yong, T.K., H. Hiki and S.P. Kyong, 2004. Performance analysis of advanced hybrid GAX cycles: HGAX. *Intl. J. Refrigeration*. 27: 442-448.

12. Arh, S., 1990. Gaspersic. Development and comparison of different advanced absorption cycles. *Revue Internationale du Froid*, 13: 41-50.
13. Kairouani, L., M. Hassairi and T. Zermani, 2004. Performance of cooling tower in south of Tunisia. *Building and Environment*, 39: 351-355.
14. Burghardt, M.D., 1982. *Engineering Thermodynamics with Applications*, 2nd Edn., Harper and Row, Publishers, New York, pp: 547.
15. Brunin, O., M. Feidt and B. Hivet, 1997. Comparison of the working domains of some compression heat pumps and a compression-absorption heat pump. *Intl. J. Refrigeration*, 20: 308-318.
16. Maake, W., H.J. Eckert and J.L. Cauchepin, 1993. *Le Pohlmann*, tome 1, Deuxième Edition PYC Edn, Ivry-sur-Seine, pp: 350-382.