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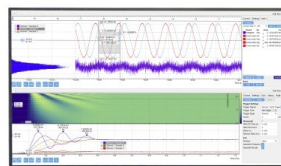
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Indigenous Fruits and Vegetables Storage Using continuous Adsorption Refrigeration: A System Modelling

Kolthoum Missaoui^{1, 2,a}, Slimane Gabsi¹, Nader Frikha¹, AbdelhamidKheiri²,
Mohammed El Ganaoui³

¹Gabès University, Energy, water, Environment & Processes (LR18ES35), ENIG, Gabès, Tunisia

²Lorraine University, CNRS, LEMTA, F-54000 Nancy, France

³Lorraine University, INRA LERMAB, F-54400 Longwy, France

^aCorresponding author: kolthoummissaoui@gmail.com

Abstract. The present work proposes a continuous solar adsorption refrigerator for a positive cold storage room of 231 m³ used for fruits and vegetables preservation and installed in the south of Tunisia. After the estimation of the storage room cooling load which is equal to 13 kW, a parametric study, aims to improve the performance of the chiller is done in order to achieve the storage room need and to estimate the required heat in the refrigerator generator. A theoretical model, based on the thermodynamics of the adsorption process, heat and mass transfer within the porous medium and energy balance in the system components, is developed and a simulation with Matlab/Simulink, is carried out. The model was established and validated with the literature experimental data.

A maximum coefficient of performance of 0.52 and cooling load of 22 kW are reached at cycle time of 800 seconds. The inlet temperatures of heating, cooling and chilled water are 85°C, 22°C and 15°C, respectively.

Keywords: Cold room; adsorption refrigeration; continuous; cooling load; simulation; optimization; performance

I. INTRODUCTION

The cooling processes achieved by the conventional refrigeration machines consume a lot of energy, especially when it is hot. As reported by International Institute of Refrigeration the energy consumption of refrigeration and air-conditioning processes of various system kinds present 15% of the world's energy[1].

Due to the scarcity of electricity, the agriculture field in Tunisia suffers from problem of storage of natural product especially fruits and vegetables. Tons of products are wasted every year. Furthermore, the classical technologies use CFCs or HCFCs, which have a negative impact on the environment due to their undesirable Ozone Depletion Potential (ODP) and Global Warming Potential (GWP) levels.

In this context, thermally driven refrigeration systems using renewable energies and especially solar energy attract currently considerable attention. The coincidence between the needs of cooling power and the availability of the solar irradiation in hot regions and during hot periods makes solar cooling system as a great solution to overcome these problems without contributing neither to global warming nor to the generation of polluting gases.

Among the different solar refrigeration technologies which are presented in **FIGURE 1** adsorption systems are a suitable choice since they can be driven by low-grade heat sources and because of their environmental impact like the use of environmentally friendly refrigerants.[2] However, these systems present some drawbacks, solar adsorption refrigeration chillers generally work with the basic simple cycle which uses a single, intermittent bed, and which has a low efficiency and low cooling effect so it constitutes barriers to a more use of this technology. Numerous attempts have been made in the literature to develop and design a continuous operation and to increase cycle performance and

efficiency. The application of low cost silica gel-water adsorption cooling systems has attracted attention for on farm storage. However, the drawback till now was the low cooling performance.

In order to improve the performance of adsorption refrigeration systems, **Makgafele et al** [2] demonstrated experimentally the capacity of a discontinuous single bed silica gel adsorption chiller working in optimum operating conditions to store fresh mangoes during 21 day.

Therefore, this work focused on improving the cooling performance of a continuous adsorption chiller at different cycle times, various heat water regeneration temperature and cooling inlet temperature in order to demonstrate its capacity to drive a cold storage room used for fruits and vegetables conservation after predicting its required frigorific capacity.

