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Thermal Study and Modeling of the Cold Room of a Solar Adsoption Refrigerator

Amadou Konfe[1], Mahamadi Savadogo[1], Loukman Ouedraogo[1], Ousmane Moctar[2], Sié Kam[1]

[1]Laboratory of Thermal and Renewable Energies, University Joseph Ki-Zerbo, Ouagadougou, Burkina Faso
[2]Sciences and Technologies Faculty, University of Agadez, Agadez, Niger
Email: konfeamadou@gmail.com

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Abstract

In this work, we are interested in the study of the thermal exchanges which take place at the evaporator of an adsorption refrigerator. Due to the cost of designing experimental devices and the impossibility of studying the influence of certain parameters experimentally, an alternative would be simulation. The aim is to provide a model for predicting the thermal behavior of the various elements in the cold room of an adsorption solar refrigerator. A dynamic modelling of the refrigerator taking into account fluid flow, heat and mass transfer phenomena in the cold room was made. The calculation code obtained using COMSOL 5.1 software makes it possible to analyze and study the influence of the various parameters on the performance of the system. In a second step, the theoretical results obtained were compared with the experimental results in order to validate the model. The analysis of the influence of the physical-thermal properties of the insulating material on the temperature of the chamber makes it possible to conclude that a material having a low density ρ, a low thermal conductivity λ and a low specific heat capacity offers better performance to the cold room. Better thermal insulation also implies having a reasonable insulation thickness.

Keywords

Thermal, Evaporating, Adsorption, Cooling, Modeling

1. Introduction

A study conducted by the FAO on food losses and waste (FAO, 2012) indicates that globally a third (1/3) of the world's agricultural production is lost between the period of production and that of consumption due to a lack of means of conservation. To remedy this situation and especially in a context where the field of refrigeration is very energy intensive and very polluting to the environment due to the use of fossil fuels, adsorption systems are positioned as a sustainable solution for these problems. The cold room is a crucial element for the refrigerator because it ensures both the production and the preservation of the cold. When it is badly sized or badly insulated, we see a drop in the performance of the machine. Complete insulation of the cold room could increase the performance of these machines and make them more attractive.

J. Mayor *et al.* [1] have built a machine with materials that minimize system mass. The use of vacuum insulation panels (VIPs) to manufacture the refrigeration casing considerably reduced heat loss, while maintaining a large refrigeration volume in relation to the overall dimensions.

H. Ambarita *et al.* [2] made a study of solar-powered adsorption refrigeration cycle with generator filled by different adsorbents, which has been tested by exposing to solar radiation in Medan city of Indonesia.

Their main conclusion is that for Indonesian condition and flat-plate type solar collector, the pair of activated carbon and methanol is the better than activated alumina.

A. Mostafa *et al.* have conducted a simulation of a solar assisted adsorption refrigeration system utilized in cold stores applications for hot-humid and hot-dry climatic conditions [3].

They found that, based on the annual system performance, Solar Fraction (SF) and Coefficient of Performance (COP) increase as cold store temperature increases for both climatic conditions. In fact, hot-dry climate has higher SF and COP at 13˚C than hot-humid by 6% and 11%, respectively.

Many researchers over the world investigate the performance characteristics of the solar driven ARS working with silica-gel/water pair either theoretically or experimentally [4] [5].

In the experimental investigations, the system has been used in domestic refrigerator, water cooler and ice makers applications with solar COP ranges from 0.055 to 0.23 [3] [6]. Therefore, several theoretical investigations have been found in literature to enhance its lower solar COP.

Currently, research focuses on improving the heat transfer properties of materials on the one hand, and on the other, on the consolidated adsorbents, and a literature review on the subject is provided by [7].

Most of the studies in the literature do not make theoretically or experimentally study of the influence of the thickness of the sideboard on the performance of the refrigerator. The theoretical study initiated in this work, based on the modelling and simulation of the operation of the refrigerator which will offer more leeway.

The aim is to provide a model for predicting the thermal behavior of the various elements in the cold room of an adsorption solar refrigerator.

2. Material and Method

2.1. Material

The material used in this theoretical study is COMSOL MULTIPHYSICS 5.1

software. It is a general software platform that allows the modelling and simulation of physical phenomena that can be described by partial differential equations.

