

# SOLAR POWERED RESORPTION COOLING SYSTEM

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**Abstract:** *This paper investigates the relative performance of a thermally activated, environmentally friendly cooling system: a NH<sub>3</sub>-H<sub>2</sub>O resorption system. This system can be driving at relatively low heat source temperatures such as those achieved by solar collectors. The study explores the relative thermal performance, i.e. the performance coefficient (COP) and refrigeration capacity of the system, and a qualitative comparison based on the type of solar collector, i.e. compound parabolic collector. The geographical functioning location of the system was chosen for the city of Braşov, Romania for July.*

**Key words:** *absorption, resorption, solar refrigeration, sustainable cooling, environmentally friendly technologies.*

## 1. Introduction

Modern methods for the management of fuel and energy use in commercial and industrial fields are based on recoverable energetically resources from the technological processes and, if possible, outside it, for energy purposes such as: producing hot-water, heating, air conditioning, refrigeration etc. One promising way is the use of the solar energy.

For commercial, industrial, technological refrigeration or for comfort technological air conditioning the refrigeration plants with vapor compression, which generally use electrical energy, can be replaced with resorption refrigerating systems. These systems use directly the recoverable energy resources saving hence saving electricity.

The resorption refrigerating systems assume:

- the energetic potential of the recoverable energy resources to be bigger than the energy needed for the cooling production;
- the simultaneously existence of a heating source and cooling user.

## 2. Recoverable Resources with Low Thermal Potential

The resorption refrigeration system was designed, built and analyzed in terms of using the hot water provided from the solar compound parabolic collector located in Braşov, Romania.

The Brasov climatic data for solar irradiance and average daily temperature for July are reported in Figures 1-3 [8].

The maximum solar radiation is about 700 Wm<sup>-2</sup>, while the ambient temperature reaches its maximum of 25 °C at 14:07 h.

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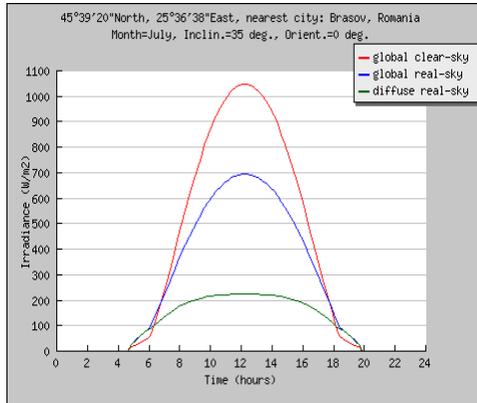


Fig. 1. Daily irradiance on a fixed plane

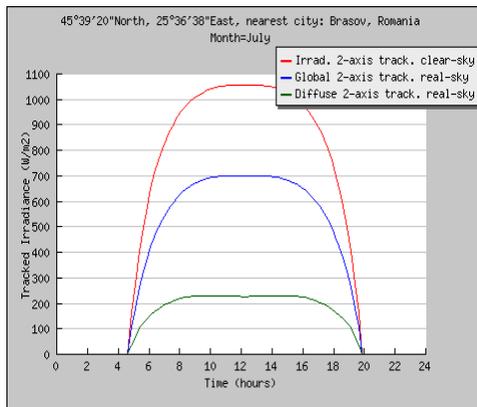


Fig. 2. Daily irradiance on a 2-axis tracking plane

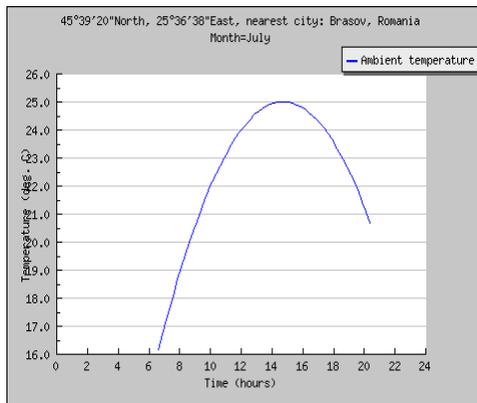


Fig. 3. Average daily temperature profile

## 2.1. The Solar Collector

The heat source consists of an enhanced compound parabolic concentrator (CPC) developed by Solarfocus-GmbH with a gross area of 2.42 m<sup>2</sup>. An increase of 20% in efficiency is claimed by the manufacturer for these new CPCs compared to classical ones [1]. The efficiency  $\eta_{SC}$  of the CPC as a function of the ambient temperature, the solar radiation and the heating medium temperature is given by manufacturer's data with the relation:

