Original Research Paper

Numerical Investigation of Silica Gel-Water Solar Adsorption Cooling System with Simulink

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Abstract: This paper aims to study a numerical model of an adsorption chiller driven with solar energy that is confronted to experimental measurements. This article deals with numerical study of refrigeration systems using a pair of silica gel and water with simulink. The results include the temperature of desorber, the temperature of chilled, cooling and hot water, temperature of two adsorption/desorption beds, condenser and evaporator, the refrigerant saturation pressures in the two compartments and the time variation of water uptake in two adsorbent beds.

Keywords: Solar Adsorption Chiller, Silica Gel, Simulation, Performance

Introduction

Adsorption refrigeration method as the green refrigeration technology has received much attention due to the increase of energy consumption and environment protection. They are powered by heat, such as solar energy, biomass or recovered heat. It contributes to the reduction of CO_2 emissions by using a non-fossil energy sources. Thus, adsorption refrigeration machines have the advantage of being environmentally friendly, with zero Ozone Depletion Potential (ODP) and zero Global Warming Potential (GWP).

The adsorbents are often chosen according to their adsorption capacity and their high thermal conductivity. The adsorption capacity must be higher at low temperature and the equilibrium pressure of the evaporator. On the other hand, it must be low at high temperature of the cycle associated with the equilibrium pressure in the condenser. Since adsorption is a surface phenomenon, the choice is made for the microporous adsorbent with a greater specific surface area (activated carbon, zeolites, silica gel and activated alumina).

The silica gel ($SiO_2.NH_2O$) is amorphous synthetic silica whose pore structure creates, by agglomeration, substantially spherical microparticles of approximately uniform size and between 2 and 20 nm. This structure is responsable for the very large surface area, typically 500-800m²/g. it is hydrophobic and has the advantage of adsorbing moisture. The silica gel has a high adsorpion

capacity of water (400 g kg^{-1}) at a water vapor pressure close to saturation, so it is widely used with water as a pair for the cold production.

For this purpose silica gel-water is selected as the best adsorbent pair for adsorption refrigeration applications (Aldadah and Rezk, 2010).

The two-bed silica gel-water adsorption chiller was studied by several researchers (Benelmir *et al.*, 2013; Saha *et al.*, 2001; 2003; Hildbrand *et al.*, 2004; Liu *et al.*, 2005a; Zhai and Wang, 2010; Luo *et al.*, 2010; Rezk and Al-Dadah, 2012; Voyiatzis *et al.*, 2008). Moreover, this adsorption couple is widely studied in the literature. Ghilen *et al.* (2016) evaluated an numerical and experimental study in an experimental platform that has been developed combining thermal solar collectors, a cogenerator, an adsorption chiller and wooden construction (two compartments), they developed a simulation model that is validated with experimental measurements. They presented the time variation of total solar radiation, the temperature variation of chilled, cooling and hot water Temperature of cogenerator and room. Restuccia *et al.* (2004) reported an experimental and numerical study of a laboratory scale adsorption cooling system using an adsorbent composed of silica gel impregnated with $CaCl_2$. At a 90-95°C heat sources, the authors showed that the measured COP values were up to 0.6. With the aim of improving silica gel-water adsorption cooling system design with two adsorption beds,

Nunez *et al.* (2007) depicted the development of a small prototype of an adsorption chiller using silica gel-water pair. In fact, for air-conditioning application of 12-15°C, a cooling COP of 0.5 is found. Xia *et al.* (2008) presented an improved two compartments adsorption heat pump using a silica gel-water pair. The improved chiller is composed of three adsorption beds: one heat pipe working vacuum chamber and two adsorption vacuum chambers. A heat pipe is used to combine the two adsorption beds of the evaporators. A COP of at least 12% was obtained by comparing it to the formers adsorption chillers. Chen *et al.* (2010) studied an improved silica gel-water refrigerating system. Heat and mass recovery process are added to improve the performance of the system. The measured COP is about 0.49. Liu *et al.* (2005b) investigated a new adsorption refrigerating system with mass recovery process, using the silica gel-water as a working pair. The COP range was 0.2-0.42 according to operating conditions.