2.2. Method

First, we undertook a dynamic modelling of our refrigerator taking into account fluid flow, heat and mass transfer phenomena in the cold room. Our model is based on the classical foundations of energy and mass conservation where the equations are partial differentials. They are solved by the finite element numerical method using the COMSOL 5.1 simulation software. On this basis, the calculation code makes it possible to analyze and study the influence of the various parameters on the performance of the system.

In a second step, the theoretical results obtained will be compared with the experimental results in order to validate the model.

2.2.1. Dynamic Modeling

Given the limits of maneuvers observed during experimental studies, a three-dimensional (3D) physical model (**Figure 1**) that is the most representative of the chamber will be adopted. It is a rectangular parallelepiped of dimension 120 cm × 150 cm × 100 cm with a proper volume of dimension 90 cm × 120 cm × 70 cm.

2.2.2. Working Hypotheses

The simplifying assumptions on the heat transfers and the physical model are as follows:

- Heat exchanges are three-dimensional;

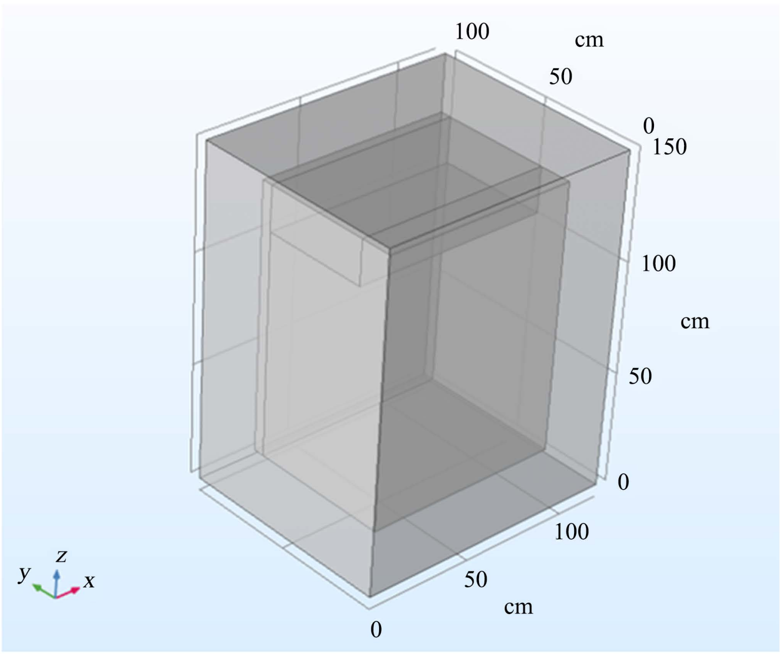

Figure 1. Physical model designed with COMSOL MULTIPHYSICS.

- Heat transfers by radiation are neglected;
- Only the thermal gains through the walls are considered;
- The convective exchanges between the internal walls and the chamber are natural convections;
- The insulating walls are isotropic and made of the same material;
- The temperature is assumed to be uniform at any point on a face;
- The thermodynamic properties of the insulating walls are constant. These are thermal conductivity (λ), density (ρ) and heat capacity (Cp);
- The air flow in the chamber is laminar;
- The air in the chamber is considered as an ideal, homogeneous and incompressible gas;
- The adsorbent/adsorbate couple used is silica gel/water.

2.2.3. Mathematical Modeling

1) Heat transfer equation at the level of the insulating wall

Taking into account the simplifying assumptions, the heat transfer equation at the level of the insulating walls is:

$$\rho_{ins} Cp_{ins} \frac{\partial T}{\partial t} - \lambda_{ins} \left(\frac{\partial^2 T}{\partial^2 x} + \frac{\partial^2 T}{\partial^2 y} + \frac{\partial^2 T}{\partial^2 z} \right) = 0 . \quad (1)$$

The coefficients ρ_{ins}, Cp_{ins}, λ_{ins} are respectively the density, the heat capacity, the thermal conductivity of the insulation.