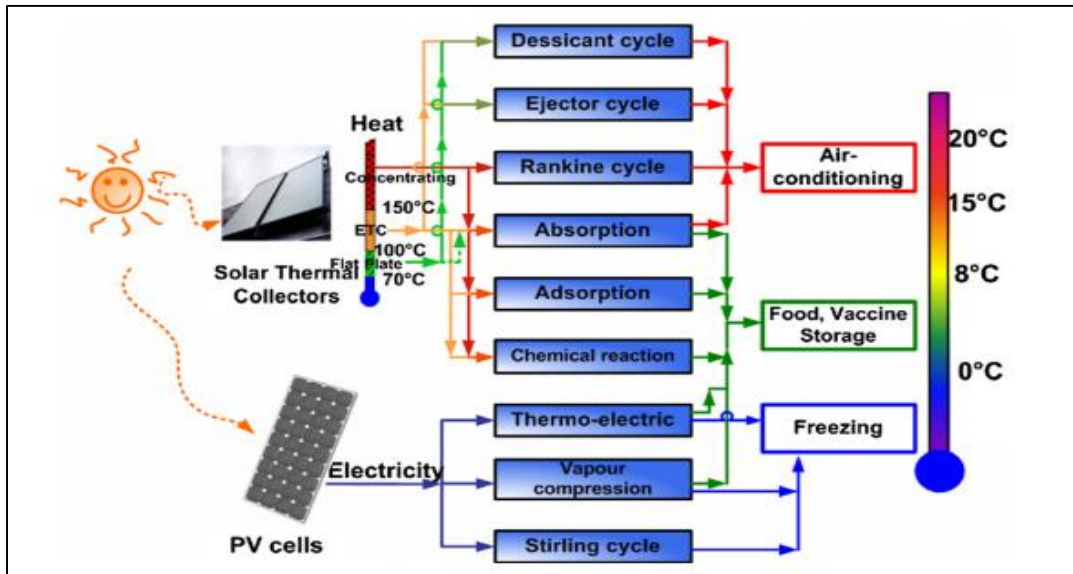


FIGURE 1. Solar cooling paths

II- COLD STORAGE ROOM

1) Cooling Load Calculation

The system's cooling load is the total heat that must be removed from the interior of a cold room. It is calculated taking into consideration heat transfer through the walls, the stored products and the package.[3]

The major heat sources are:

- The heat transfer rate through the walls surrounding the cooled space, the ceiling and the floor.
- The heat transfer rate of air infiltrations.
- The heat transfer rate of the stored product.
- The heat transfer rate from machines (fans, light, rolling stock...).
- The heat transfer rate due to people working inside the room.

The cooling capacity to be installed is therefore the sum of all of this heats which must be increased by 15% to take account of incalculable contributions and changes that the refrigeration installation must also support.

Assuming that there is no particular radiation contribution or additional ventilation to be expected and based on the ASHRAE method (American Society of Heating, Refrigerating and Air-Conditioning Engineers) as presented in [4][5][6] a mathematical model was established in order to determine the cooling load of the refrigeration system .

2) Cold Room Working Conditions

In this study we estimated the cooling load of a cold storage room located in Sfax, southern zone in Tunisia, which was used to store fruits and vegetables particularly some kinds of melon during summer months.

FIGURE 2 presents the daily ambient temperature in Sfax for twelve months. This graph was obtained using the Meteonorm 7.1.3 program (Meteonorm, 2018). As shown this region has a high ambient temperature mostly in summer.

The cold room's dimensions were $7 \times 6 \times 5.5\text{m}^3$ (231m^3) and was used to store 20 tons of melon at optimum storage conditions, temperature between 9 and 13°C and relative humidity of 90% which aim to ensure the highest storage life of this fruit. [7]

The type of storage used in this room is storage in wooden boxes with a capacity of 30 kg stacked on wooden pallets disposed in blocks.

Three persons work inside the cold room for handling purposes during an average period of 2 hours per day.

1) Calculation Result

According to the Tunisian National Institute of Meteorology, the average ambient temperature in Sfax for the summer months is about 35°C . Therefore, an average ambient temperature of 35°C and a storage temperature of 9°C are considered.

A MATLAB code was established to predict the cooling load needed for the refrigeration system, we found that, an average power of 13 kW is enough for the chamber operation. **FIGURE 3** shows the heat gains distributions in the storage room

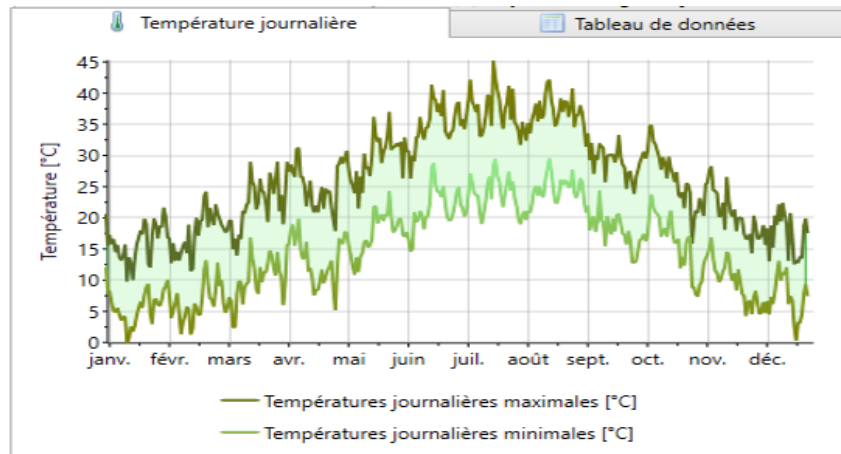


FIGURE 2. Daily temperature in Sfax (Southern of Tunisia) during a year.

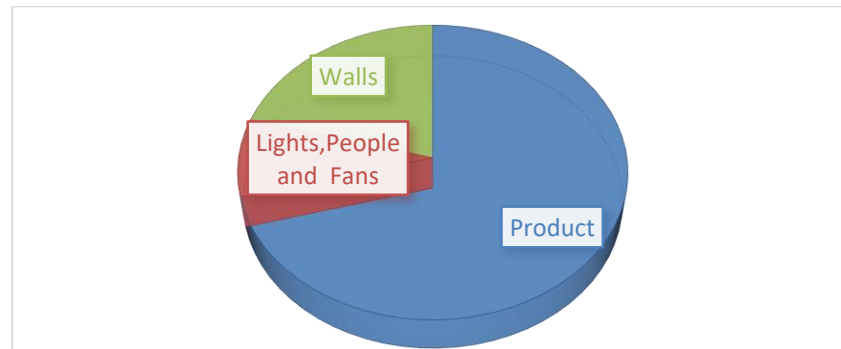


FIGURE 3. Cooling load distribution in the cold room.

