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Numerical Simulation of a Refrigeration Installation Operated in Climatic Conditions of Southwest Algeria

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Abstract

In this work, analytical calculations of the energy balance and numerical simulations by Trnsys_16 of a solar absorption machine are presented. The NH_3 / H_2O pair was chosen as the operating fluid under the climatic data for southwestern Algeria. The detailed dynamic simulation model of the solar installation and the machine components are developed for different operating conditions in order to predict the performance of the system studied. The results obtained show that the coefficient of performance COP of the simulated machine, its value is 1.79 corresponds to a condensing temperature of 44°C.

Keywords: Solar air conditioning; Energy balance; Simulation; Trnsys_16

Introduction

The binary ammonia-water couple has been the main solution in the operation of absorption refrigeration machines for several years. Several studies have been carried out on the vapor-liquid equilibrium and on the thermodynamic and transport properties of this mixture.

On the various cold production processes, the system considered in this study is the absorption system, it uses the ammonia-water couple. In the operation of an absorption-diffusion machine, a support gas is used which balances the pressures between the condenser and the evaporator while allowing the evaporation of the refrigerant and therefore the production of cold. For this, an H_2O separator is used which makes it possible to reinforce the reliability of these systems while maintaining their operating flexibility. This type of separator is used in diffusion machines whose operation is based on the fact that the total pressure is the same throughout the device, the difference between the partial pressures being compensated by the introduction of inert gas.

This pressure difference ensures circulation between the evaporator and the absorber. The refrigerant in the form of vapor diffuses into the inert gas and circulates from the evaporator to the absorber. The heavy gases descend to the absorber where the solute is absorbed; the inert gas then rises to the evaporator via an exchanger. The presence of this inert gas implies, during absorption and evaporation, resistance in the gas phase: the diffusion in the inert gas constitutes the main resistance to the transfer of matter. Over the years, a number of researchers have studied and described the performance of various absorption-diffusion cycles, graphically, experimentally, and numerically.

Reistad [1], proposed a graphical method for the calculation of concentrations, thermal capacities, and cycle temperatures, applying the enthalpy-concentration diagram. Chen et al. [2], designed a new generator including an exchanger that reuses waste heat for cycle rectification. The new cycle configuration demonstrated a significant improvement in COP (5%) compared to the original generator configuration. S.A Akam et al. [3] carried out an experimental study of an absorption-diffusion cooling loop. The experimental results are obtained using a test bench for two heating modes: electric power and butane gas. They concluded that in both heating modes, there is no problem with the operation of the machine and the COP values are higher in the case of electric heating than in the case of butane gas heating.

Another study of the absorption-diffusion cycle, using helium as an inert gas, has been presented by Srikhirinet et al. [4]; mass and energy balances were applied for each element of the cycle, the COP varies in the range 0.09 - 0.15. Maiya [5] presented a simulation of the NH_3-H_2O-H_2 absorption-diffusion cycle, which showed that the use of helium is more important than that of hydrogen, although it requires a larger size of propulsion due to its greater viscosity. This study proved that a higher operating pressure causes a decrease in COP. Zohar et al. [6] have developed a thermodynamic model for the simulation of an NH_3-H_2O-H_2 absorption-diffusion refrigeration cycle and obtained numerical results using the EES (Engineering Equation Solver) software. This study shows that the best COP was obtained for a concentration of the rich solution varying between 0.25 and 0.3, for generator temperatures varying from 195 to 205°C, the recommended values for the concentrations of the rich and lean solutions are respectively 0.3 and 0.1. Helium has been found to be preferable to hydrogen as an inert gas for the cycle. The COP of a system running on helium is higher (up to 40%) than the same system running on hydrogen.

Makhlouf M et al. [7] presented a comparison between the model chosen by Zohar et al. [6], for the calculation of the thermodynamic properties of the binary couple NH_3-H_2O (model of B. Ziegler et al. [8]) and the model chosen for the study. This study allowed a clear improvement in the COP of the absorption-diffusion cycle. The gain obtained between this study and Zohar et al is 0.01038 (COP depends on the concentration of the rich solution and the temperature of the generator), is 0.00602 (COP depending on the temperature of the generator and the concentration of the solution poor).