2) Heat transfer equation at the level of the chamber to be cooled

The hypotheses make it possible to obtain the heat transfer equation in the air:

$$\begin{aligned} &\rho_{air} Cp_{iair} \left[\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right] \\ &- \left[\frac{\partial}{\partial x} \left(\lambda_{air} \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(\lambda_{air} \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(\lambda_{air} \frac{\partial T}{\partial z} \right) \right] = Q \end{aligned} \quad (2)$$

For temperatures between 0˚C and 100˚C, a correlation giving λ_{air} as a function of temperature has been proposed by Yves. JANNOT [8].

$$\lambda_{air} = 7.57 \times 10^{-5} T + 0.02452 \quad (T \text{ in } ^\circ\text{C}) \quad (3)$$

3) Chamber air flow equation

The equation governing the fluid flow including gravity is based on the Navier-Stockes equations [9]:

$$\begin{aligned} &\rho_{air} \left[\frac{\partial U_{air}}{\partial t} + \left(U_{air} \cdot \nabla \right) U_{air} \right] \\ &= \nabla \cdot \left[-p_{air} I + \mu_{air} \left(\nabla U_{air} + \left(\nabla U_{air} \right)^T \right) + \rho_{air} g \right] \end{aligned} \quad (4)$$

$$\rho_{air} \nabla \left(U_{air} \right) = 0 \quad (5)$$

4) Heat and mass transfer equation at the evaporator

The equation reflecting the heat and mass exchanges is given by:

$$Cp_e\left[\frac{\partial(\rho_e T)}{\partial t}+u\frac{\partial(\rho_e T)}{\partial x}+v\frac{\partial(\rho_e T)}{\partial y}+w\frac{\partial(\rho_e T)}{\partial z}\right] -\left[\frac{\partial}{\partial x}\left(\lambda_e*\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\lambda_e*\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(\lambda_e*\frac{\partial T}{\partial z}\right)\right]=Q_f \tag{6}$$

With u, v, w the components of the velocity field in the three directions.

According to Yves. JANNOT [8], we have:

$$\lambda_e=-9.87\times10^{-6}T^2-2.238\times10^{-3}T+0.5536 \quad (T\text{ en }^\circ\text{C}) \tag{7}$$

$$\rho_e=-0.0038T^2-0.05050T+1002.6 \tag{8}$$

The quantity of heat necessary for the vaporization of the quantity $\dot{m}$ of water through the wall of the evaporator according to Alfa Oumar DISSA *et al.* [10], would then be:

$$Q_f=\dot{m}L_v-\dot{m}Cp_e\left(T_{cd}-T_{evap}\right) \tag{9}$$

The equation for mass transfer by convection between the wall of the evaporator and the ambient air of the chamber, by analogy to Newton's equation for heat transfer by convection, is written

$$\dot{m}=k_m S\left(C_{v,sat-e}-C_{v,air}\right) \tag{10}$$

Water vapor, like the air in the room, is considered an ideal gas:

$$C_{v,sat-e}=\frac{P_{v,sat-e}}{R*T_{eq}} \tag{11}$$

and

$$C_{v,air}=\frac{P_{v,air}}{R*T_{eq}} \tag{12}$$

From Equations (11) and (12), Equation (10) becomes:

$$\dot{m}=\frac{k_m S}{R*T_{eq}}\left(P_{v,sat-e}-P_{v,air}\right) \tag{13}$$

with:

$$P_{v,air}=HR*P_{v,sat-e} \tag{14}$$

By replacing Equation (13) and Equation (14) in Equation (9) we have:

$$Q_f=-\left[L_v-Cp_e\left(T_{cd}-T_{evap}\right)\right]*\left[\frac{k_m S}{R*T_{eq}}*P_{v,sat-e}(1-HR)\right] \tag{15}$$

With Equation (6) and Equation (15), we obtain the heat transfer equation combined with the mass transfer at the evaporator:

$$Cp_e\left[\frac{\partial(\rho_e T)}{\partial t}+u\frac{\partial(\rho_e T)}{\partial x}+v\frac{\partial(\rho_e T)}{\partial y}+w\frac{\partial(\rho_e T)}{\partial z}\right] -\left[\frac{\partial}{\partial x}\left(\lambda_e*\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\lambda_e*\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(\lambda_e*\frac{\partial T}{\partial z}\right)\right] =-\left[L_v-Cp_e\left(T_{cd}-T_{evap}\right)\right]*\left[\frac{k_m S}{R*T_{eq}}*P_{v,sat-e}(1-HR)\right] \tag{16}$$

The expressions of L_v and $P_{v,sat-e}$ are given by YVE JANNOT [8]:

3. Results and Discussions

3.1. Temperature Variation in the Chamber and in the Evaporator

Figure 2 shows the evolution of the temperature in the cold room and in the evaporator.