The aim of this study is the development of a global numerical model using Simulink. To validate our SIMULINK model, we have compared simulation results of temperature profile with those of our experimental results down in Enerbat platform in Nancy France.

Experimental Device

Figure 1 illustrates the experimental platform, driven by solar thermal energy, provided from fuel source such as the natural gas and the solar panels. This unit combines solar cooling, cogeneration and sustainable construction. Two adjacent similar chambers, who need opposite comfort, are the users of heating and refrigeration.

The major components contained in the platform and ENERBAT which are included in the experiments carried out are.

Solar Panel

16 solar collectors, 2.4 m² each is installed on the roof.

Hot Water Tank Thermal Stratification

The heat provided by the co-generator or by the solar panel is stored in the hot water cylinder, to thermal lamination, with a capacity of 1500 liters. The hot water fed from the storage tank to the adsorption chiller.

Dry Cooler

The dry cooler constitutes the cooling circuit of the machine adsorption

Two-Room Climate

It consists of two rooms a warm room and a hot room represents the test cell.

Adsorption Refrigerating Machine

The adsorption machine, SorTech brand, the product chilled water circulating through the cooling ceiling of the cold room. Hot water supplied from the refrigerating machine of the balloon.

Co-Generator

The co-generator is fueled by natural gas. This co-generator is an internal combustion engine coupled to an asynchronous electric generator which upgrades the lubricating oil, the coolant heat and the exhaust gas.

With a thermal output of 4 to 12.5 kW and producing an electric power of the order of 1.3 to 4.7 kW, linked to the electric power produced, the co-generator allows to value 90% of the energy of the fuel Including 25% electricity and 65% heat.

It is a supplementary heat production system in case of low sunlight or high heat demand by the users. It supplies electricity to the refrigerating machine.

Data Acquisition

The hydraulic pumps are piloted and the results are obtained using a Data acquisition which also allows the real-time visualization of the plant overview and the saving of the results.