Zohar et al. [9] Made another study on the absorption diffusion machine (DAR) which uses fluid circulating inside, a binary solution from which the refrigerant, the absorbent, and the inert gas. The commercial refrigerating machine generally operates with the ammonia solution (water-ammonia) and as inert gas hydrogen or helium. The digital study focuses on the performance of the simplified DAR system which operates with an organic absorbent and five other different refrigerants as well as helium as an inert gas.

The properties of the mixture (ammonia-water) have been studied and evaluated for over 100 years. To achieve this goal, a large number of experimental research has been carried out the results of which have been materialized in a series of tables and diagrams with useful data. Enick et al [10] used the Peng-Robinson equation of state to predict the thermodynamic properties of the mixture. Weber [11] presented a model to estimate the second and third coefficients of the NH_3-H_2O mixture. Rukes & Dooley [12] employed the calculation of the Helmotz free energy of the components for the thermodynamic properties at saturation.

The formulation of thermodynamic properties of the ammonia-water mixture has been used by Ziegler & Trepp [13]; Ibrahim & Klein [14], Xu & Goswami [15] and Jordan [16], and Souad Himoun [17]. Each researcher has formulated the vapor-liquid equilibrium conditions with different means, Ziegler and Trepp (1984), Nag & Gupta [18], and Xu & Goswami [15] formulated it by equalizing phagocytes of the components in the two phases. In this work, we anticipate an analytical calculation of the energy balance and numerical simulations by Trnsys_16 of the solar absorption machine using the couple (NH_3 / H_2O) as the working fluid under the climatic data of the southwest of Algeria. The detailed dynamic simulation model of the solar installation and the machine components was developed for different operating conditions and compared with the measurement data in order to predict the performance of the system studied.

Method

The modelling is based on the laws of heat and mass conservation at the level of each element, to which we add the equilibrium equations of the NH_3-H_2O mixture, the enthalpy at every point of the machine cycle. The operating principle of this solar air conditioning configuration is relatively similar to a conventional compression system. Indeed, the process uses a condenser and an evaporator. There are also high pressure and low-pressure zone. The difference comes from the fact that compression is not mechanical but results from the absorption/desorption phenomenon.

Description of operation

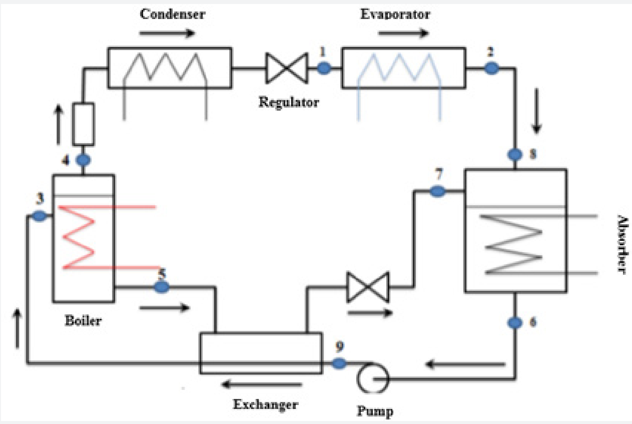

Figure 1: Model studied.

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In the evaporator (Figure 1), the refrigerant passes into vapor form, thus cooling the building's cooling water. The refrigerant in vapor form will go to the absorber or it will be absorbed by the absorbent. It should be noted that the absorption reaction takes place better at low temperatures. A water circuit therefore cools the absorbent.

The absorbent saturates with refrigerant and is then directed to the generator. The latter is heated by a heat source, which can be thermal solar panels, a cogeneration unit, etc. The high temperature of this reactor causes the disruption of the refrigerant, which then passes into vapor form. The absorbent returns to the absorber via a heat exchanger, allowing both to cool it, but also to preheat the saturated absorbent going to the generator. The refrigerant will go to the condenser, which is cooled by the same circuit as the absorber. It passes into liquid form and goes to the evaporator for a new cycle.

Energy balance

According to the Scheme (Figure 1), the energy balance was carried out on each component exchanging heat or work with the outside environment.