The curves of the evolution of the temperatures of the chamber and of the evaporator are identical (Figure 2). Indeed, the two curves show a brief period of increase at the beginning of the evaporation phase followed by a long period of decrease indicating the production of cold. The growth period is due to the thermal gains of the chamber through the walls during the day. This heating of the chamber also leads to an increase in the temperature of the evaporator because these two compartments exchange heat. There is good agreement between the theoretical (Figure 2) and experimental results obtained by NONGUIERMA *et al.* [11] illustrated by Figure 3.

3.2. Parametric Study of the System

3.2.1. Influence of Insulation Thickness

Increasing the insulation thickness results in lower internal temperatures as shown in Figure 4.

Indeed, the larger it is, the more it blocks the transmission of heat in the chamber through the walls. This keeps the room cool for a long time. However, it is important to make a reasonable choice of thickness (5 to 15 cm) because an insulator having a greater thickness (greater than 20 cm) stores a greater quantity of heat which will be transmitted mainly to the room whose temperature is relatively low compared to ambient.

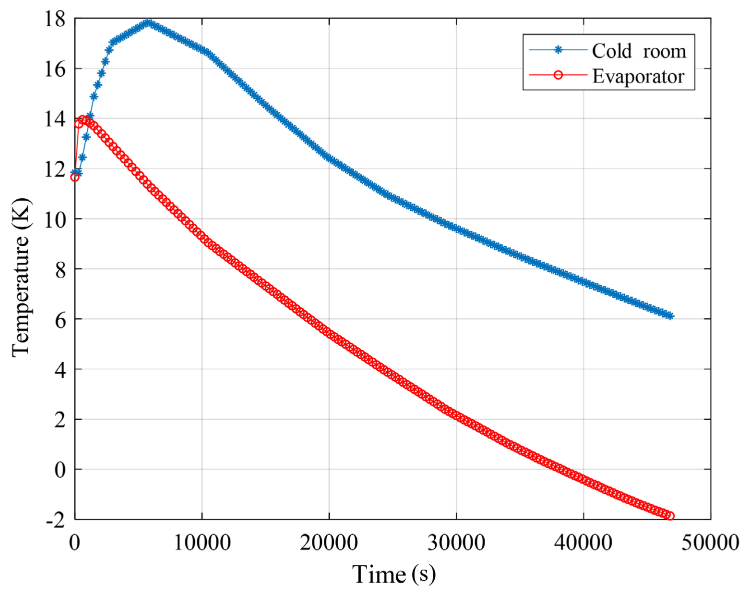

Figure 2. Evolution of the temperature in the chamber and in the evaporator.