III- TWO BED ADSORPTION REFRIGERATION CHILLER

Working Principal

FIGURE 4 (a) shows a schematic diagram of a simple solar driven single-stage 2-bed adsorption chiller, the heat source is solar radiation, which will be absorbed by solar thermal collector and then transferred to the adsorption refrigeration chiller.

A simple two-bed adsorption refrigeration cycle consists of an evaporator, a condenser, an expansion device, and a pair of adsorption beds (adsorber or desorber based on operating mode) which are packed with a desiccant material (silica gel) to desorb/adsorb the refrigerant (water).

The operation of an ideal adsorption chiller starts with the switching period which, consists as presented in **FIGURE 4** (b) of an isosteric preheating (1-2) of the desorber (bed1) by solar hot water allowing the increase of its pressure and isosteric precooling (4-3) of adsorber (bed2) by the cooling water to decrease its pressure. This phase continues until the pressures of bed 1 become almost equal to the condensing pressure and the pressure of the second one become almost equal to the evaporation pressure. During this phase, all the valves are closed to keep the amount of refrigerant in the reactors constant.

The heating process is continued and the valve between bed 1 and the condenser is opened, therefore the bed pressure remains constant (isobaric desorption process (2-3)) and the released superheated refrigerant (water) goes to the condenser.

Hence, the refrigerant is condensed in the condenser by transferring its heat to the cooling medium (cooling water). The saturated liquid refrigerant is expanded in an expansion device and then flows through the evaporator to transfer the heat from a chilled water (cold production phase).

As desorption process occurs in bed 1, the adsorption process takes place in bed 2, cooling process is continued and the valve between bed 2 and the evaporator is opened therefore the bed pressure remains constant (isobaric adsorption process (4-1)) and the refrigerant vapor passes from the evaporator to be adsorbed by bed 2. When the production of cold decreases (saturation of the adsorbent in water vapour), the functions of the two compartments are switched (by opening and closing valves).

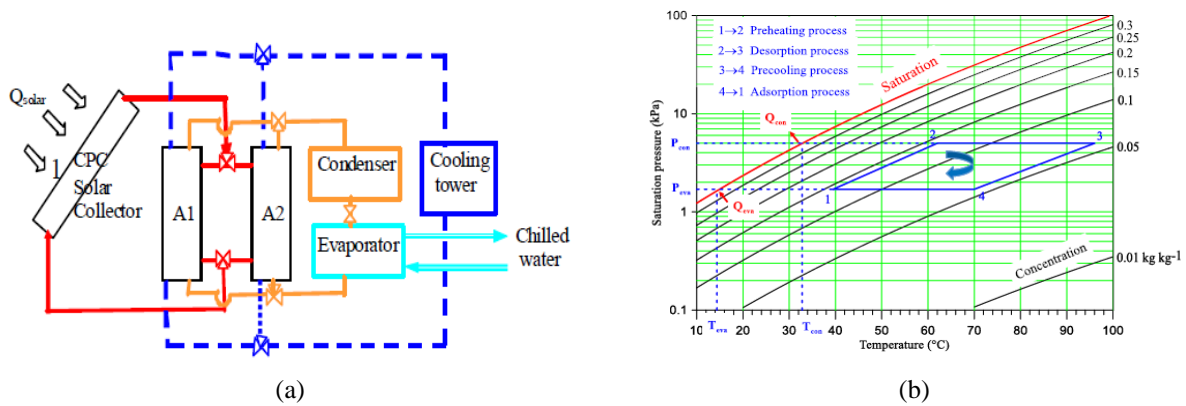


FIGURE 4. Adsorption refrigeration chiller

Mathematical Model

1) Assumptions

In order to simplify the mathematical model, the following assumptions are made.

1. The temperature, pressure and the amount of water vapor adsorbed throughout the adsorber bed are assumed uniform.
2. The water-silica gel granules system is treated as a continuous medium.
3. Resistance to mass transfer in micropores, macropores, and interstitial spaces is neglected.
4. The convection effects inside the porous bed are considered negligible.
5. There is no external heat loss to the environment as all the beds are well insulated.

6. The condensate can flow into the evaporator easily.
7. All desorbed water vapor from the desorber will flow into the condenser immediately and the condensate will flow into the evaporator directly.
8. The condensate will evaporate instantaneously in the evaporator and will be adsorbed in the adsorber immediately.
9. The adsorbed phase is considered as a liquid and the adsorbed gas is assumed to be an ideal gas.
10. The thermal resistance between the metal tube and the adsorbent bed is neglected.
11. The properties of the fluid, the metal tube and adsorbed vapor are constant.

