Bulletin of the *Transilvania* University of Braşov
Series I: Engineering Sciences • Vol. 6 (55) No. 2 - 2013

SOLAR POWERED ABSORBTION SYSTEM

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Abstract: *This paper investigates the relative performance of a thermally activated environmentally friendly cooling system: a NH_3-H_2O absorption system. This system can be activated with relatively low heat source temperatures such as those achieved by solar collectors. The study explores the relative thermal performance, i.e. the performance coefficient and refrigeration capacity of the system. The geographical functioning location of the system was chosen for the city of Braşov, Romania for July. The thermodynamic model of ammonia-water binary mixture was used in the calculations. The advantage is given by the high evaluation accuracy of the state points, compared with the enthalpy-concentration diagram.*

Key words: *absorption, solar refrigeration, simulation, environmentally friendly technology.*

1. Introduction

The management of fuel and energy use in commercial and industrial fields are based usually on electrical energy. As modern methods on energy saving and decreasing the CO_2 emissions is the use of the energy from recoverable resources from the technological processes.

Many requirements for energy use purposes such as: producing hot-water, heating, air conditioning, refrigeration etc., are based on electrical energy but one promising way is the use of the solar energy combined with heating and cooling plants.

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

The absorption refrigerating systems assume the energetic potential of the recoverable energy resources to be bigger than the energy needed for the cooling production and the simultaneously existence of a heating source and cooling user.

2. Heat Source

The absorption 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 Braşov climatic data for solar irradiance and average daily temperature for July are reported in Table 1 [8]. The maximum solar radiation is about 700 Wm^{-2}, while the ambient temperature reaches its maximum of 25 °C at 14:07 h.

In Table 1, there are the following measured variables:

G - global irradiance on a fixed plane,

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PVGIS Radiation values for July Table 1

Time	Braşov Location: 45°39'20" North, 25°36'38" East, Elevation: 564 m a.s.l., Inclination of plane: 35 deg. Orientation (azimuth) of plane: 0 deg.						
	G	G_d	G_c	A	A_d	A_c	T_d
	[W/m²]	[W/m²]	[W/m²]	[W/m²]	[W/m²]	[W/m²]	[°C]
04:52	28	28	16	105	33	165	-
05:52	83	82	46	378	139	580	-
06:52	202	127	210	530	187	807	16.7
07:52	348	170	440	618	215	933	18.7
08:52	482	199	663	667	228	1000	20.4
09:52	587	214	850	690	231	1040	21.8
10:52	658	221	981	699	229	1050	23.0
11:52	692	223	1040	701	227	1060	23.9
12:52	687	223	1030	701	227	1060	24.5
13:52	644	220	954	698	230	1050	24.9
14:52	564	211	808	686	231	1030	25.0
15:52	451	193	610	657	226	989	24.8
16:52	312	161	381	600	210	908	24.4
17:52	166	114	156	499	177	762	23.7
18:52	70	69	39	326	122	498	22.7
19:52	0	0	0	0	0	0	21.4
20:37	0	0	0	0	0	0	20.3

G_d - diffuse irradiance on a fixed plane,

G_c - global clear-sky irradiance on a fixed plane,

A - global irradiance on 2-axis tracking plane,

A_d - diffuse irradiance on 2-axis tracking plane,

A_c - global clear-sky irradiance on 2-axis tracking plane,

T_d - average daytime temperature profile.

The heat source consists of an enhanced compound parabolic concentrator (CPC) developed by Solarfocus-GmbH with a gross area of 2.42 m². The efficiency η_{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 [1]:

$$\eta_{SC} = 0.75 - 2.57 \cdot (T_{hw,mean} - T_{amb}) / I_{SC} - 4.67 \cdot \left((T_{hw,mean} - T_{amb}) / I_{SC}\right)^2, \quad (1)$$

where $T_{hw,mean}$ is the heat transfer fluid mean temperature: $T_{hw,mean} = 0.5(T_{hw,out} + T_{hw,in})$, 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:

$$M_{SC} C_{SC} \frac{dT_{SC}}{dt} = + \eta_{SC} I_{SC} A_{SC} + \dot{m}_{hw} c_{p,hw} \cdot (T_{hw,in} - T_{hw,out}), \quad (2)$$

where M_{SC}, C_{SC}, T_{SC} are the 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. Absorption Refrigeration System

An absorption refrigerating system (ARS), Figure 1, is made by a vapor Generator, an Absorber, Heat exchanger (economizer), an Evaporator and solution circulation Pump. This configuration makes the installation more voluminous in comparison with the simple compression refrigerating system. The main advantage of the absorption refrigerating system is the fact that the moving parts are involved just in Pump parts. The other absorbtion plant components are not containing such as moving parts. In fact, that means the maintenance of the installation is much more facile on long term period.