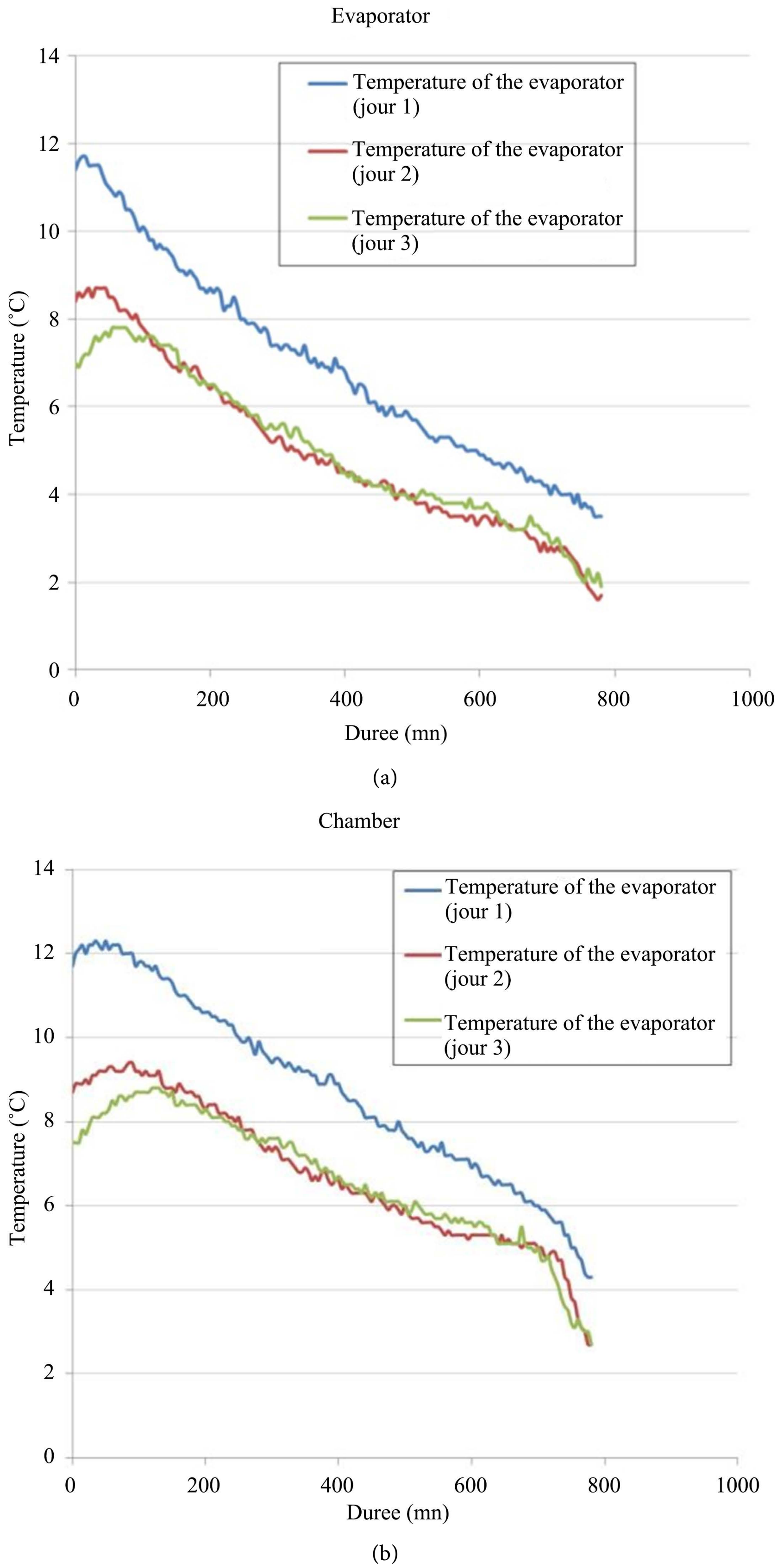

Figure 3. Evolution of the temperatures of the evaporator (a) and the chamber (b).

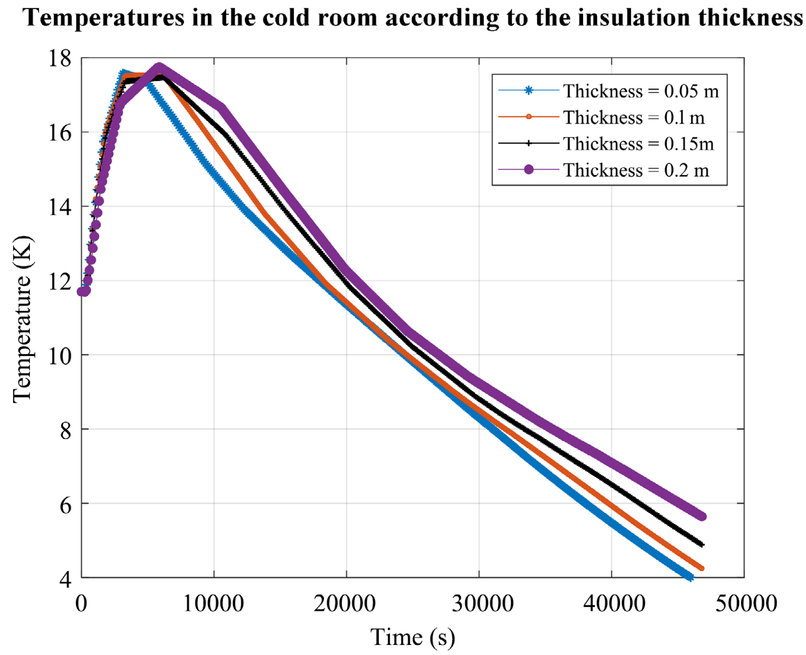

Figure 4. Evolution of the temperatures in the cold room with the insulation thickness.