Working Principle of the Adsorption Chiller

In an adsorption refrigerating system, a Heat Source (HT-source) is needed to heat the adsorbents put in a desorbed. Due to the thermal energy, the water vapor (adsorbate) is desorbed from the silica gel (adsorbent) during the desorption process and condenses in the condenser. The condensate (liquid water) enters in the evaporator and the heat of the chilled adsorbate is adsorbed. The latter supplies the cooling effect. The adsorbate vapor is then on adsorbed by the adsorbents in the adsorber which has been cooled by cooling water (MT-source) so that adsorption occurs (the adsorption process). For a double bed system, two adsorbers (one for desorption and one for adsorption) work alternatively so that a continuous cooling effect is produced in the LT-source (evaporator). Concerning the design of the solar system, the hot water is heated in a solar collector by solar radiation, stored in a Stratified balloon and then transferred by a circulating water pump to the desorber (Fig. 2). The innovation of this design is the silica gel is coated on to the heat exchangers The motivation of using the coating method is to promote heat transfer. Poor contact among these metal surfaces and solids that are exposed to the conventional method causes very big thermal resistance between thermal source and adsorbent, resulting in a large swing of chiller performance due to adsorbent packing density.
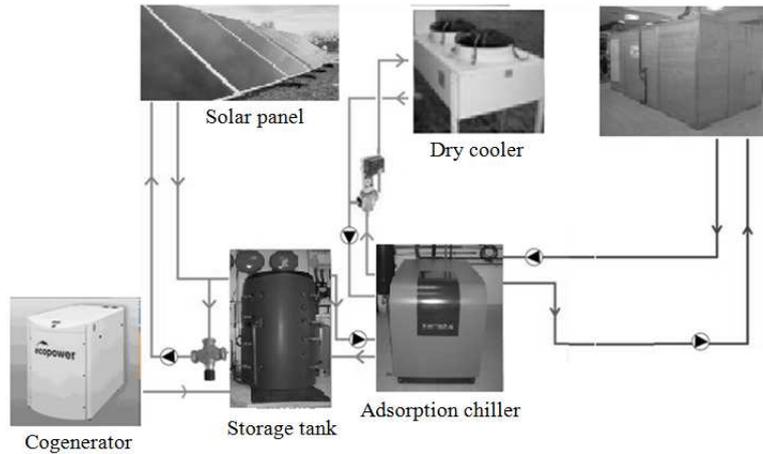
Fig. 1. Experimental device (Enerbat plateform)
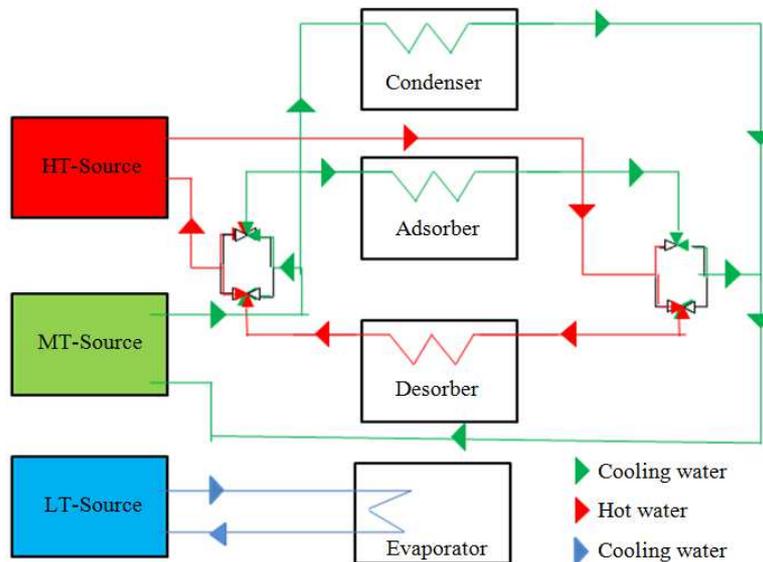
Fig. 2. Schematic diagram of the double bed solar-powered adsorption chiller

Mathematical Model

Assumptions

A number of assumptions are need in order to develop our mathematical model.

- The pressure, temperature and the amount of water vapor adsorbed are uniform throughout the beds adsorption
- All the beds are well-insulated (there is no external heat loss to the environment)
- The liquid water (condensate) can flow into the evaporator easily
- The condensate will flow into the evaporator directly and all desorbed water vapor from the desorber will flow into the condenser immediately

- The condensate will evaporate instantaneously in the evaporator and will be adsorbed in the adsorber immediately
- The adsorbate gas is assumed to be an ideal gas and the adsorbed phase is considered as a liquid
- The thermal resistance between the adsorbent bed and the metal tube is neglected
- Flow resistance arising from the water flowing in the pipeline is neglected
- The properties of the metal tube, the fluid and adsorbate vapor are constant

The dynamic behavior of heat and mass transfer inside different components of the adsorption chiller can be written as showed below according to these assumptions.

Rate of Adsorption/Desorption

The linear driving force kinetic equation is used to calculate the rate of desorption or adsorption, Chihara and Suzuki (1983) have determinate the coefficients of LDF equation for silica gel/water pair and are given below:

$$\frac{\partial w}{\partial t} = K_s (w^* - w) \tag{1}$$

The effective mass transfer coefficient inside the pores K_s is defined as follows:

$$K_s = F_0 \frac{D_s}{R_p^2} \tag{2}$$

The effective diffusivity is given by:

$$D_s = D_{s0} e^{-E_a / RT} \tag{3}$$

Where:

$$D_{so} = 2.54 \ 10-4 m^2 / s, E_a = 4.2 \ 104 \ J / mol$$
$$R_p = 1.7 \ 10^{-4} \text{m}, R = 8,314 \ J / mol \ K, F0 = 15$$