At the level of the boiler

$$\Phi_b = q_m \times Q_b \qquad (1)$$

with:

$$q_m = \Phi_0 / \Delta H_0 \qquad (2)$$

$$Q_b = (H_4 - H_5) + f(H_5 - H_3) \qquad (3)$$

$$f = 1 - X_P / X_r - X_p \qquad (4)$$

with:

a) H_1 = 363.5 kJ/kg

b) H_2 = H8 = 486.2 kJ/kg

c) H_3 = 232 kJ/kg

d) H_4 = 682 kJ/kg

e) H_5 = 560 kJ/kg

f) Φ_0= 46000 fg/h

g) X_p= 15%

h) X_r = 75

i) f = (1 - 0.15) / (0.75 – 0.15) = 1.41 (1kg of the rich solution / 1kg of the refrigerant)

j) Q_b = (682 - 560) + 1.41 (560 - 332) = 443.48 kJ/kg

k) q_m = 46 × 103 × 4.18 (486.2 – 363.5) = 0.43 kg/s

l) Φ_b = q_m × Q_b = 0.43 × 443.48 = 190.69 KW

At the level of the absorber

$$\Phi_a = q_m \times Q_a \qquad (5)$$

$$Q_a = (H_8 - H_7) + f(H_7 - H_6) \qquad (6)$$

with:

a) H_6 = 320 kJ/kg

b) H_7 = 475 kJ/kg

c) H_8 = 486.2 kJ/kg

d) Q_a = (486.2 - 475) + 1.41 (475 - 320) = 229.75 kJ/kg

e) Φ_a = 0.43 × 229.75 = 98.77 KW

At the level of the pump

$$\Phi_p = q_m \times Q_p \qquad (7)$$

$$Q_p = f \Delta H_p \qquad (8)$$

with:

a) H_6 = 320 kJ/kg

b) H_9 = 325.5 kJ/kg

c) Q_p = 1.41 (325.5 - 320) = 7.75 kJ/kg

d) Φ_p = q_m × Q_p = 0.43 × 7.75 = 3.33 KW

At the level of the condenser

$$\Phi_b + \Phi_a + \Phi_p + \Phi_a + \Phi_c <=> \Phi_c = \Phi_0 + \Phi_b - \Phi_a \qquad (9)$$

with:

Φ_c = (46000 × 4.18/3600 + 190.69 + 3.33) - 98.77 = 148.66 KW

Energy balances set conditions

$$\Phi_b + \Phi_0 + \Phi_p - \Phi_a - \Phi_c \qquad (10)$$

Coefficient of thermal performance of the installation (COP)

$$COP = Q_0 / Q_b \qquad (11)$$

$$D'ou :'$$

$$COP = T_0 / T_b (T_b - T_c) / (T_c - T_0) \qquad (12)$$

with:

a) T_0 = 4°C

b) T_b= 101°C

c) T_c = 44°C

Which implies that:

COP = 1.79

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Simulation of the Study Model

The purpose of establishing the thermal balance of an installation is to determine the cooling capacity necessary to ensure proper operation of the installation in accordance with the established program; the other constituent elements of the installation are then calculated according to the cooling capacity and the operating conditions of the installation. The amount of total heat removed maintains the desired temperature level. This essential part allowed us to know the thermal balances as well as all the parameters that go into our calculation program, in order to start our simulation using the Trnsys_16.

Figure 2 represents the modeling of the absorption refrigeration machine powered by solar energy by dynamic simulator (Trnsys_16), for making use of the following types:

a) File of the climatic conditions of the study area

b) Flow meter

c) Calculator

d) Two types of deviser

e) Three types of mixer

f) Two types of pump

g) Sensor

h) Storage tank

i) Boiler

j) Condenser

k) Evaporator

l) Absorber

m) Heat exchanger

n) Two types for plotting the results

o) solar captor

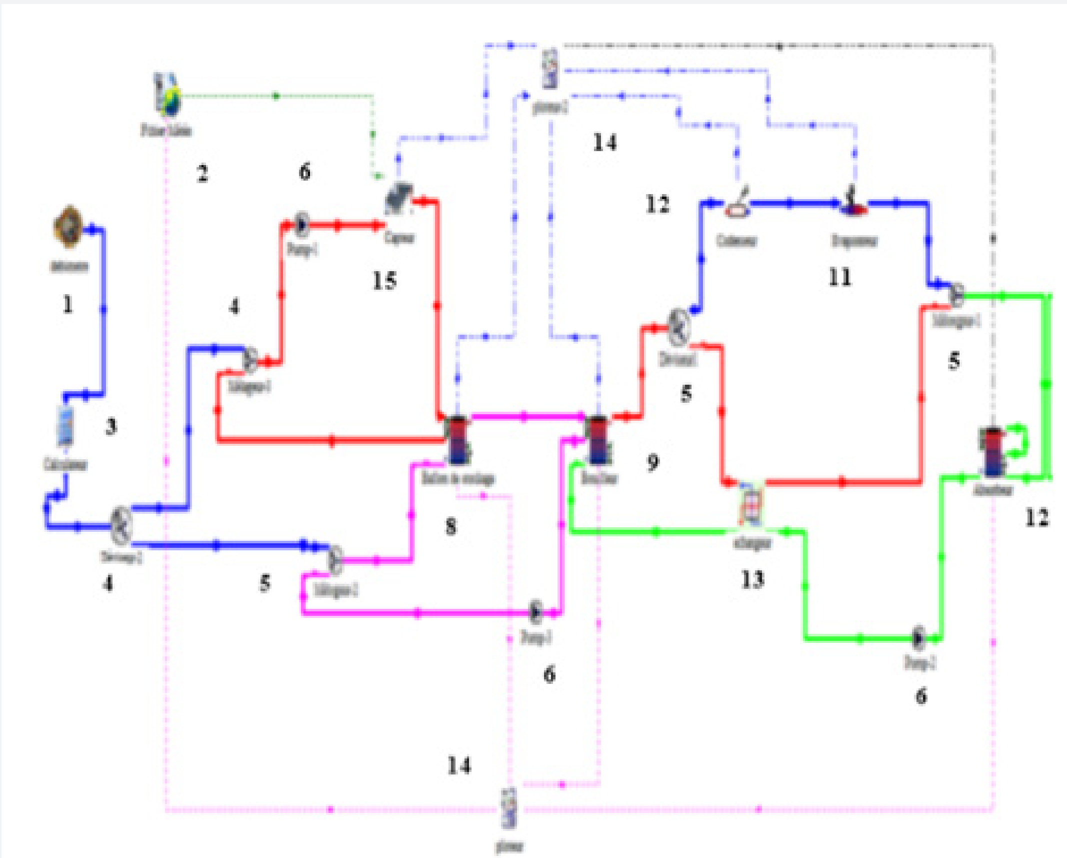

Figure 2: Simulation model for the study area.

The two Figure 3 represent the temperatures of the fluid at the level of the components of the machine during a week of simulation for a surface of 6m^2 of the collection as a function of the climatic conditions of the study area. Note at the time of capture a phase shift at the three outlet temperatures, the greatest phase shift is due to the conservation of the amount of heat during condensation. The fluid is completely heated the outlet temperature and, the evaporator is stable.

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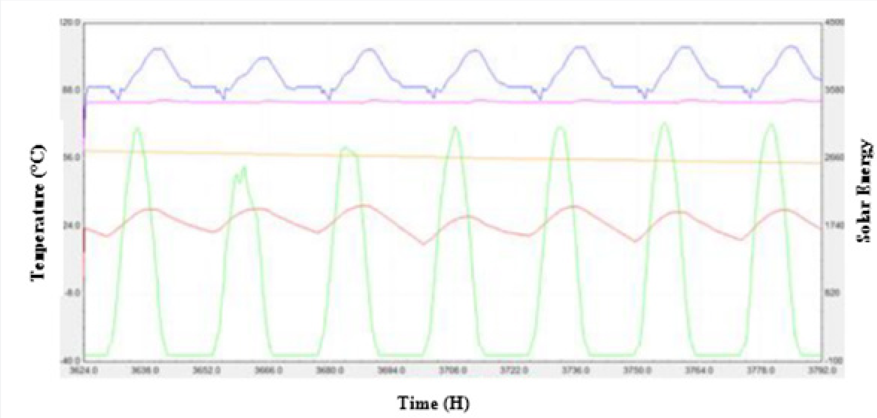

Figure 3: Fluid load temperatures at the machine organs during a simulation week vs. climatic condition (Solar energy and ambient temperature of the study area).

In the Figure 4, we study the variation of the outlet temperature of the sensor as a function of the collecting surface during a week, where from the economic point of view the outlet temperature of the sensor is high even in a small area of the sensor. The four curves are almost superimposed so the effect of increasing the catchment area is almost negligible (Figure 5).