3.2.2. Influence of the Thermal Conductivity *λ* of the Insulation

Figure 5 shows the influence of the conductivity coefficient on the chamber temperature. In this figure we can see that the temperature in the chamber at the end of the evaporation is low for low values of thermal conductivity. Indeed, an insulator with a thermal conductivity equal to 0.045 (W/m·K) makes it possible to obtain at the end of evaporation a temperature of 6˚C in the chamber, while an insulator with a thermal conductivity of 0.005 (W/m·K) offers 4˚C, *i.e.* a variation of 2˚C. This shows that is a very important property in the operation of the refrigerator. The explanation of this influence lies in the fact that when the thermal conductivity of an insulating wall is low, it has a higher insulation capacity, which leads to a limitation of heat exchange with the external environment.

3.2.3. Influence of the Density *ρ* of the Insulation

Figure 6 illustrates the influence of the density of the insulation on the internal temperature at the end of evaporation. The lowest chamber temperature is obtained at the end of evaporation with the lowest density of the insulation ($\rho = 25$ kg/m^3).

The lower ρ, the better the wall is insulating. Indeed, this situation is explained by the fact that less dense materials contain air molecules. The matter of these materials being separated by the air it contains, can no longer transmit the heat in the chamber by conduction step by step. These types of materials therefore behave as thermal insulators and prevent thermal gains.

3.2.4. Influence of the Specific Heat Capacity *Cp* of the Insulation

The temperature of the chamber at the end of the evaporation is relatively low with the low values of specific heat capacity *Cp* shown in **Figure 7**. Indeed, materials with a low specific heat capacity *Cp* have a low absorption capacity. Thus they heat up slowly, which means that the heat transferred into the chamber is low compared to materials with higher *Cp* values.

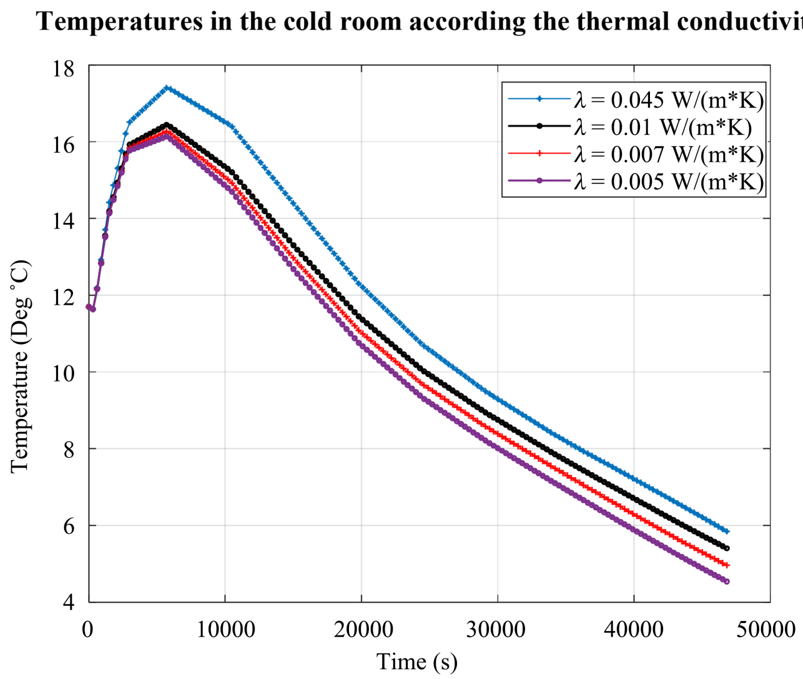

Figure 5. Evolution of the temperature in the chamber according to the thermal conductivity.