$$\eta_{SC} = 0.75 - 2.57 \cdot F - 4.67 \cdot F^2, \quad (1)$$

$$F = (T_{hw,mean} - T_{amb}) / I_{SC},$$

where  $T_{hw,mean}$  is the heat transfer fluid mean temperature:  $T_{HW} = 0.5(T_{hw,out} + T_{hw,in})$ ,  $T_{amb}$  is the ambient temperature and  $I_{SC}$  is the solar radiation. The energy balance for the solar collector is given by the following equation:

$$M_{SC} C_{SC} \frac{dT_{SC}}{dt} = \eta_{SC} I_{SC} A_{SC} + \quad (2)$$

$$+ \dot{m}_{hw} c_{p,hw} \cdot (T_{hw,in} - T_{hw,out}),$$

where  $M_{SC}$ ,  $C_{SC}$ ,  $T_{SC}$  are mass, heat capacity and temperature of the collector, and subscripts  $hw$  are related to the hot water circulated in collector, the mass flow, heat capacity, inlet and outlet temperature respectively.

## 3. Resorption Refrigeration System

A resorption refrigerating system, Figure 4, is made by two vapor generators, GV and Deg, two absorbers, Ab and Res, two heat exchangers (economizer), EC1 and EC2 and two solution circulation pumps, P1 and P2. This configuration makes the installation more voluminous and more complicated in comparison with the simple

absorption refrigerating system. The main advantage of the resorption refrigerating system is the fact that the pressures  $p_0$  and  $p_F$  from the low-pressure and high pressure

part of the system, respectively, are no longer determined by the vaporizing temperature  $t_0$ , and the condensing temperature  $t_k$ , as they are on an absorption refrigerating system.

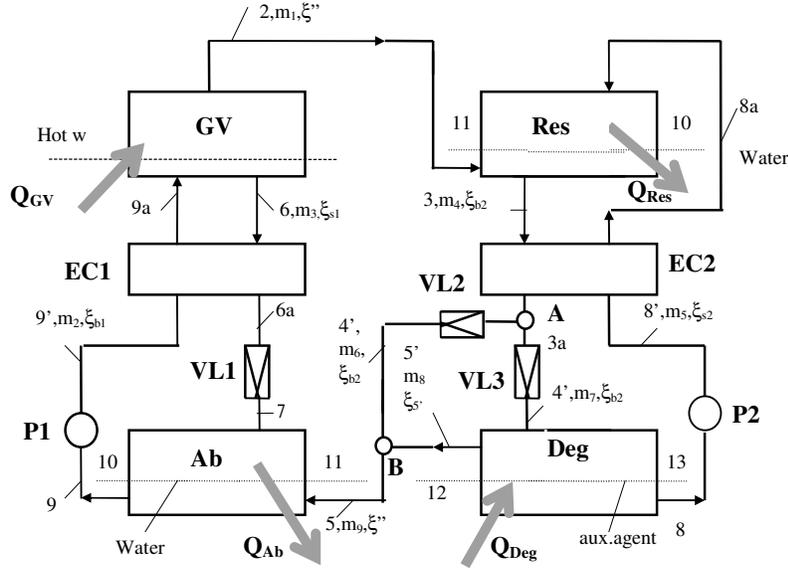


Fig. 4. *The resorption refrigeration system*

The vapor absorption processes in the resorber and the vapor producing process in the degasser can take place at any pressure values, being independent of the temperature. The pressures  $p_0$  and  $p_F$  can be chosen randomly, especially near the atmospheric pressure value. Thus, problems like tightening, dimensioning, pump construction are simplified and the thermal potential of the heating agent in the vapor generator can be reduced, making possible the use of the recoverable energy resources with a lower thermal potential than the one used on absorption refrigerating systems.