2) Adsorption equilibrium and kinetics

a) Rate of adsorption/desorption

The rate of adsorption or desorption is calculated by the linear driving force kinetic equation, the coefficients of LDF equation for silica gel/water are determined by Chihara and Suzuki [8][9]:

$$\frac{dw}{dt} = K_s (w^* - w) \quad (1)$$

Where:

w^* is the equilibrium amount adsorbed, which is calculated by modified Freundlich model as a function of pressure and temperature.

It is defined for silica gel/water as follow:

$$w^* = 0,346 \left(\frac{P_s(T_w)}{P(T_a)} \right)^{1/1.6} \quad (2)$$

$P_s(T_w)$ and $P_s(T_{si})$ are respectively the saturated vapor pressures of the refrigerant at temperatures T_w and adsorbent temperature T_a .

K_s is the effective mass transfer coefficient inside the pores, it is given by:

$$K_s = F_p \frac{D_s}{R_p^2} \quad (3)$$

The effective diffusivity D_s is defined as follow:

$$D_s = D_{s0} \exp(-E_a / RT) \quad (4)$$

b) Water properties

Saturated vapor pressure of water is given by Antoine's equation:

$$P_s = 133.32 \exp\left(18.3 - \frac{3820}{T - 46.1}\right) \quad (5)$$

c) Energy balance equations

- Adsorber/desorber energy balance

$$\begin{aligned} & (m_{des/ad} C_{p_{de/ad}} + m_a C_a + m_w C_{p_r}) \frac{dT}{dt} = \alpha m_a H_{ads} \frac{dw}{dt} + \alpha(1-\beta) m_a C_{p_{r,v}} \frac{dw}{dt} [T_{ev} - T_{ad}] \\ & + \dot{m}_f C_{p_f} (T_{f,in} - T_{f,out}) \end{aligned} \quad (6)$$

The flag α is equal to 0 during the switching mode and 1 during isobaric adsorption/desorption process. β is equal to 0 or 1 during desorption process or adsorption process respectively.

The outlet temperature of the working fluid from the major system components: adsorber, desorber, evaporator and condenser are estimated using the LMTD method as given by Mitra et al.[10]:

$$T_{f,out} = T + (T_{f,in} - T) \exp\left(\frac{-U_{des/ad} A_{des/ad}}{\dot{m}_f C_{p_f}}\right) \quad (7)$$

- Condenser energy balance

$$(m_{cd} \times C_{p_{cd}}) \frac{dT_{cd}}{dt} = -m_a \frac{dw_{des}}{dt} L_V - m_a C_{p_{r,v}} \frac{dw_{des}}{dt} (T_{des} - T_{cd}) + \dot{m}_{cw} C_{p_{cw}} (T_{cw,in} - T_{cw,out}) \quad (8)$$

$$T_{cw,out} = T + (T_{cw,in} - T) \exp\left(\frac{-U_{cd} A_{cd}}{\dot{m}_{cw} C_{p_{cw}}}\right) \quad (9)$$

- Evaporator energy balance

$$(m_{ev} \times C_{p_{ev}} + m_{r,ev} \times C_{p_r}) \frac{dT_{ev}}{dt} = -m_a L_V \frac{dw_{ad}}{dt} - m_a \frac{dw_{des}}{dt} C_{p_r} (T_{cd} - T_{ev}) + \dot{m}_{chw} C_{p_{chw}} (T_{chw,in} - T_{chw,out}) \quad (10)$$

$$T_{chw,out} = T_{ev} + (T_{chw,in} - T_{ev}) \exp\left(\frac{-U_{ev} A_{ev}}{\dot{m}_{chw} C_{p_{chw}}}\right) \quad (11)$$

d) Mass balance

Neglecting the gas phase, the mass balance equation for water is expressed as:

$$\frac{dm_{r,ev}}{dt} = -m_a \left(\frac{dw_{des}}{dt} + \frac{dw_{ad}}{dt} \right) \quad (12)$$

e) System performance

The performance of the adsorption chiller can be determined by the cooling capacity Q_{ev} and the thermal performance coefficient COP_{th} these parameters are calculated for one complete cycle time (t_{cycle}) which is the sum of adsorption time, preheating time, desorption time, and precooling time[11]:

$$Q_{ev} = \frac{\dot{m}_{chw} \int_0^{t_{cycle}} C_{p_{chw}} (T_{chw,in} - T_{chw,out}) dt}{t_{cycle}} \quad (13)$$

$$COP_{th} = \frac{Q_{ev}}{Q_{qn}} \quad (14)$$

$$Q_{qn} = \frac{\dot{m}_f \int_0^{t_{cycle}} C_{p_f} (T_{hw,in} - T_{hw,out}) dt}{t_{cycle}} \quad (15)$$

RESULTS AND DISCUSSION

Simulink Modeling for Adsorption Chiller

As shown the dynamic response of the adsorption cooling systems are governed by a set of ordinary differential equations. Various simulation tools have been reported in literature to simulate the dynamic response of these systems like Modelica, Java language, Fortran, Matlab programming, Trnsys...

Generally, these simulation tools develop defined functions and additional subroutines for the analysis of thermal systems. Simulink is a dynamic solver that uses mathematical and signal blocks with minimum need for additional user coding[12][13]. Therefore, it is more robust for the analysis and comparative studies.