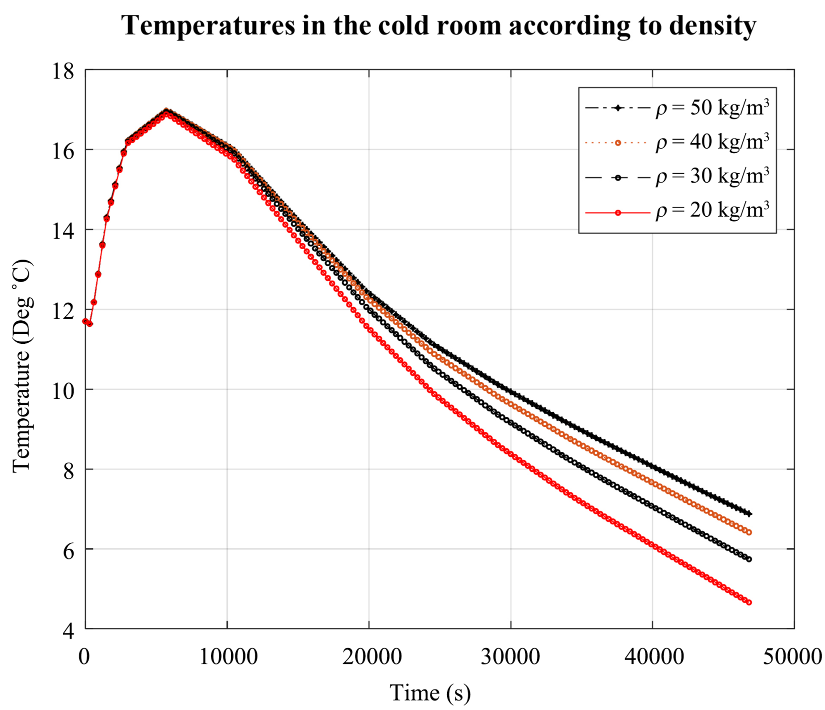

Figure 6. Evolution of the temperature in the chamber according to the density.

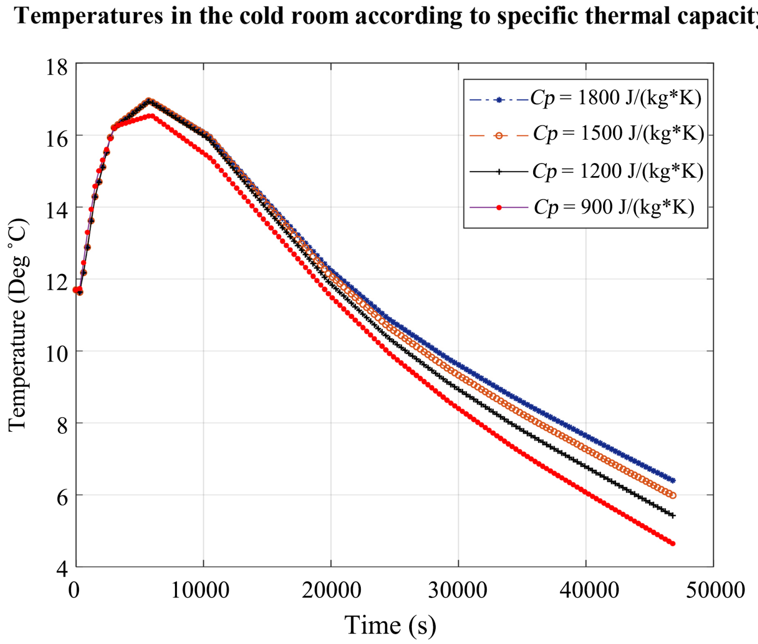

Figure 7. Evolution of the temperature in the chamber according to the thermal capacity.

4. Conclusion

This work presents a modeling of a solar adsorption refrigeration system using the silica-gel/water couple followed by a confrontation of the numerical results with those of experimental. The model developed was the subject of simulations to verify the impact of certain parameters on the performance of the machine. The analysis of the influence of the physico-thermal properties of the insulating material on the temperature of the chamber makes it possible to conclude that a material having a low density ρ, a low thermal conductivity λ and a low specific heat capacity offers better performance. Also, better thermal insulation involves having a reasonable thickness of insulation so as not to create heat stocks in the walls and which will be transmitted mainly to the room, the temperature of which is relatively low compared to the ambient temperature. In the future, it will be useful to carry out an experimental production of the cold chamber using the best-performing materials to confirm this study.

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Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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