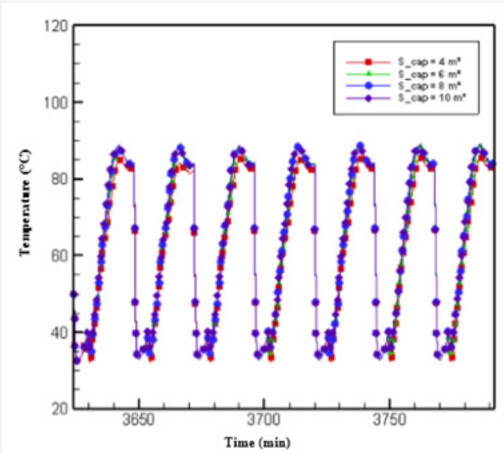

Figure 4: Sensor output temperatures as a function of simulation time.

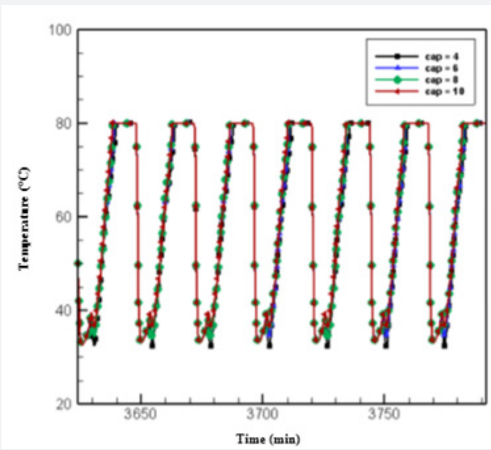

Figure 5: Boiler output temperature based on simulation time.

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The increase in the normal catchment area (Figure 6) has no effect on the boiler outlet temperature. This leads us to propose an increase in the catchment area to enhance the increase in storage volume in parallel with user needs. The coefficient of performance (COP) of the simulated absorption machine is shown in Figure 6, its value is 1.79. This value is identical to the value calculated by the energy balance of the real machine, noting that there are two regimes, the first is a transient regime at the start of the machine influenced by the initial state of the system, the second is a steady-state represents the normal machine operation, influenced by the inertial effect of the system.

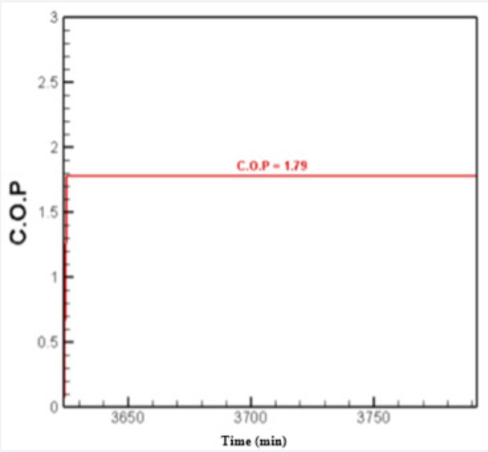

Figure 6: The coefficient of performance of the simulated machine.

Conclusion

The calculation of the energy balance and the simulations presented during this work allow us to draw the following conclusions: The analytical and numerical results obtained show that the COP of the absorption of the system is approximately 1.79 corresponds to a condensing temperature of 44°C. The increase in the normal catchment area has no effect on the boiler outlet temperature. The increase in the storage volume leads to a decrease in the outlet temperature of the storage tank. This leads us to propose an increase in the catchment area to enhance the increase in storage volume in parallel with user needs. The results obtained for the study area seem interesting, which saves energy when using such a system.

Nomenclature

q_m: Mass flow (kg/s)

Q_b: The amount of heat exchanged with the power source (kJ/kg)

Q_a: The amount of heat exchanged at the absorber (kJ/kg)

Q_p: Mechanical work exchanged at the pump (kJ/kg)

ΔH_p: The difference in entropy created by the pump (Kcal/kg)

ΔH_0: The difference in enthalpy between the inlet and the outlet of the evaporator (Kcal/kg)

T_0: The temperature in the evaporator (°C)

T_b: The temperature in the boiler (°C)

T_c: The temperature in the condenser (°C)

Φ_a: The heat at the absorber (Kcal/h)

Φ_b: The heat provided at boiler (K cal/h)

Φ_c: Heat at the condenser (Kcal/h)

Φ_p: The pump work (Kcal/h)

Φ_0: The refrigeration capacity (Kcal/h)

f: Mass flow rate of the solution (kg)

X_p: The rate of the poor solution

X_r: The rate of the rich solution

COP: Coefficient of performance.

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