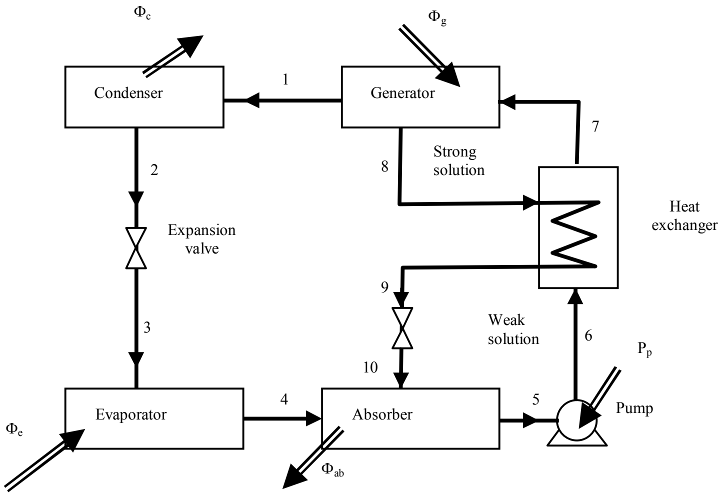

Fig. 1. *Schematic of absorption refrigeration plant*

The vapor absorption processes in the Absorber and the vapor producing process in the Generator can take place at pressure values, being dependent of the temperature. The pressures p0 and pF can be calculated from thermodynamic properties at saturation values for the NH_3-H_2O mixture.

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.

4. Thermodynamic Modelling of ARS

The absorption refrigeration modeling is based on the energy and mass conservation equations. In order to analyze the absorption refrigeration system, mass, component and energy balance must be performed for each system part like below.

For evaporator:

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_{ref}\,, \tag{3}$$

$$\Phi_e = \dot{m}_{ref}(h_4 - h_3)\,. \tag{4}$$

For the expansion valves:

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_{ref}\,,\ h_2 = h_3\,, \tag{5}$$

$$\dot{m}_9 = \dot{m}_{10}\,,\ h_9 = h_{10}\,. \tag{6}$$

For the generator:

$$\dot{m}_7 = \dot{m}_1 + \dot{m}_8\,, \tag{7}$$

$$\dot{m}_7 x_7 = \dot{m}_1 x_1 + \dot{m}_8 x_8\,, \tag{8}$$

$$\Phi_g = \dot{m}_1 h_1 + \dot{m}_8 h_8 - \dot{m}_7 h_7\,. \tag{9}$$

From the Equations (7) and (8), the strong solution and the weak solution mass flow rate can be obtained:

$$\dot{m}_8 = \frac{x_7 - x_1}{x_8 - x_7}\dot{m}_1, \quad (10)$$

$$\dot{m}_7 = \frac{x_8 - x_1}{x_8 - x_7}\dot{m}_1. \quad (11)$$

From Eq. (11), the circulation ratio can be obtained:

$$f = \frac{\dot{m}_7}{\dot{m}_1} = \frac{x_8 - x_1}{x_8 - x_7}. \quad (12)$$

For the absorber:

$$\dot{m}_4 + \dot{m}_{10} = \dot{m}_5, \quad (13)$$

$$\Phi_a = \dot{m}_4 h_4 + \dot{m}_{10} h_{10} - \dot{m}_5 h_5. \quad (14)$$

Dividing by $\dot{m}_4$, it results that:

$$\dot{q}_a = (h_4 - h_{10}) + f(h_{10} - h_5), \quad (15)$$

where $\dot{q}_a$ represents the heat dissipated per unit mass, and f the mass flow ratio. The first term of the right side represents the phase change and the second the cooling of the mixture.

For the pump:

$$\dot{m}_5 = \dot{m}_6, \quad (16)$$

$$P_p = \dot{m}_5 (h_6 - h_5). \quad (17)$$

For the condenser:

$$\dot{m}_1 = \dot{m}_2, \quad (18)$$

$$\Phi_c = \dot{m}_{ref}(h_1 - h_2). \quad (19)$$

For strong solution-weak solution heat exchanger:

$$\dot{m}_8 + \dot{m}_6 = \dot{m}_7 + \dot{m}_9, \quad (20)$$

$$h_7 = h_6 + \frac{\dot{m}_8}{\dot{m}_6}(h_8 - h_9). \quad (21)$$

The performance coefficient of the system is:

$$COP = \frac{\Phi_e}{\Phi_g}. \quad (22)$$

The energy efficiency, η_{ex}, is an important criteria for performance evaluation of the absorption refrigerating system. The exergetic efficiency is calculated as:

$$\eta_{ex} = \frac{Ex(\Phi_e)}{Ex(\Phi_g)} = \frac{\Phi_e}{\Phi_g} \cdot \frac{T_{amb} - T_{om}}{T_{Fm} - T_{amb}} \cdot \frac{T_{om}}{T_{Fm}}, \quad (23)$$

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

5. Results

A thermodynamic model of the H_2O-NH_3 binary mixture [2-4], [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-ξ diagram [5], [7], were the reading accuracy of the values is relative.

The calculations were made for condenser temperature t_{cond} = +30 °C, 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 exergetic efficiency, η_{ex} and the performance coefficient, COP.