### 3.1. Thermodynamic Modeling of the Resorption Refrigeration System

The resorption refrigeration modeling is based on the energy and mass conservation equations.

These equations for resorption refrigeration are:

- energetic balance on the degasser:

$$\dot{m}_8 (h_4 - h_8) \cdot \frac{\xi_{b2} - \xi_{s'}}{\xi_{b2} - \xi_{s2}} + h_{s'} - h_4 = \dot{Q}_0; \quad (3)$$

- material inventory on the degasser:

$$\dot{m}_5 = \dot{m}_8 \cdot \frac{\xi_{s'} - \xi_{b2}}{\xi_{b2} - \xi_{s2}}; \quad (4)$$

- mass inventory on the degasser:

$$\dot{m}_7 = \dot{m}_5 + \dot{m}_8; \quad (5)$$

- energy balance on the absorber:

$$\dot{Q}_{Ab} = \dot{m}_9 \cdot h_5 + \dot{m}_3 \cdot h_7 - \dot{m}_2 \cdot h_9; \quad (6)$$

- material inventory on the absorber:

$$\dot{m}_3 = \dot{m}_9 \cdot \frac{\xi'' - \xi_{b1}}{\xi_{b1} - \xi_{s1}}; \quad (7)$$

- mass inventory on the absorber:

$$\dot{m}_2 = \dot{m}_3 + \dot{m}_9; \quad (8)$$

- energy balance on the economizer EC1:

$$\dot{m}_3(h_6 - h_{6a}) = \dot{m}_2(h_{9a} - h_9); \quad (9)$$

- energy balance on the vapor generator GV:

$$\dot{Q}_{GV} = \dot{m}_1 \cdot h_2 + \dot{m}_3 \cdot h_6 - \dot{m}_2 \cdot h_{9a}; \quad (10)$$

- material inventory on the GV:

$$\dot{m}_1 \xi'' + \dot{m}_3 \xi_{s1} = \dot{m}_2 \xi_{b1}; \quad (11)$$

- mass inventory on the GV:

$$\dot{m}_1 + \dot{m}_3 = \dot{m}_2; \quad (12)$$

- energy inventory on the resorber:

$$\dot{Q}_{Res} = \dot{m}_4 \cdot h_3 + \dot{m}_1 \cdot h_2 + \dot{m}_5 \cdot h_{8a}; \quad (13)$$

- material inventory on the resorber:

$$\dot{m}_1 = \dot{m}_5 \cdot \frac{\xi_{b2} - \xi_{s2}}{\xi'' - \xi_{b2}}; \quad (14)$$

- mass inventory on the resorber:

$$\dot{m}_4 = \dot{m}_5 + \dot{m}_1; \quad (15)$$

- energy inventory on the economizer EC2:

$$\dot{m}_4(h_3 - h_{3a}) = \dot{m}_5(h_8 - h_{8a}); \quad (16)$$

- material inventory on node A:

$$\dot{m}_6 = \dot{m}_4 - \dot{m}_7; \quad (17)$$

- material inventory on the node B:

$$\dot{m}_9 = \dot{m}_8 + \dot{m}_6. \quad (18)$$

The problem may be solved as the number of the equations is equal to the number of the unknowns:  $m_1, m_2, m_3, m_4, m_5, m_6, m_7, m_8, m_9, \xi_5, h_{6a}, h_{8a}, h_5, Q_F, Q_{Ab}$  and  $Q_{Res}$ .

This results in  $m_1 = m_2$  and  $\xi_5 = \xi''$  which are essential conditions for continuous functioning of the resorption refrigerating system.

The performance coefficient of the system is:

$$COP = \frac{Q_0}{Q_{GV}}. \quad (19)$$

The energy efficiency,  $\eta_{ex}$ , is an important criteria for performance evaluation of the resorption refrigerating system.