For the silica gel-water pair the equilibrium uptake is estimated using the equation developed by Boelman *et al.* (1995):

$$w^* = 0.346 \left(\frac{P_s(T_r)}{P_s(T_a)} \right)^{1/1.6} \tag{4}$$

where, $P_s(T_w)$ and $P_s(T_s)$ are respectively the corresponding saturated vapor pressures of the refrigerant at temperatures T_r (water vapor) and T_a (silica gel). P_s for water vapor is estimated using the following equation:

$$P_s(T) = 133,32 \exp \left(18,3 - \frac{3820}{T - 46,1} \right) \tag{5}$$

Energy Balance of the Adsorber

The adsorption energy balance can be expressed as:

$$(m_{ad} c_{ad} + m_a c_a + m_a w c p_r) \frac{dT_{ad}}{dt}$$
$$= m_a \Delta H_{ads} \frac{dw}{dt} + m_a c p_{r,v} \frac{dw}{dt} (T_{ev} - T_{ad}) \tag{6}$$
$$+ \dot{m}_{f,ad} c p_f (T_{f,in} - T_{f,out})$$

The outlet temperature of cooling water is described by:

$$T_{ad,out} = T_{ad} + (T_{ad,in} - T_{ad}) \exp \left(-\frac{U_{ad} A_{ad}}{\dot{m}_{f,ad} c p_{f,ad}} \right) \tag{7}$$

Energy Balance of the Desorber

The desorption energy balance is can be expressed as:

$$(m_{de} c_{de} + m_a c_a + m_a w c p_r) \frac{dT_{de}}{dt}$$
$$= m_a \Delta H_{ads} \frac{dw}{dt} + \dot{m}'_{f,de} c p_f (T_{f,in} - T_{f,out}) \tag{8}$$

The outlet temperature of hot water is described by:

$$T_{de,out} = T_{de} + (T_{de,in} - T_{de}) \exp \left(-\frac{U_{de} A_{de}}{\dot{m}_{f,de} c p_f} \right) \tag{9}$$

Energy Balance of the Condenser

The condenser energy balance equation can be expressed as

$$(m_{r,cd} c p_r + m_{cd} c_{cd}) \frac{dT_{cd}}{dt}$$
$$= -m_a \frac{dw_{des}}{dt} L_v - m_a c p_{r,v} \frac{dw_{des}}{dt} (T_{de} - T_{cd}) \tag{10}$$
$$+ \dot{m}_{f,cd} c p_f (T_{f,in} - T_{f,out})$$

The outlet temperature of cooling water can be written as:

$$T_{cd,out} = T_{cd} + (T_{cd,in} - T_{cd}) \exp \left(-\frac{U_{cd} A_{cd}}{\dot{m}_{f,cd} c p_{f,cd}} \right) \tag{11}$$

Energy Balance of the Evaporator

The energy balance in the evaporator can be written as:

$$(m_{ev} c_{ev} + m_{r,ev} c_{p,r}) \frac{dT_{ev}}{dt}$$
$$= -m_a \frac{dw_{ads}}{dt} L_v - m_a \frac{dw_{des}}{dt} c_{p,r} \tag{12}$$
$$(T_{cd} - T_{ev}) + \dot{m}_{f,ev} c_{p,f} (T_{f,in} - T_{f,out})$$

The outlet chilled water temperature can be written as:

$$T_{ev,out} = T_{ev} + (T_{ev,in} - T_{ev}) \exp(-\frac{U_{ev} A_{ev}}{\dot{m}_{f,ev} c p_{f,ev}}) \tag{13}$$

Mass Balance in the Evaporator

The mass balance for the refrigerant (the gas phase is neglecting) can be expressed as:

$$\frac{dm_{r,ev}}{dt} = -m_a \left(\frac{dw_{ads}}{dt} + \frac{dw_{des}}{dt} \right) \tag{14}$$

where, m_a is the adsorbent (silica gel) mass.