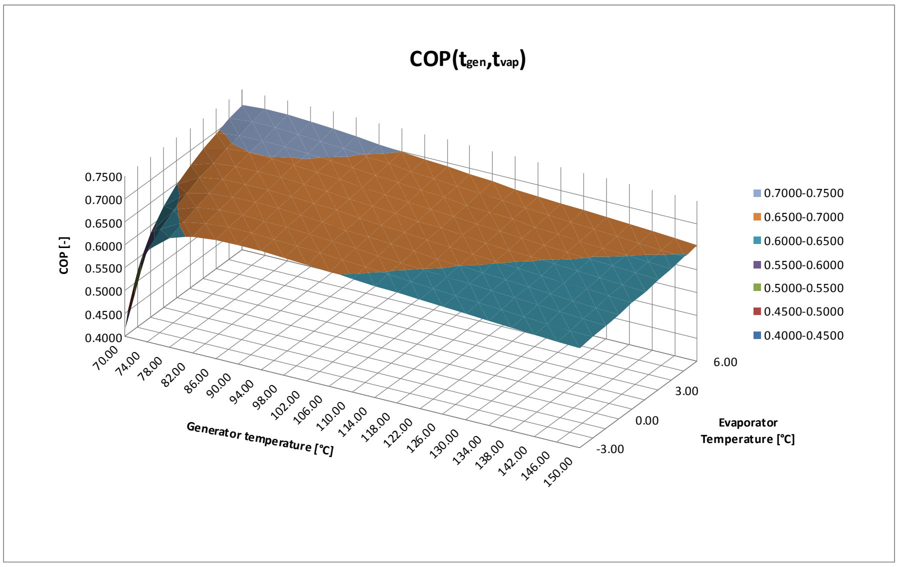

Fig. 2. *The COP in function of generator and evaporator temperatures*

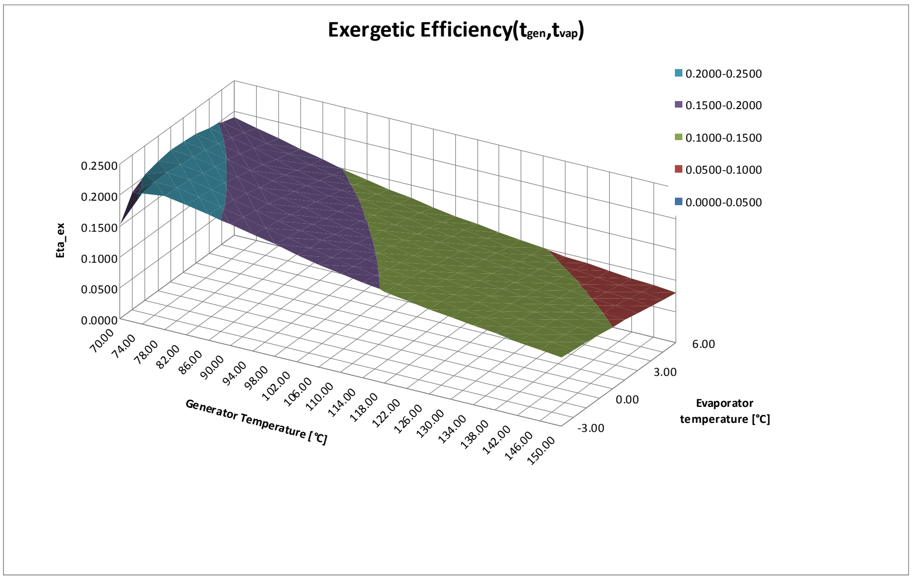

Fig. 3. *The exergetic efficiency in function of generator and evaporator temperatures*

According to simulation, the heat flux from the vapor generator, Φ_g, is decreasing when the value of the evaporator temperature is increasing and is increasing when the boiling temperature increased in the temperature range greater than 90 °C for generator. An interesting result is the fact that, for values of boiling temperature between 90 °C and 105 °C, this flux presents minimum values which are more pronounced for small values of the evaporator temperature, values below -1 °C.

In Figure 2 and 3 are shown the influence of the evaporator temperature and of the boiling temperature, corresponding to the energy level of the source of the recovering heat, upon the exergetic efficiency of the refrigeration cycle. It can be noticed that when the boiling temperature t_{gen} rises from 70 °C to 150 °C, the exergetic efficiency decreases for the evaporator temperature t_{vap} = +6 °C, in despite of the variation for the evaporator temperature t_{vap} = -3 °C where the exergetic efficiency have a maximum at t_{gen} = 78 °C.

The performance coefficient, COP, depending on the boiling temperature presents a maximum around the value of 90 °C (Figure 2) for the evaporator temperature t_{vap} = -3 °C, after which it uniformly decreases. The behavior is the same for the evaporator temperature t_{vap} = +6 °C, whith a maximum value, but at lower boiling temperature which is about t_{gen} = 74 °C.

6. Conclusions

This paper has presented an adaptation of the absorption refrigerating installation following the thermodynamic model of the absorption refrigerating cycle and that of the H_2O-NH_3 binary mixture with a solar system with CPC 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.

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