FIGURE 5 shows a schematic diagram of a Simulink model for a two-bed adsorption cooling system consisting of four main subsystems describing the governing equations for the two adsorber beds, condenser and evaporator.

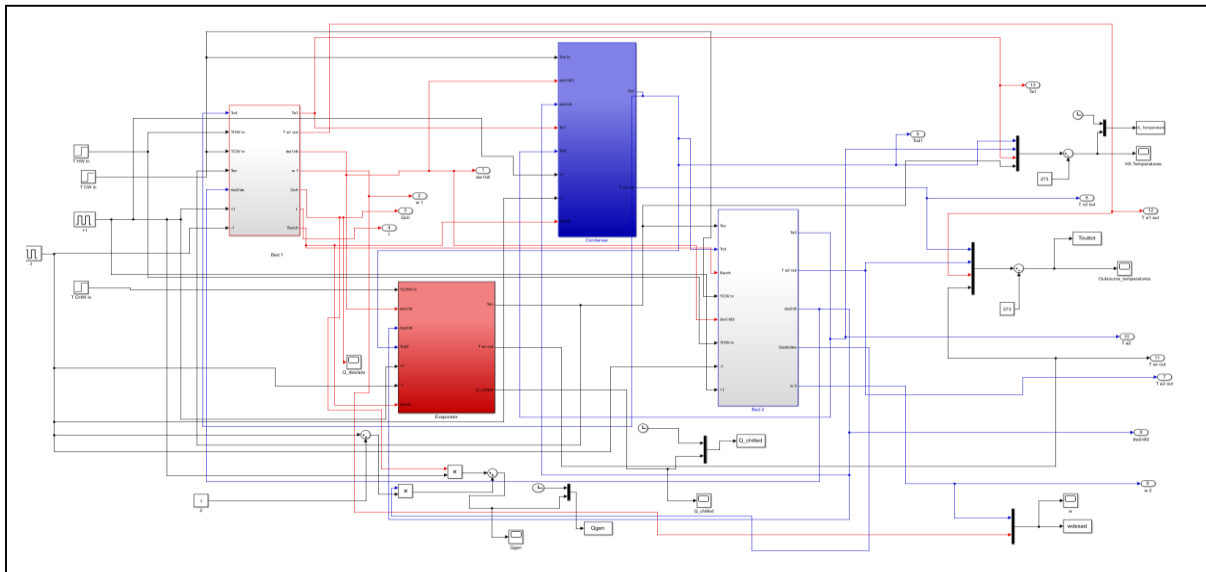


FIGURE 5. Arrangement of chiller component in Matlab/Simulink simulation environment.

Model Validation and Results

The proposed mathematical model was validated with the experimental data from the literature for a two bed adsorption chiller operating with a constant heat source temperature.[14]

FIGURE 6 displays a comparison between the measured values of the water outlet temperatures for the desorber, adsorber, condenser and evaporator with current simulation results. It is clear from this figure that, a good agreement between the experimental and simulation results was reached.

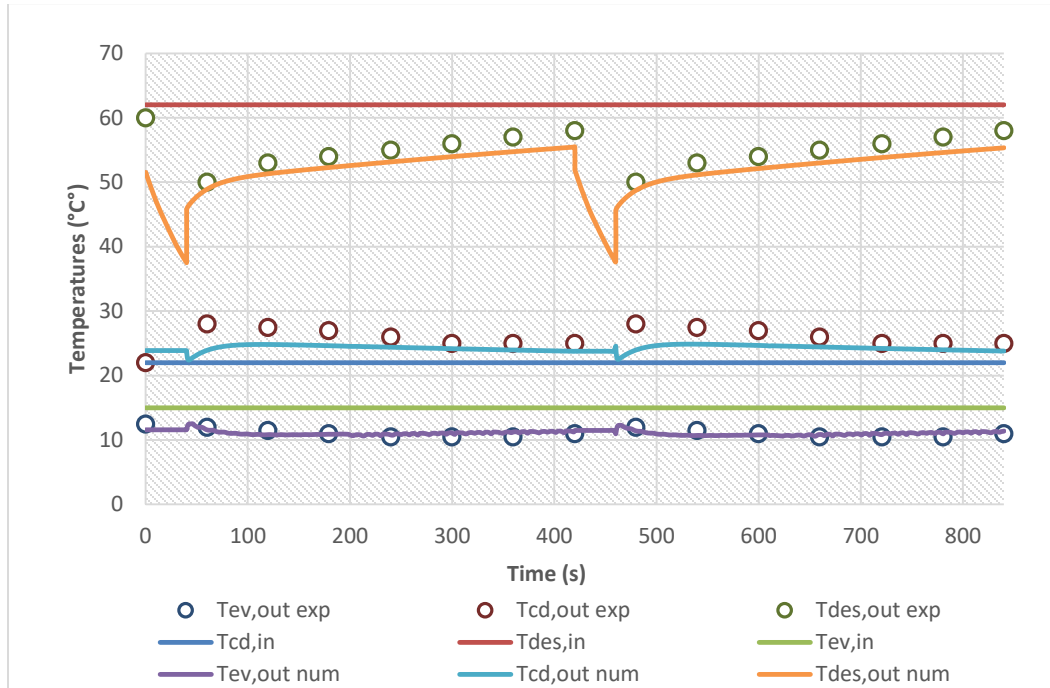


FIGURE 6. Comparison between simulated and experimental water outlet temperature profiles.

