THEORETICAL STUDY ON SOLAR POWERED ABSORPTION COOLING SYSTEM

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Abstract. This paper investigates the relative performance of a thermally activated environmentally friendly cooling system: a NH3-H2O sorption 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 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. Firstly, the solar system collectors was investigated: the efficiency of the solar collectors as a function of the ambient temperature, the solar radiation and heating medium temperature, daily irradiance on a fixed plane, daily irradiance on a 2-axis tracking plane, average daily temperature profile. The absorption system is analyzed. The mathematical model developed is based on the energy and mass conservation equations. The thermodynamic model of ammonia-water binary mixture was used in the calculations. The advantage of this model is given by the high evaluation accuracy of the state points, compared with the common method, with the enthalpy-concentration diagram, where the reading accuracy of the values is relative. 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 ammonia-water binary mixture, once completed, enables different types of absorption thermodynamic cycles to be studied with the purpose of improving their efficiency.

Keywords: absorption, solar refrigeration, sustainable cooling, 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 CO2 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. RECOVERABLE RESOURCES WITH LOW THERMAL POTENTIAL

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 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$^{-2}$, while the ambient temperature reaches its maximum of 25°C at 14:07 h.

3. 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$^2$. An increase of 20% in efficiency is claimed by the manufacturer for these new CPCs compared to classical ones. 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\(^1\):

\[
\eta_{SC} = 0.75 - 2.57 \cdot F - 4.67 \cdot F^2
\]

whith

\[ F = (T_{hw,\text{mean}} - T_{\text{amb}}) / I_{SC} \]

where \( T_{hw,\text{mean}} \) is the heat transfer fluid mean temperature; \( T_{HW} = 0.5(T_{hw,\text{out}} + T_{hw,\text{in}}) \), \( T_{\text{amb}} \) – the ambient temperature and \( I_{SC} \) is the solar radiation.

The energy balance for the solar collector is given by the following equation:

\[
M_{SC}C_{SC} \frac{dT_{SC}}{dt} = + \eta_{SC}I_{SC}A_{SC} + \dot{m}_{hw}C_{p, hw} \cdot (T_{hw,\text{in}} - T_{hw,\text{out}})
\]

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.

### 4. ABSORPTION REFRIGERATION SYSTEM

A absorption refrigerating system (Fig. 4) 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 refrigeration system. The main advantage of the absorption refrigeration system is the fact that the moving parts are involved just in Pump parts. The other sorption 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.

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 \( p_0 \) and \( p_F \) can be calculated from thermodynamic properties at saturation values for the NH\(_3\)-H\(_2\)O 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.

5. THERMODYNAMIC MODELING OF THE ABSORPTION REFRIGERATION SYSTEM

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:

\[
\dot{m}_7 = \dot{m}_1 + \dot{m}_8 \tag{7}
\]

\[
\dot{m}_7x_7 = \dot{m}_1x_1 + \dot{m}_8x_8 \tag{8}
\]

\[
\Phi_g = \dot{m}_1h_1 + \dot{m}_8h_8 - \dot{m}_7h_7 \tag{9}
\]

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

\[
\dot{m}_8 = \frac{x_7 - x_1}{x_8 - x_7}\dot{m}_1 \tag{10}
\]

\[
\dot{m}_7 = \frac{x_8 - x_1}{x_8 - x_7}\dot{m}_1 \tag{11}
\]

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

\[
f = \frac{\dot{m}_7}{\dot{m}_1} = \frac{x_8 - x_1}{x_8 - x_7} \tag{12}
\]

For the absorber:

\[
\dot{m}_4 + \dot{m}_{10} = \dot{m}_5 \tag{13}
\]

\[
\Phi_a = \dot{m}_4h_4 + \dot{m}_{10}h_{10} - \dot{m}_5h_5 \tag{14}
\]

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

\[
\dot{q}_a = (h_4 - h_{10}) + f(h_{10} - h_5) \tag{15}
\]

where: \( \dot{q}_a \) represents the heat dissipated per unit mass; \( f \) – 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:

\[
\dot{m}_5 = \dot{m}_6 \tag{16}
\]

\[
P_p = \dot{m}_5(h_6 - h_5) \tag{17}
\]

For the condenser:

\[
\dot{m}_1 = \dot{m}_2 \tag{18}
\]

\[
\Phi_c = \dot{m}_{ref}(h_1 - h_2) \tag{19}
\]

For strong solution-weak solution heat exchanger:

\[
\dot{m}_8 + \dot{m}_6 = \dot{m}_7 + \dot{m}_9 \tag{20}
\]

\[
h_7 = h_6 + \frac{\dot{m}_8(h_8 - h_9)}{\dot{m}_6} \tag{21}
\]

The performance coefficient of the system is:

\[
COP = \frac{\Phi_e}{\Phi_g} \tag{22}
\]

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

\[
\eta_{ex} = \frac{Ex(\Phi_e)}{Ex(\Phi_g)} = \frac{\Phi_e}{\Phi_g} \cdot \frac{T_{amb} - T_{am}}{T_{bm} - T_{amb}} \tag{23}
\]

where: \( T_{am} \) is the average temperature of the solution in the evaporator: \( T_{am} = 0.5(T_{am, in} + T_{am, out}) \), \( T_{bm} \) – the average boiling temperature of the solution in the vapour generator: \( T_{bm} = 0.5(T_1 + T_6) \).

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

- the minimal boiling temperature of the ammonia-water solution in the evaporator \( T_{min} = -3 \, ^\circ C \);
– the initial temperature of the absorber cooling water \( t_{wv} = +30 ^\circ C \);
– the ammonia-water solution temperature at the absorber and condenser exit \( t_0 = t_3 = +30 ^\circ C \);
– the refrigeration power of the system \( \Phi_e = 100 \text{ kW} \).

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 \( T_v = T_6 = +70...+150 ^\circ C \);
– the boiling temperature of the ammonia-water solution in the evaporator \( T_0 = -3...+6 ^\circ C \);
– the refrigeration power of the system \( \Phi_g \).

6. DISCUSSIONS

A thermodynamic model of the H2O-NH3 binary mixture \([3],[4],[6]\) 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 \([5],[7]\), were the reading accuracy of the values is relative.

The calculations were made for condenser temperature \( t_{\text{cond}} = +30 ^\circ C \), 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 heat flow at the boiler, \( Q_{gi} \); the exergetic efficiency, \( \eta_{ex} \); the performance coefficient, \( COP \).

![Fig. 5. Generator heat flux vs. boiling temperature and evaporator temperature.](image)

According to Figure 5 the heat flux from the vapor generator, \( \Phi_g \), 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.

![Fig. 6. The exergetic efficiency, \( \eta_{ex} \) and the performance coefficient, \( COP \) in function of boiling temperature in generator and at evaporator temperature \( t_{vap} = -3 ^\circ C \).](image)

Figures 6 and 7 show 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 frigorific 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 \( t_{vap} = +6 ^\circ C \), in despite of the variation for the evaporator temperature \( t_{vap} = -3 ^\circ C \) where the exergetic efficiency have a maximum at \( t_{gen} = 78 ^\circ C \).

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

7. CONCLUSION

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.

REFERENCES