System Performance Equation

The COP value is expressed by:

$$COP = \frac{Q_{ev}}{Q_{de}} \tag{15}$$

The cooling capacity of the chiller is defined by the following equation:

$$Q_{ev} = \frac{\int_0^{t_{cycle}} \dot{m}_{f,ev} cp_f \left(T_{ev,in} - T_{ev,out} \right) dt}{t_{cycle}} \tag{16}$$

Where:

$$Q_{de} = \frac{\int_0^{t_{cycle}} \dot{m}_{f,de} cp_f \left(T_{de,in} - T_{de,out} \right) dt}{t_{cycle}} \tag{17}$$

The specific Cooling Power is expressed by:

$$SCP = \frac{Q_{ev}}{m_a} \tag{18}$$

Where :

m_a	=	50 Kg
ΔH_{ads}	=	2800
L_v	=	2500
Ccd, Cev, C_{ad}	=	0.386 kJ
$Cp_{r,v}$	=	1.85 kJ

$0.924\ kJ/kg.K, 4.18\ kJ/kg.k, \dot{m}_{f,ad} = 1.6\ m^3/h, \dot{m}_f\ 3.7$

$m^3/h, \dot{m}_{f,ev} = 2\ m^3/h, T_{ev,in} = 15°C, T_{cd,in} = 22°C, T_{de,in} = 62°C$

Results and Discussion

Model Validation

Simulink is a dynamic solver that uses mathematical and signal blocks, it is more robust to use for the analysis of adsorption chiller.

The schematic diagram of the Simulink model is shown in Fig. 3 for the two beds adsorption chiller it is consisting of five main subsystems describing the governing equations for the condenser, adsorber beds, evaporator and the overall mass balance of the chiller.

The temperature profiles of the cooling, hot and chilled water for the duration of 190s is shown in Fig. 4. A good agreement between numerical and experimental results is noticed, for an hot water inlet temperature of 74°C, the outlet hot water temperature varies between 60 and 71°C, we have a desorption time of 150 s and a preheating time of 40 s.

The evolution of the temperature inside desorber (Fig. 5) is measured by means of probes, at the beginning of cycle the compartment was adsorber its temperature is 27.5°C and then its increases due to the heating by a driving hot water from the solar sensors, the temperature reached by the desorber at the end of a half cycle is 72°C, Good agreement between experimental and numerical results.

Figure 6 shows the experimental and numerical inlet/outlet temperature profile of the heat transfer fluid obtained for the above mentioned standard conditions. The outlet temperature of hot water is nears her inlet temperature, after 7 min, at this point we have a minimum of heat consumption: the silica gel had heated thoroughly to allow desorption of water vapor. Whereas, the inlet cooling water temperature increases during the precooling and preheating process and decreases during the condensation/desorption process,, because during the condensation/desorption process it is necessary to extract an amount of energy to enable the condensation of desorbed refrigerant (water). While evaporator temperature (chilled water temperature), the outlet temperature increases during the evaporation/adsorption process and decreases during the pre-cooling and preheating process (switching time), because during the evaporation phase, the heat to evaporate the refrigerant and cold production are needed.