Parametric Study

The property values and design parameters used in the simulation model are the same as Chua et al 1999.[8]

The system optimal performance depends basically on three important operating conditions which are the adsorption / regeneration cycle duration, regeneration temperature and cooling water inlet temperature.

In this parametric analysis optimum operating conditions are correlated in a wide range of a hot water inlet temperature (55°C to 95°C), a cooling water inlet temperature (25°C to 40°C) and half cycle time (200 s to 500). whereas chilled water inlet temperature is held constant at 15 °C.

The initial operating conditions of the two-bed adsorption chiller are listed in **TABLE 1** .

TABLE 1.Cycle standard operating conditions

Physical parameter	Value	Units
Chilled water supply	15	°C
Hot regeneration water	85	°C
Cooling water temperature	30	°C
Chilled water flow rate	0.7	Kg/s
Hot water flow rate	1.3	Kg/s
Condenser cooling water flow rate	1.6	Kg/s
Adsorber cooling water flow rate	1.3	Kg/s
Half cycle time	420	Second
Switching time	30	Second

1) Effect of Cycle Time

FIGURE 7 shows the effect of cycle time on the cooling capacity and COP of the chiller at the rated operating conditions. Hot, cooling and chilled water inlet temperatures are kept constant at 85, 30 and 15°C, respectively. The cooling capacity increases with cycle time and has a maximum value of 14.2 kW at a cycle time of 800 s. Then at cycle time longer than 800 seconds it decreases due to a less rate of adsorption because the adsorbent mass approaches to equilibrium conditions. The COP of the chiller increases uniformly with cycle time. The highest values of cooling capacity for silica gel/water pair were obtained when cycle time is between 700 seconds and 900 seconds means when adsorption/desorption time lies between 350 and 450 seconds.

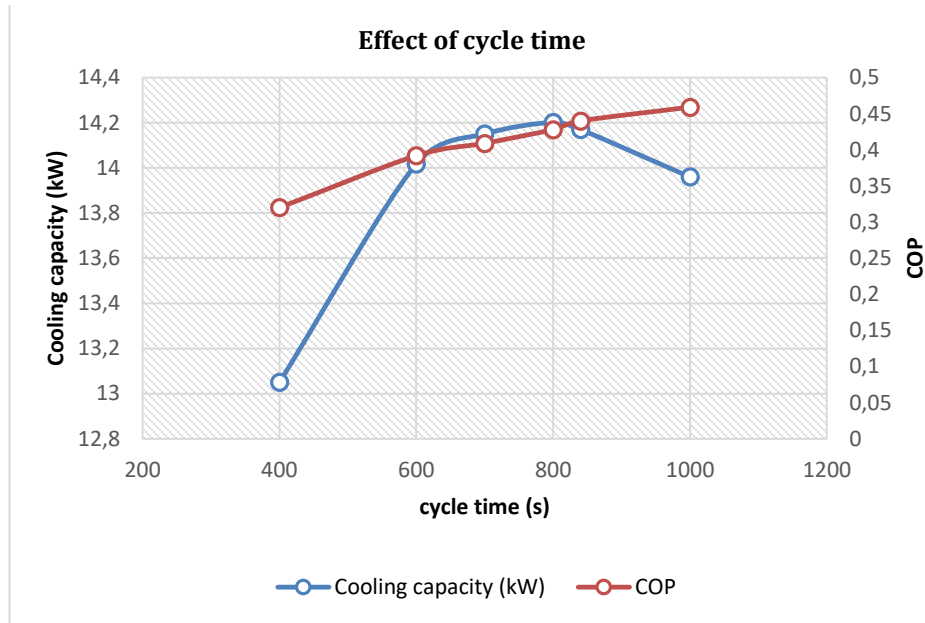


FIGURE 7. Cycle time effects on refrigeration capacity and COP of Silica gel/water adsorption cooling cycle.

2) Effect of Operating Temperatures

a) Effect of Hot Water Inlet Temperature

In order to investigate the effect of the hot water inlet temperature on the system cooling capacity, cooling/chilled water inlet temperatures and half cycle time are taken constant at 25°C, 15°C and 840 s, respectively. As shown in **FIGURE 8** the cooling capacity increases from 4 kW to 16.66 kW as the hot water inlet temperature is increased from 55°C to 95°C. This is because a higher regeneration temperature leads to increase the amount of desorbed refrigerant and the evaporation of more refrigerant. However, the values of COP show an increase for relatively higher heat source temperatures and decrease slightly for hot water inlet temperatures up to 85°C. This means that the heat consumed to desorb the water from the silica gel becomes relatively higher than the cooling effect generated.

b) Effect of Cooling Water Inlet Temperature

The effect of cooling water inlet temperature on both cooling capacity and COP is illustrated in **FIGURE 9**. Obviously, refrigeration capacity and COP decrease with the increase of cooling water inlet temperature. The cooling capacity of the chiller decreases from 20.8 to 4.9 kW while the COP decreases from 0.5 to 0.25 when the cooling water inlet temperature increases from 22°C to 40°C. This is because at lower adsorption temperatures, larger

amount of refrigerant is adsorbed and as a result, a larger amount of refrigerant is desorbed during each cycle and a higher cooling capacity and COP.