The exergetic efficiency is calculated via:

$$\begin{aligned} \eta_{ex} &= \frac{Ex(Q_0)}{Ex(Q_{GV})} \\ &= \frac{Q_0}{Q_{GV}} \cdot \frac{T_{amb} - T_{om}}{T_{fm} - T_{amb}} \cdot \frac{T_{om}}{T_{fm}}, \end{aligned} \quad (20)$$

where:  $T_{om}$  is the average boiling temperature of the solution in the degasser zone:  $T_{om} = 0.5(T_{om,in} + T_{om,out})$ ;  $T_{fm}$  is the average boiling temperature of the solution in the vapour generator:  $T_{fm} = 0.5(T_1 + T_6)$ .

For the resorption refrigeration system with ammonia-water solution other qualities can be calculated:

- the heating agent mass flow in the vapor generator:

$$\dot{m}_{GV} = \frac{Q_{GV}}{\Delta h} [\text{kg s}^{-1}], \quad (21)$$

where:  $\Delta h$  is the steam enthalpy variation;  
 - the absorber cooling water mass flow:

$$\dot{m}_{w,Ab} = \frac{Q_{Ab}}{c_w(T_{11} - T_{10})} [\text{kg s}^{-1}], \quad (22)$$

where:  $T_{10}$  is the initial temperature of the cooling water;  $T_{11}$  is the final temperature of the cooling water;

- the resorber cooling water mass flow:

$$\dot{m}_{w,Res} = \frac{Q_{Res}}{c_w(t_{11} - t_{10})} [\text{kg s}^{-1}], \quad (23)$$

where:  $t_{10}$  is the initial temperature of the cooling water; generally it is considered to be equal with the initial temperature of the absorber cooling water;  $t_{11}$  is the final temperature of the cooling water; considering a temperature variation in the absorber  $\Delta t_{w,Res} = 5^\circ\text{C}$ , and  $t_{11} = t_{10} + \Delta t_{w,Res}$ ;

- the degasser intermediary cooled agent mass flow:

$$\dot{m}_{a,i} = \frac{Q_0}{c_{a,i}(t_{12} - t_{13})} [\text{kg s}^{-1}], \quad (24)$$

where:  $c_{a,i}$  is the specific heat capacity of the cooling agent;  $t_{12}$  is the initial temperature of the cooling agent;  $t_{13}$  is the final temperature of the cooling agent; considering a temperature variation in the degasser  $\Delta t_{ai} = 4 \dots 6^\circ\text{C}$ , it results  $t_{13} = t_{12} - \Delta t_{ai}$ ;

- the total cooling water mass flow:

$$\dot{m}_w = \dot{m}_{w,Ab} + \dot{m}_{w,Res}. \quad (25)$$

The resorption refrigerating system calculation was made under the following conditions:

- the refrigeration power of the system  $Q_0 = Q_{deg} = 100 \text{ kW}$ ;

- the minimal boiling temperature of the ammonia-water solution in the degasser  $t_{0\min} = -10^\circ\text{C}$ ;

- the initial temperature of the absorber and resorber cooling water  $t_{wr} = 20^\circ\text{C}$ ;

- the ammonia-water solution temperature at the absorber and resorber exit  $t_9 = t_3 = 30^\circ\text{C}$ .

The following variable parameters were also considered:

- the heating agent temperature in the vapor generator and the maximum boiling temperature of the ammonia-water solution  $t_F = t_6 = +90 \dots +150^\circ\text{C}$ ;

- the boiling pressure of the hydro-ammonia solution  $p_F = 8, 9$  and  $10$  atm on the condition  $p_F < p_k$ ;

- the degassing zone  $\Delta\xi$ , determined by the temperature variation limits in the degasser,  $\Delta t_D = 4 \dots 12^\circ\text{C}$ .

#### 4. Discussions

A thermodynamic model of the  $\text{H}_2\text{O}-\text{NH}_3$  binary mixture [2], [3], [4], [6] was used in the calculations. The advantage of using this model is given by the high evaluation accuracy of the state points, compared with the common method, with the  $h-\xi$  diagram [5], [7], where the reading accuracy of the values is relative.