The refrigerant temperature inside the compartments at the same conditions is shown in Fig. 7. The temperature of both adsorbent beds are alternated, the adsorption temperature is about 32°C and the desorption temperature is about 60°C. The condenser temperature decreases during the desorption/condensation, because the refrigerant had cooled by the cooling system reaches its minimum value when the condensation is finished because the refrigerant is perfectly cold and heat is released through the dry cooler. During the switching phase (precooling/preheating) the condenser temperature increases, this is due to the accumulation of heat in the condenser. The evaporation of refrigerant is produced at 10°C, during the adsorption/evaporation phase the evaporator temperature increases; this is explained by the heat provided of the chilled water circuit which allows the evaporation of refrigerant in order to produce cold. The latter decreases during the switching phase (preheating/precooling): This is due to the dissipation of heat in the evaporator, since there is more refrigerant within the evaporator during this phase.
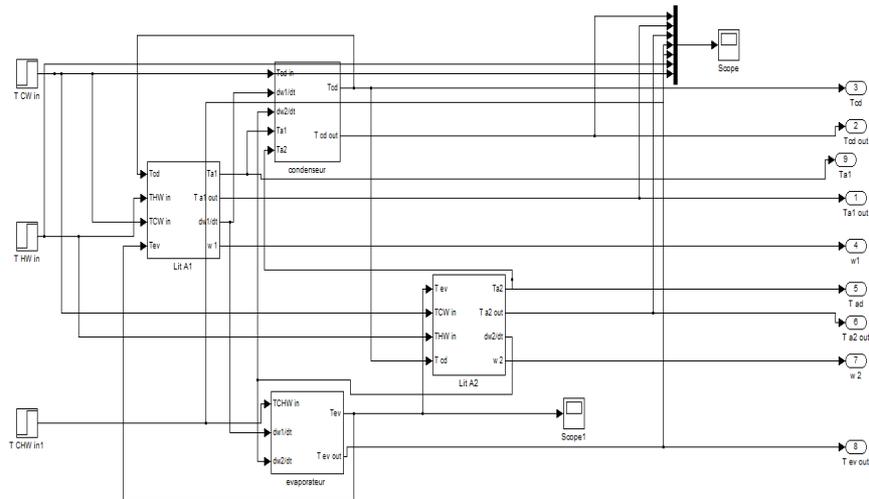
Fig. 3. Arrangement of chiller component in Simulink simulation environment
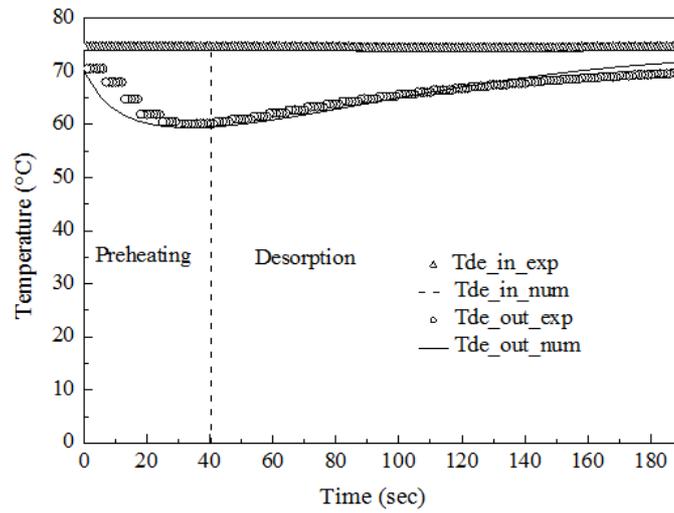
Fig. 4. Desorber temperature variation inputs outputs for half cycle
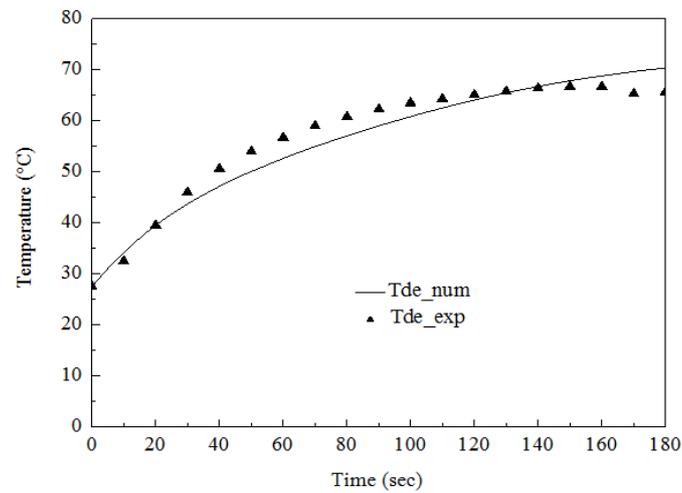
Fig. 5. Temperature of the desorber bed for a half cycle
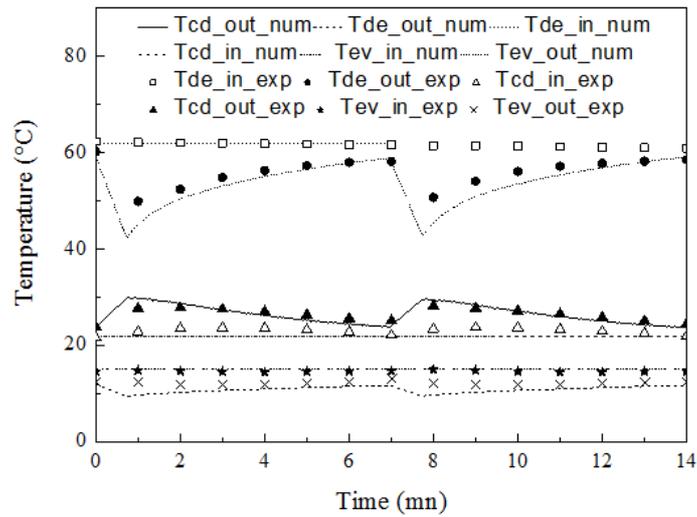
Fig. 6. Overall outlet temperature profile of heat transfer fluid for two beds adsorption chiller
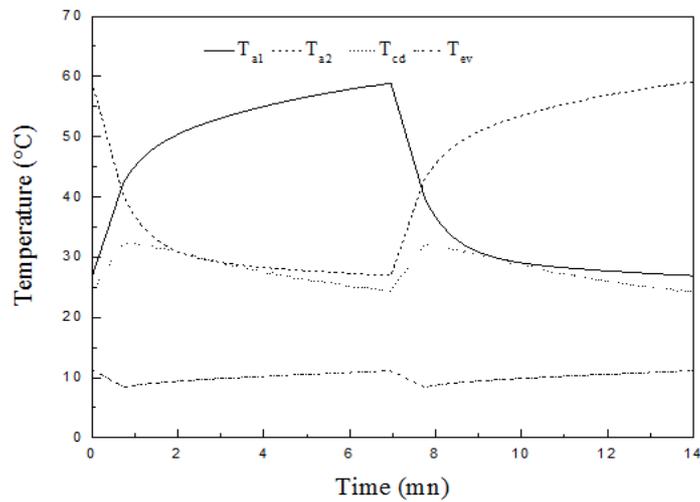
Fig. 7. Temperature profiles for adsorption beds, condenser and evaporator. (Tde_in = 62,15°C, Tref_in = 22°C, Tev_in = 15°C)
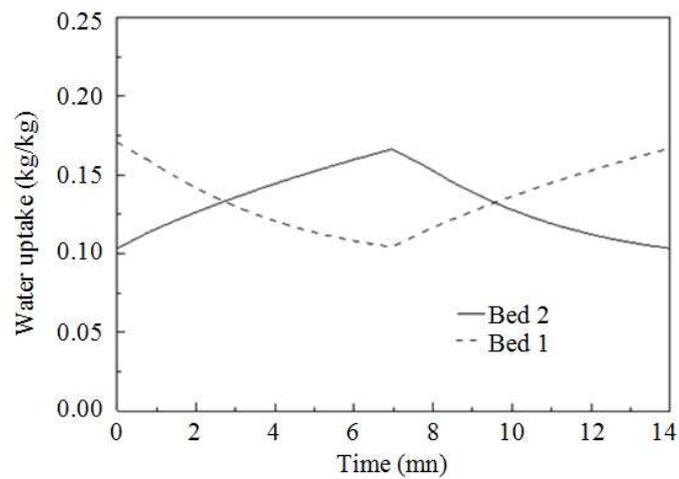
Fig. 8. The time variation of water content in two adsorbers. (Tde_in = 62,15°C, Tref_in = 22°C, Tev_in = 15°C)
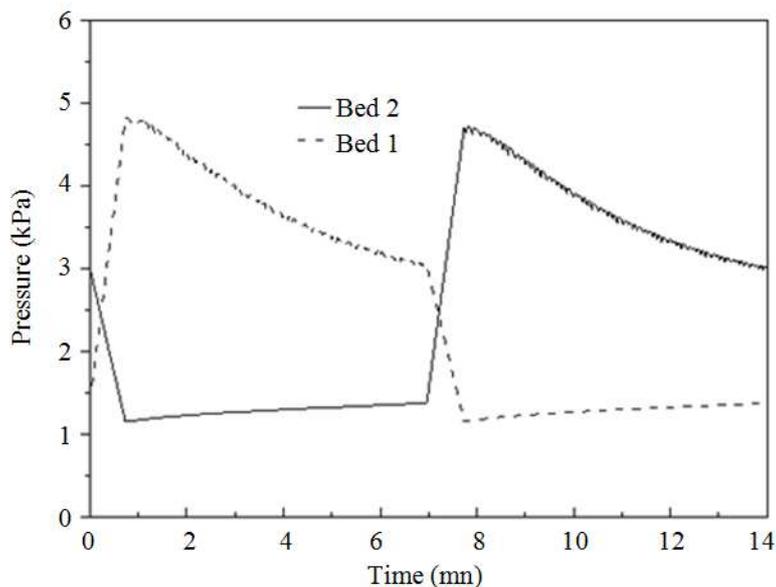
Fig. 9. Refrigerant saturation pressures in the two compartments during a cycle (Tde_in = 62,15°C, Tref_in = 22°C, Tev_in = 15°C)