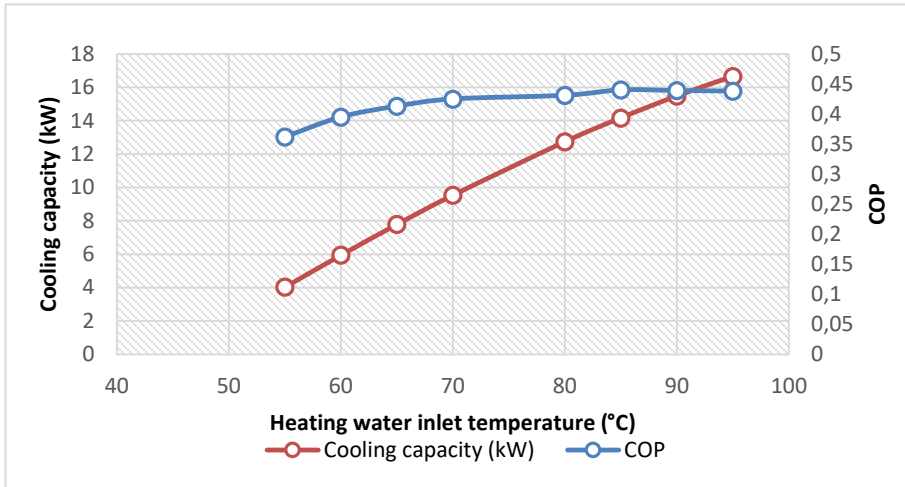


FIGURE 8. Hot water inlet temperature effects on refrigeration capacity and COP.

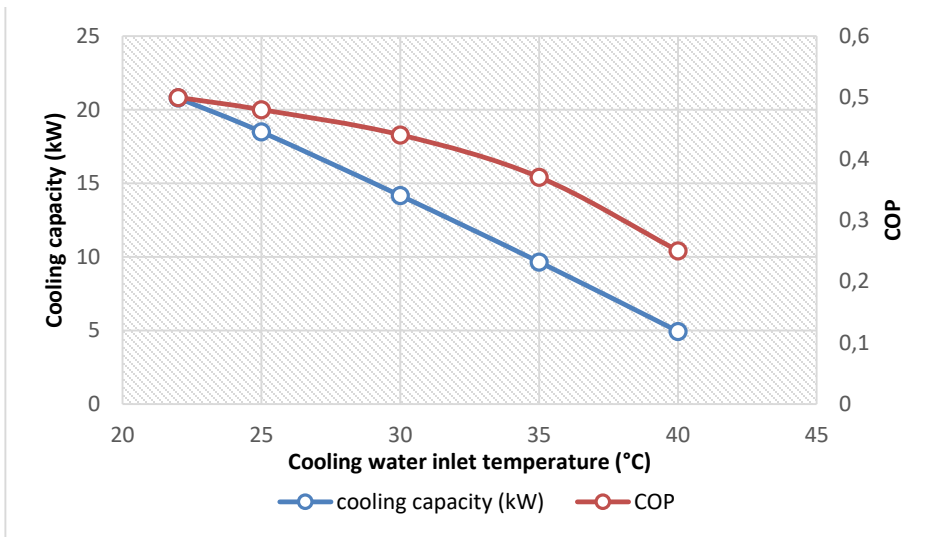


FIGURE 9. Cooling water inlet temperature effects on refrigeration capacity and COP.

3) Outlet Temperature Profile

The outlet temperature profiles of each component of the chiller in optimum conditions (cycle time =400seconds, cooling inlet temperature =22 °C and heat inlet temperature=85°C) are presented in **FIGURE 10**.

In this conditions for a COP equal to 0.52, and cooling load of 22 kW, the heat needed for the generator is 42.5kW.

FIGURE 11 presents the outlet temperature profile in the case of a chiller driven by a variable heating source between 60 and 90 °C.