The calculations were made for  $p_F = 9$  atm, taking into consideration that the heating agent temperature of the vapor generator is more important than the maximum work temperature in the refrigerating system.

The performance evaluation of the resorption refrigerating system is presented for:

- the heat flow at the boiler,  $Q_{GV}$ ;

- the exergetic efficiency,  $\eta_{ex}$ ;

- the performance coefficient,  $COP$ .

According to Figure 5 the heat flux from the vapor generator,  $Q_{gv}$ , decreased when the value of the degassing zone increased and increased when the boiling temperature increased. An interesting result is the fact that, for values of boiling temperature between  $90^\circ\text{C}$  and  $105^\circ\text{C}$ , this flux presents minimum values which are

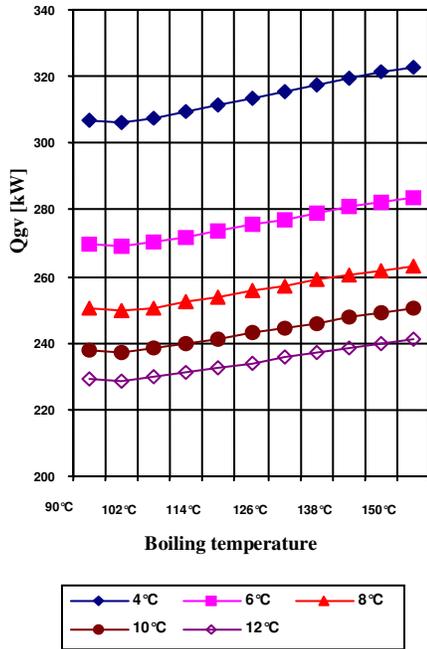


Fig. 5. The boiler heat flow as a function of the boiling temperature and the degassing zone;  $t_k = +30\text{ }^\circ\text{C}$ ,  $t_0 = -10\text{ }^\circ\text{C}$ ,  $p_F = 9\text{ atm}$ ,  $\Phi_0 = 100\text{ kW}$

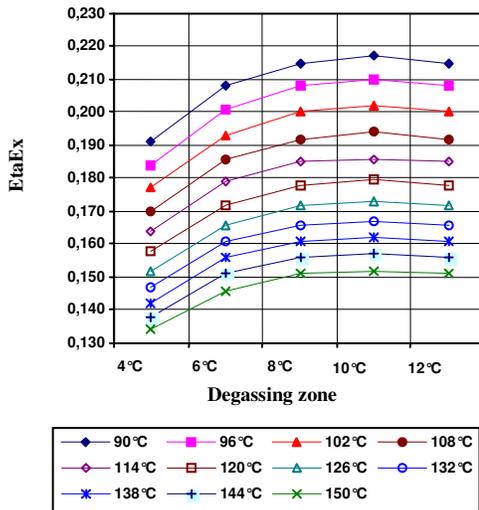


Fig. 6. Exergetic efficiency as a function of the degassing zone and the boiling temperature;  $t_k = 30\text{ }^\circ\text{C}$ ,  $t_0 = -10\text{ }^\circ\text{C}$ ,  $p_F = 9\text{ atm}$ ,  $\Phi_0 = 100\text{ kW}$

more pronounced for small values of the degassing zone ( $4\text{ }^\circ\text{C}$ ).

Figures 6 and 7 shows the influence of the degassing zone and of the boiling temperature (corresponding to the energy level of the source of the recovering heat) upon the exergetic efficiency of the frigorific cycle. It can be noticed that when  $t_F$  rises from  $90\text{ }^\circ\text{C}$  to  $150\text{ }^\circ\text{C}$ , the exergetic efficiency decreases, presenting a maximum around the value of the degassing zone  $\Delta t_D = 4\text{ }^\circ\text{C}$ .

The performance coefficient,  $COP$ , increases along with the degassing zone (Figure 8). Depending on the boiling temperature, the performance coefficient presents a maximum around the value of  $96\text{ }^\circ\text{C}$  (Figure 9), after which it uniformly decreases.