The evolution of water content is shown in Fig. 8. Desorption attained equilibrium value of 0,166 kg/kg, for duration of 7 mn and reaches its equilibrium adsorption value of 0.104 after 7 min.

The Fig. 9 shows the variation of refrigerant saturation pressure in the bed 1 and bed 2, it is noted that the adsorption pressure is equal to the evaporation pressure, in the range of 1.155 to 1.37 kPa respectively at the beginning and the end of adsorption and the desorption pressure is equal to the condensation pressure which is respectively of 4.63 and 3 kPa at the beginning and the end of desorption. This change in pressure is caused by temperature changes.

Conclusion

This work presents a solar adsorption chiller; silica-gel-water is used as working pair. It has the advantage of total autonomy and continuity in cold production since it causes no environmental effect and it is driven by a free renewable energy. We have been developing a numerical model for simulating the mass and heat transfer of the regeneration and adsorption processes in the two beds. The technology platform ENERBAT provided experimental measurements allowing the tuning of a simulation model. The model is global and allows following the variation of the physical parameters as the temperature and pressure in every component of the machine as well as the mass of water vapor adsorbed during the cold production. The numerical results from this model correlated well with experimental measurements.

Acknowledgement

Author's Contributions

Ghilen Najeh: Designed the research plan, organized the study, create discussion and contributed to the writing of the article.

Messai Souad and Gabsi Slimane: Develop the methodology used in this study, correcting references, article review and the literature.

El Ganaoui Mohammed and Benelmir Riad: Designed the research plan and organized the study.

Ethics

Authors should address any ethical issues that may arise after the publication of this manuscript.

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Nomenclature

A	Heat transfer area, m^2
COP	Coefficient of performance of the machine
Cp	Specific heat, $kJ.(kg.K)^{-1}$
DS0	Coefficient, $m^2.s^{-1}$
E	Heat exchanger efficiency
Ea	Activation energy, $kJ.kg^{-1}$:
L	latent heat of vaporization, $kJ.kg^{-1}$
m	Mass, kg
m'_f	Mass flow rate, $kg.s^{-1}$
ΔH	Isosteric heat of adsorption, $kJ.kg^{-1}$
P	Pressure, Pa
Q	Power, kW
SCP	Specific cooling power, $kW.kg^{1-}$
t	Time, s
T	Temperatures, °C
U	Overall conductance, $W.(m^2.K)^{-1}$
w,w*	Instantaneous Uptake, Equilibrium uptake , kg of réfrigérant.(kg of adsorbant)$^{-1}$

Subscripts

a	Adsorbent
ad	Adsorber
ads	Adsorption
cd	Condenser
Cycle	Cycle
de	Desorber
ev	Evaporator
f	Coolant
in	Inlet
j	Coolant indice
max	Maximum
min	Minimum
num	Numerical
out	Outlet
r	Refrigerant
r,v	Refrigerant vapor
v	Vapor