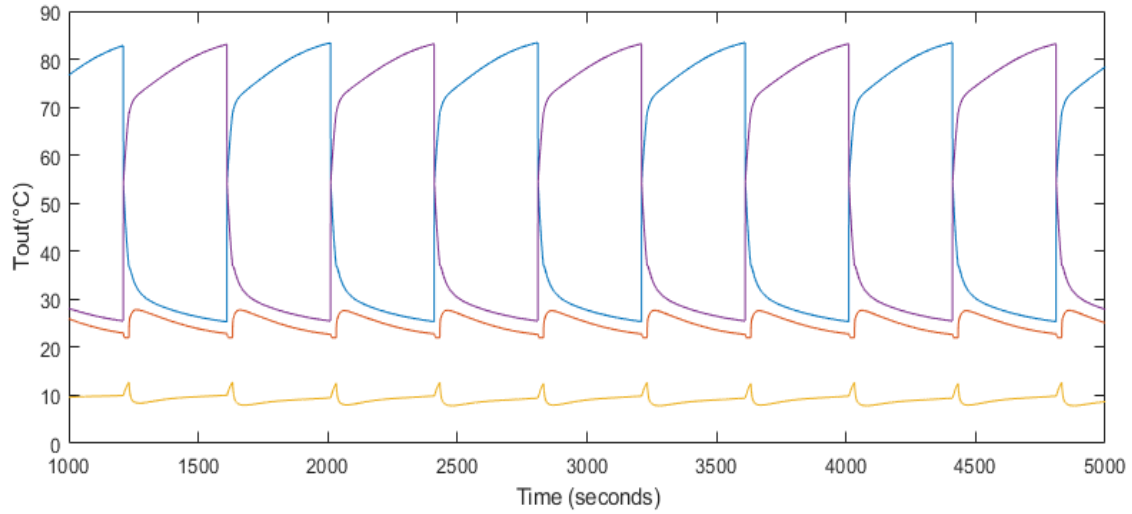


FIGURE 10.Temperature profiles of different adsorption chiller components driven by constant heating source.

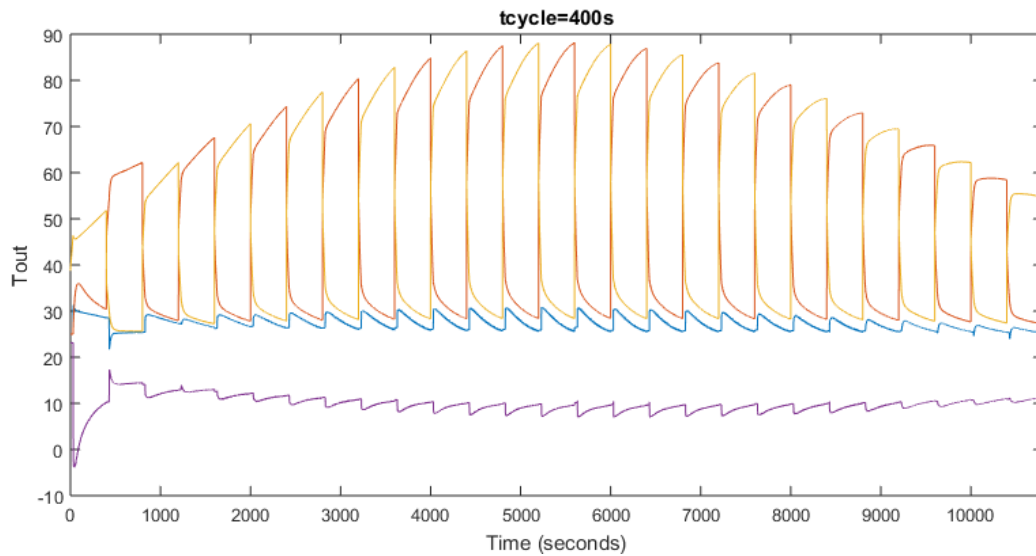


FIGURE 11.Temperature profiles of different adsorption chiller components driven by solar heating source.

CONCLUSION

This research demonstrated the capability of a two bed adsorption chiller to generate low air temperature in a cold storage room destined for preserving fruits and vegetables in the south of Tunisia.

The following conclusions are drawn from the foregoing discussion:

- A parametric study aims to choose the optimum operating conditions should be done to find the best solutions for running an adsorption refrigeration system with high performance.
- A maximum average cooling power of 22 kW is obtained for heat source generator of 42.5 kW.

- The heating and cooling inlet temperatures influence the cooling capacity of the chiller more than cycle time. However, this parameter influences most the reduction in storage temperature, the outlet temperature of the evaporator decreases with the decrease of cycle time duration, However, in order to adopt a high performance adsorption based cooling system we need to find solutions to supplement a high continuous solar heat source.

Nomenclature

A	Area [m ²]
C _p	Specific heat [kJ/kg]
D _{so}	Surface diffusion constant [m ² s ⁻¹]
E _a	Activation Energy [J mol ⁻¹]
H _{ads}	Isosteric heat of adsorption [kJ.kg ⁻¹]
F _p	Particle shape factor
m	Mass [kg]
m _a	Adsorbent mass [kg]
ṁ	Mass flow rate [kg/s]
Q _{gen}	The heat generator input [kW]
Q _{ev}	Cooling capacity of the evaporator [kW]
R	Universal gas constant = 8.314 [J mol ⁻¹ K ⁻¹]
R _p	Particle radius [m]
T	Temperature [K]
t	Time (s)
U	Overall heat transfer coefficient [Wm ⁻² K ⁻¹]
w	Instantaneous uptake [kg/kgads]
w*	Equilibrium uptake [kg/kgads]

Subscript

a	Adsorbant
ad	Adsorber
cd	Condenser
chw	Chilled water
cw	Cooling water to condenser
des	Desorber
ev	Evaporator
f	Heating/Cooling fluid to desorber/adsorber
gen	Generation
in	Inlet
out	Outlet
r	Refrigerant
v	Vapor

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