### 5. Conclusions

This paper has presented an adaptation of the resorption refrigerating installation following the thermodynamic models of the resorption refrigerating cycle and that

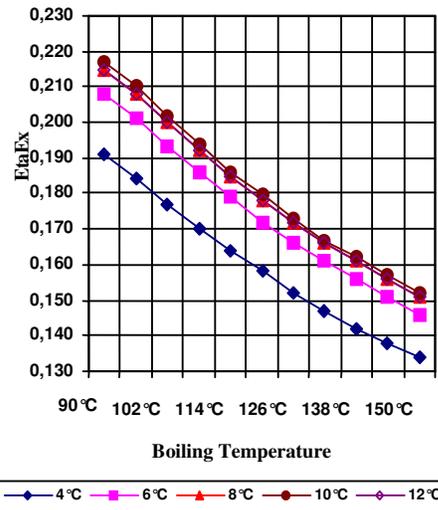


Fig. 7. Exergetic efficiency as a function of the boiling temperature and the degassing zone;  $t_k = 30\text{ }^\circ\text{C}$ ,  $t_0 = -10\text{ }^\circ\text{C}$ ,  $p_F = 9\text{ atm}$ ,  $\Phi_0 = 100\text{ kW}$

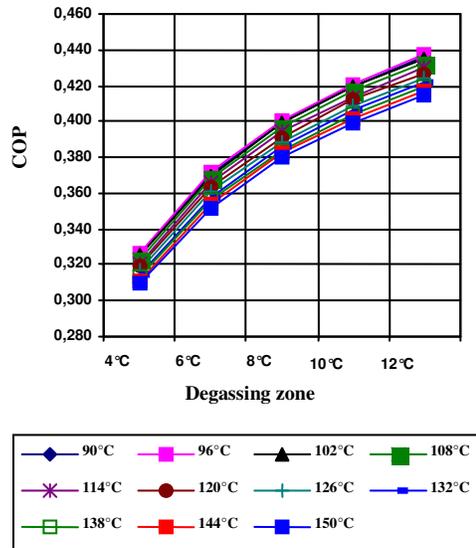


Fig. 8. COP as a function of the degassing zone and the boiling temperature;  
 $t_k = +30\text{ }^\circ\text{C}$ ,  $t_0 = -10\text{ }^\circ\text{C}$ ,  $p_F = 9\text{ atm}$ ,  
 $\Phi_0 = 100\text{ kW}$

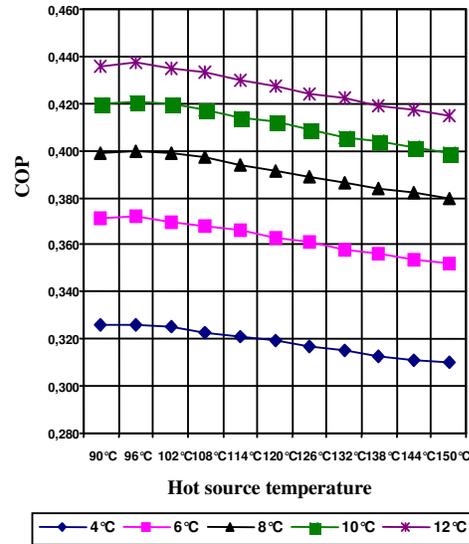


Fig. 9. COP as a function of the boiling temperature and the degassing zone;  
 $t_k = +30\text{ }^\circ\text{C}$ ,  $t_0 = -10\text{ }^\circ\text{C}$ ,  $p_F = 9\text{ atm}$ ,  
 $\Phi_0 = 100\text{ kW}$

of the  $\text{H}_2\text{O-NH}_3$  binary mixture for the purpose of improving the overall performance. The results show an improvement to the design parameters currently used when calculating these types of refrigerating installation. In addition, a minimum number of “requirements” for the refrigerating cycle have been identified in order for this to be able to work within the designed parameters and with acceptable values for the performance coefficients. The thermodynamic model of the  $\text{H}_2\text{O-NH}_3$  binary mixture, once completed, enables different types of resorption thermodynamic cycles to be studied with the purpose of improving their efficiency.

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